
PUMP HANDBOOK

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In memory of our good friends and colleagues

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IGOR KARASSIK, now deceased, was an original editor of this book. Many of his extensive contributions to the earlier editions remain in this edition. A major figure in the pump industry for the greater part of the past century, he also authored six books in this field. Beginning in 1936, he wrote more than 600 articles on centrifugal pumps and related subjects, which appeared in over 1500 publications worldwide. For the greater part of his career, he held senior engineering and marketing positions within the Worthington Pump & Machinery Company, which after a number of permutations became part of the Flowserve Corporation. Igor Karassik received his B.S. and M.S. degrees in Mechanical Engineering from Carnegie Mellon University. He was a Life Fellow of the American Society of Mechanical Engineers and recipient of the first ASME *Henry R. Worthington Medal* (1980).

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SI UNITS— A COMMENTARY

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P. COOPER

Since the publication of the first edition of this handbook in 1976, the involvement of the world in general, and of the United States in particular, with the SI system of units has become common. Accordingly, throughout this book, SI units have been provided as a supplement to the United States customary system of units (USCS).

The designation SI is the official abbreviation, in any language, of the French title “Le Système International d’Unités,” given by the 11th General Conference on Weights and Measures (sponsored by the International Bureau of Weights and Measures) in 1960 to a coherent system of units selected from metric systems. This system of units has since been adopted by the International Organization for Standardization (ISO) as an international standard.

The SI system consists of seven basic units, two supplementary units, a series of derived units, and a series of approved prefixes for multiples and submultiples of the foregoing. The names and definitions of the basic and supplementary units are contained in Tables 1A and 1B of the Appendix. Table 2 lists the units and Table 3 the prefixes. Table 10 provides conversions of USCS to SI units. Conversions can also be found in NIST Special Publication 811 “Guide for the Use of the International System of Units” available at <http://www.nist.gov/metric>.

As with the earlier editions, the decision has been made to accept variations in the expression of SI units that are widely encountered in practice. The quantities mainly affected are pressure and flow rate, the situation with each being explained as follows.

The standard SI unit of pressure, the pascal, equal to one newton* per square meter,[†] is a minuscule value relative to the pound per square inch (1 lb/in² or 1 psi = 6,894.757 Pa) or to the old, established metric unit of pressure the kilogram per square centimeter

*The newton (symbol N) is the SI unit of force, equal to that which, when applied to a body having a mass of 1 kg, gives it an acceleration of 1 m/s².

[†]In countries using the SI system exclusively, the correct spelling is *metre*. This book uses the spelling *meter* in deference to prevailing U.S. practice.

($1 \text{ kgf/cm}^2 = 98,066.50 \text{ Pa}$). In order to eliminate the necessity for dealing with significant multiples of these already large numbers when describing the pressure ratings of modern pumps, different sponsoring groups have settled on two competing proposals. One group supports selection of the kilopascal, a unit which does provide a numerically reasonable value ($1 \text{ lb/in}^2 = 6.894757 \text{ kPa}$) and is a rational multiple of a true SI unit. The other group, equally vocal, supports the bar ($1 \text{ bar} = 10^5 \text{ Pa}$). This support is based heavily on the fact that the value of this special derived unit is close to one atmosphere. It is important, however, to be aware that it is not exactly equal to a standard atmosphere ($101,325.0 \text{ Pa}$) or to the so-called metric atmosphere ($1 \text{ kgf/cm}^2 = 98,066.50 \text{ Pa}$), but is close enough to be confused with both.

As yet, there is no consensus about which of these units should be used as the standard. Accordingly, both are used, often in the same metric country. Because the world cannot agree and because we must all live with the world as it is, the editors concluded that restricting usage to one or the other would be arbitrary, grossly artificial, and not in the best interests of the reader. We therefore have permitted individual authors to use what they are most accustomed to, and both units will be encountered in the text.

Units of pressure are utilized to define both the performance and the mechanical integrity of displacement pumps. For dynamic pumps, however, which are by far the most significant industrial pumps, pressure is used only to describe rated and hydrostatic values, or mechanical integrity. Performance is generally measured in terms of total head, expressed as feet in USCS units and as meters in SI units. This sounds straightforward enough until a definition of head, including consistent units, is attempted. Then we encounter the dilemma of mass versus force, or weight.

The total head developed by a dynamic pump, or the head contained in a vertical column of liquid, is actually a measure of the internal energy added to or contained in the liquid. The units used to define it could be energy per unit volume, or energy per unit mass, or energy per unit weight. If we select the last, we arrive conveniently, in USCS units, as foot-pounds per pound, or simply feet. In SI units, the terms would be newton-meters per newton, or simply meters. In fact, however, metric countries weigh objects in kilograms, not newtons, and so the SI term for head may be defined at places in this volume in terms of kilogram-meters per kilogram, even though this does not conform strictly to SI rules.

Similar ambiguity is observed with the units of flow rate, except here there may be even more variations. The standard SI unit of flow rate is the cubic meter per second, which is indeed a very large value ($1 \text{ m}^3/\text{s} = 15,850.32 \text{ U.S. gal/min} = 15,850.32 \text{ gpm}$) and is therefore really only suitable for very large pumps. Some industry groups have suggested that a suitable alternative might be the liter per second ($1 \text{ l/s} = 10^{-3} \text{ m}^3/\text{s} = 15.85032 \text{ U.S. gal/min}$), while others have maintained strong support for the traditional metric unit of flow rate, the cubic meter per hour ($1 \text{ m}^3/\text{h} = 4.402867 \text{ U.S. gal/min}$). All of these units will be encountered in the text.

These variations have led to several forms of the specific speed, which is the fundamentally dimensionless combination of head, flow rate, and rotative speed that characterizes the geometry of kinetic pumps. These forms are all related to a truly unitless formulation called "universal specific speed," which gives the same numerical value for any consistent system of units. Although not yet widely used, this concept has been appearing in basic texts and other literature, because it applies consistently to all forms of turbomachinery. Equivalencies of the universal specific speed to the common forms of specific speed in use worldwide are therefore provided in this book. This is done with a view to eventual standardization of the currently disparate usage in a world that is experiencing globalization of pump activity.

The value for the unit of horsepower (hp) used throughout this book and in the United States is the equivalent of 550 foot pounds (force) per second, or 0.74569987 kilowatts (kW). The horsepower used herein is approximately 1.014 times greater than the metric horsepower, which is the equivalent of $75 \text{ kgf} \cdot \text{m/s}$ or 0.735499 kW. Whenever the rating of an electric motor is given in this book in horsepower, it is the output rating. The equivalent output power in kilowatts is shown in parentheses.

Variations in SI units have arisen because of differing requirements in various user industry groups. Practices in the usage of units will continue to change, and the reader will have to remain alert to further variations of national and international practices in this area.

PREFACE

As this comprehensive work on pumps takes on a renewed existence in the fourth edition, it is the hope of the present editors that the original purpose of the work is still being served. When the first edition appeared in 1976—and the second edition in 1985—the editors Igor J. Karassik, William C. Krutzsch, Warren H. Fraser, and Joseph P. Messina had two objectives:

- First, to present sufficient information on the theory of design and operation of both rotodynamic (or simply “dynamic”) and positive displacement (both reciprocating and rotary) pumps to assist engineers in designing, analyzing, testing, and troubleshooting all sizes and configurations of these machines.
- Second, to review a representative array of application areas and systems, describing to users, buyers, and operators how pumps are specified, purchased, selected, deployed, started, operated, and maintained to meet the requirements of several environments from water supply, marine, and mining services to chemical plants, petroleum production, electric power generation, aerospace systems, and many others.

The rapid pace of recent industrial and technological developments has made it necessary to update the third edition, which appeared in 2000, in order that the *Pump Handbook* can continue to serve the global pump community in keeping with these two major objectives. The volume of material that could be included to do this is greater than a manageable size; yet it has been found possible to add new material while retaining most of the subject areas, each of which has been treated exclusively in one of the many dedicated sections contained in the chapters of the earlier editions. In this fourth edition, these sections have been regrouped to satisfy present needs, additional chapters having been established for solids pumping, sealing, bearings, and noise. The resulting 16 chapters together with the appendix contain 71 sections, most of which have been updated and some of which are new or are completely new replacements of the earlier sections.

The new sections include Centrifugal Pump Mechanical Behavior and Vibration, including a comprehensive troubleshooting list; CFD Analysis of Flow and Performance,

providing an overview of this increasingly useful analytical tool; Centrifugal Pump Bearings, including a new treatment of rolling element bearings; Water Supply, illustrating current water distribution systems; Cryogenic Pumps for Liquefied Gas Service, detailing the role of pumps in the emerging LNG infrastructure; Pumped Storage, presenting the new machinery and plants in this time-honored energy management area; and Waterhammer, including a new and clear presentation of the transient behavior involved. In regard to transients, these latter two sections, as well as the updated and renamed earlier section, Centrifugal Pumps: Hydraulic Performance and Behavior, present the "complete characteristics," "four-quadrant," or "abnormal" behavior of pumps—both theory and test data—in the context of the particular subject area being addressed.

In updating existing sections from the earlier editions, significant new material has been included in Aircraft Fuel Pumps, which also details the emerging brushless dc electric motor technology for driving these pumps in new airframes; Screw Pumps; Vane, Gear, and Lobe Pumps; Electric Motors and Motor Controls; Permanent Magnet Adjustable Speed Drives; Variable Speed Fluid Drives; Gears; Pump Couplings; Centrifugal Pump Mechanical Seals; Drainage and Irrigation; Metallic Materials and Damage Mechanisms; Pulp and Paper Mills; Heating and Air-Conditioning; Selecting and Purchasing Pumps; and Pump Testing.

Not requiring significant updating are some of the sections that were new for the third edition, including Hydraulic Transport of Solids, an elegant, classical presentation of the flow regimes and losses in slurry pipelines; Application and Construction of Centrifugal Solids Handling Pumps, a companion to the preceding section that clearly presents the slurry pumps used for such transport; and Jet Pump Theory, another classic that treats both single- and two-phase jet pumping.

As the reader will see in the heading of each section, many contributors have prepared or assisted in the preparation of these sections for the *Pump Handbook*, and the editors take this opportunity to thank and honor these experts, who have been willing to share their knowledge and to make the effort required to present it clearly.

As in prior editions, the quantities involved are expressed in both the SI and the U.S. system of units. In each section of the book, either one of these systems is treated as primary, according to the style of the contributor. In all cases, the conversions to the other system are shown, are evident, or are not required in view of global understanding and convention.

In conclusion, the guiding principle of the editors has been to build on the previous editions while at the same time producing a work that is up to date. We recognize that new developments in the world of pumping are going on apace and that more could have been done. Nevertheless we offer this fourth edition of the *Pump Handbook* as a practical tool for the present day, and we hope that readers will benefit from this effort.

PAUL COOPER
CHARLES C. HEALD
JOSEPH P. MESSINA

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**INTRODUCTION:
CLASSIFICATION
AND
SELECTION
OF
PUMPS**

W. C. Krutzsch

Paul Cooper

INTRODUCTION

Only the sail can contend with the pump for the title of the earliest invention for the conversion of natural energy to useful work, and it is doubtful that the sail takes precedence. Because the sail cannot, in any event, be classified as a machine, the pump stands essentially unchallenged as the earliest form of machine for substituting natural energy for human physical effort.

The earliest pumps we know of are variously known, depending on which culture recorded their description, as *Persian wheels*, *waterwheels*, or *norias*. These devices were all undershot waterwheels containing buckets that filled with water when they were submerged in a stream and that automatically emptied into a collecting trough as they were carried to their highest point by the rotating wheel. Similar waterwheels have continued in existence in parts of the Orient even into the present century.

The best-known of the early pumps, the Archimedean screw, also persists into modern times. It is still being manufactured for low-head applications where the liquid is frequently laden with trash or other solids.

Perhaps most interesting, however, is the fact that with all the technological development that has occurred since ancient times, including the transformation from water power through other forms of energy all the way to nuclear fission, the pump remains probably the second most common machine in use, exceeded in numbers only by the electric motor.

Because pumps have existed for so long and are so widely used, it is hardly surprising that they are produced in a seemingly endless variety of sizes and types and are applied to an apparently equally endless variety of services. Although this variety has contributed to an extensive body of periodical literature, it has also tended to preclude the publication of comprehensive works. With the preparation of this handbook, an effort has been made to create just such a comprehensive source.

Even here, however, it has been necessary to impose a limitation on subject matter. It has been necessary to exclude material uniquely pertinent to certain types of auxiliary pumps that lose their identity to the basic machine they serve and where the user controls neither the specification, purchase, nor operation of the pump. Examples of such pumps would be those incorporated into automobiles or domestic appliances. Nevertheless, these pumps do fall within classifications and types covered in the handbook, and basic information on them may therefore be obtained herein after the type of pump has been identified. Only specific details of these highly proprietary applications are omitted.

Such extensive coverage has required the establishment of a systematic method of classifying pumps. Although some rare types may have been overlooked in spite of all precautions, and obsolete types that are no longer of practical importance have been deliberately omitted, principal classifications and subordinate types are covered in the following section.

CLASSIFICATION OF PUMPS

Pumps may be classified on the basis of the applications they serve, the materials from which they are constructed, the liquids they handle, and even their orientation in space. All such classifications, however, are limited in scope and tend to substantially overlap each other. A more basic system of classification, the one used in this handbook, first defines the principle by which energy is added to the fluid, goes on to identify the means by which this principle is implemented, and finally delineates specific geometries commonly employed. This system is therefore related to the pump itself and is unrelated to any consideration external to the pump or even to the materials from which it may be constructed.

Under this system, all pumps may be divided into two major categories: (1) *dynamic*, in which energy is continuously added to increase the fluid velocities within the machine

to values greater than those occurring at the discharge such that subsequent velocity reduction within or beyond the pump produces a pressure increase, and (2) *displacement*, in which energy is periodically added by application of force to one or more movable boundaries of any desired number of enclosed, fluid-containing volumes, resulting in a direct increase in pressure up to the value required to move the fluid through valves or ports into the discharge line.

Dynamic pumps may be further subdivided into several varieties of centrifugal and other special-effect pumps. Figure 1 presents in outline form a summary of the significant classifications and subclassifications within this category.

Displacement pumps are essentially divided into reciprocating and rotary types, depending on the nature of movement of the pressure-producing members. Each of these major classifications may be further subdivided into several specific types of commercial importance, as indicated in Figure 2.

Definitions of the terms employed in Figures 1 and 2, where they are not self-evident, and illustrations and further information on classifications shown are contained in the appropriate sections of this book.

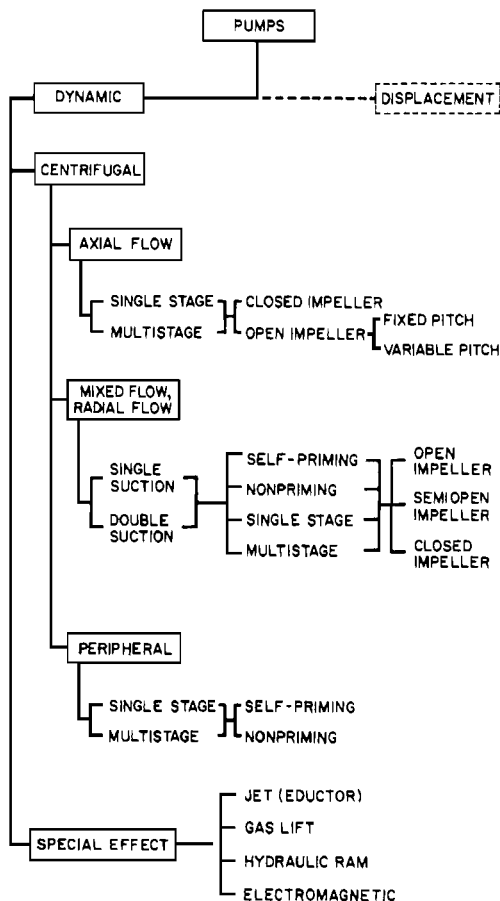


FIGURE 1 Classification of dynamic pumps

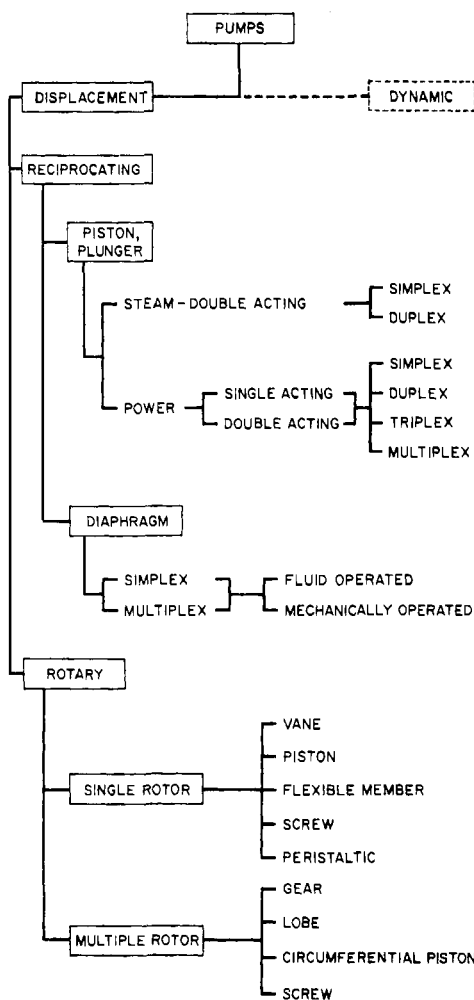


FIGURE 2 Classification of displacement pumps

OPTIMUM GEOMETRY VERSUS SPECIFIC SPEED

Fundamental to any system of classifying pumps is the rotor geometry that is optimum for each type, as illustrated in Figure 3 in terms of the specific speed N_s or Ω_s . Here Q is the volume flow rate or capacity, N is the rotative speed, Ω is the angular speed, and ΔH (or just H) is the pump head—all at the best efficiency point (BEP). Derived in Section 2.1.1 for dynamic pumps, this theoretically dimensionless quantity can also be applied to displacement pumps, at least for selection purposes. For low viscosity, Ω_s emerges as the major influence on rotor geometry. In this case, the pump performance in terms of the head coefficient $\psi = g\Delta H/(\Omega^2 r^2)$ is influenced only by the flow coefficient or “specific flow” $Q_s = Q/(\Omega r^3)$. Now, if one divides $Q_s^{1/2}$ by $\psi^{3/4}$, the rotor radius r ($= D_s/2$) drops out (which is convenient because we don’t usually know it ahead of time), and we get the universal specific speed Ω_s as the major dependent variable—in terms of which the hydraulic design is optimized for maximum efficiency, as shown in Figure 3.

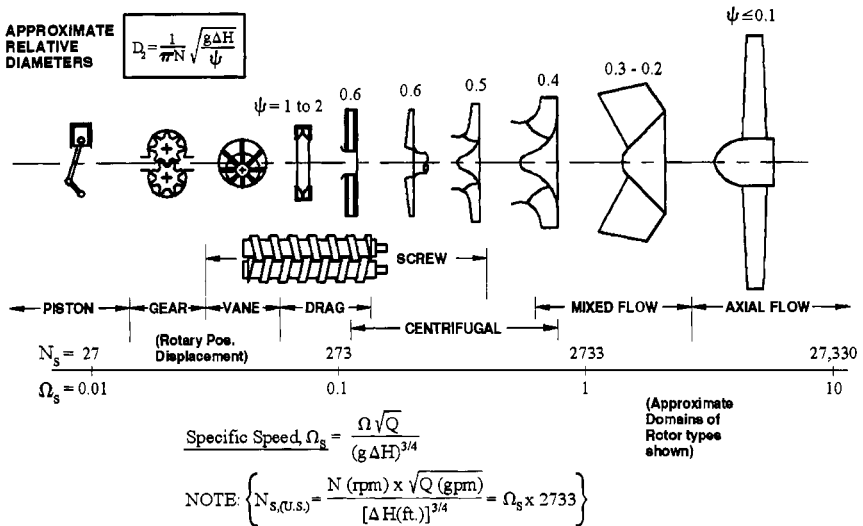


FIGURE 3 Optimum geometry as a function of BEP specific speed (for single stage rotors)

This optimum geometry carries with it an associated unique value of the head coefficient ψ , thereby effectively sizing the rotor. For dynamic, “rotodynamic” or impeller pumps, imagining speed N and head ΔH to be constant over the N_s -range shown yields increasing optimum impeller diameter as shown. This size progression shows that the optimum head coefficient ψ decreases with increasing specific speed.

Outside the N_s range shown in Figure 3 for each type of rotor, the efficiency becomes unsatisfactory in comparison to that achievable with the configuration shown for this N_s . Rotary positive displacement machines such as vane pumps, gear pumps, and a variety of screw pump configurations (all commonly called “rotary pumps”) are more appropriate for the lower values of N_s , the lowest N_s -values requiring reciprocating (piston or plunger) positive displacement pumps.

Single-suction rotors are shown in Figure 3; however, if there are two inlets to the rotor (double suction), the usual practice is that the total discharge flow rate is used for Q in the specific speed expression. An exception to this is found in some places, notably in Europe, wherein the flow rate through one of the two inlets, namely half of the discharge flow rate, is used instead. In terms of the optimum geometry of centrifugal pumps, the latter practice would appear logical for impellers, because a double-suction rotor can be viewed simplistically as two back-to-back single-suction rotors. However in terms of performance (efficiency and head coefficient), the discharge portion of a double-suction impeller and the surrounding volute usually tend to play the dominant role, and these are designed for the discharge flow rate. On the other hand, the design and performance of the inlet regions of the impeller are based on the flow rate through one eye. Nevertheless, unless specifically stated to the contrary, the usage in this book is that the value of Q in the expression for specific speed represents the total discharge flow rate.

Regarding units for these relationships, the rotative speed N is in revolutions per second (rps) unless stated to be in rpm because the quantity $g \Delta H$ usually has the units of length squared per second squared. The diameter D_2 has the same length unit as the head; for example, in the rotor size equation, head in feet would imply diameter in feet. The universal specific speed Ω_s has the same value for any combination of consistent units, and similarly shaped turbine and compressor wheels have similar values of Ω_s —making it truly “universal.” Note that for the unit of time being seconds, Ω is given as radians per second [= $N(\text{rpm}) \times \pi/30$], where radians are unitless.

SELECTION OF PUMPS

Given the variety of pumps that is evident from the foregoing system of classification, it is conceivable that an inexperienced person might well become somewhat bewildered in trying to determine the particular type to use in meeting most effectively the requirements for a given installation. Recognizing this, the editors have incorporated in Chap. 14, "Selecting and Purchasing Pumps," a guide that provides the reader with reasonable familiarity regarding the details that must be established by or on behalf of the user in order to assure an adequate match between system and pump.

Supplementing the information contained in Chap. 14, the sections on centrifugal, rotary, and reciprocating pumps also provide valuable insights into the capabilities and limitations of each of these classes. None of these, however, provide a concise comparison between the various types, except for Figure 3. Moving on from that figure by assuming for these pump types their typical ranges of rotative speed N , liquid density, number of stages, and so on, makes it possible to compare them in terms of the respective ranges of the dimensional quantities of pressure (more precisely, pressure rise) and capacity (or flow rate) to which they are commonly applied. Figure 4 has been included here to do just that.

The lines plotted in Figure 4 for each of the three pump classes represent the upper limits of pressure and capacity currently available commercially throughout the world. At or close to the limits shown, only a few sources may be available, and pumps may well be specially engineered to meet performance requirements. At lower values of pressure and capacity, well within the envelopes of coverage, pumps may be available from dozens of sources as pre-engineered, or standard, products. Note also that reciprocating pumps run off the pressure scale, whereas centrifugals run off the capacity scale. For the former, some highly specialized units are obtainable at least up to $150,000 \text{ lb/in}^2$ ($10,350 \text{ bar}$)¹ and perhaps slightly

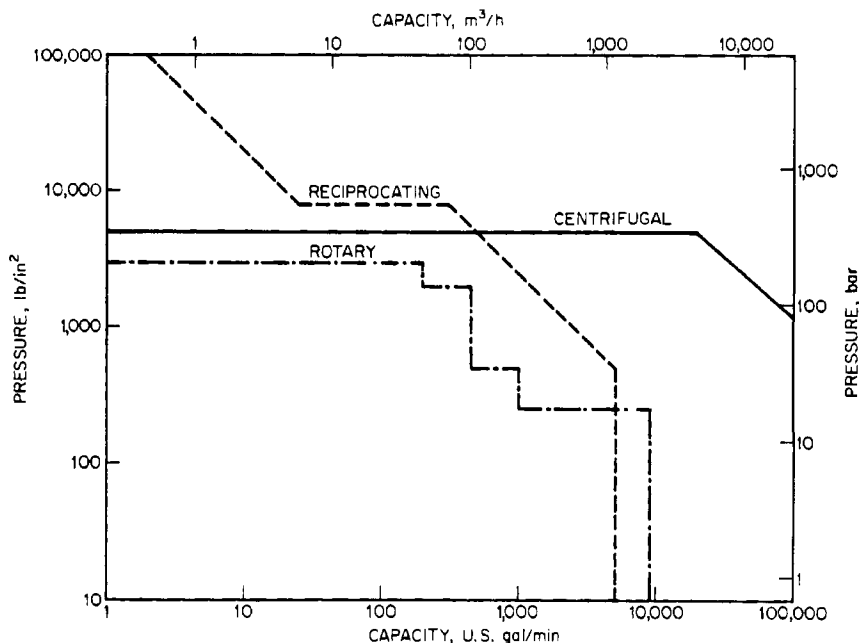


FIGURE 4 Approximate upper limit of pressure and capacity by pump class

¹1 bar = 10^5 Pa. For a discussion of bar, see "SI Units—A Commentary" in the front matter.

higher. For the latter, custom-engineered pumps would probably be available up to about 3,000,000 U.S. gal/min (680,000 m³/h), at least for pressures below 10 lb/in² (0.69 bar).

Given that the liquid can be handled by any of the three basic types and given conditions within the coverage areas of all three, the most economic order of consideration for a given set of conditions would generally be centrifugal, rotary, and reciprocating, in that order. In many cases, however, either the liquid may not be suitable for all three or other considerations—such as self-priming or air-handling capabilities, abrasion resistance, control requirements, or variations in flow—may preclude the use of certain pumps and limit freedom of choice. Nevertheless, it is hoped that the information in Figure 4 will be a useful adjunct to that contained elsewhere in this volume.

CENTRIFUGAL PUMPS

SECTION 2.1

CENTRIFUGAL PUMP THEORY, ANALYSIS, AND PERFORMANCE

2.1.1 CENTRIFUGAL PUMP THEORY

PAUL COOPER

INTRODUCTION

A centrifugal pump is a rotating machine in which flow and pressure are generated dynamically. The inlet is not walled off from the outlet as is the case with positive displacement pumps, whether they are reciprocating or rotary in configuration. Rather, a centrifugal pump delivers useful energy to the fluid or “pumpage” largely through velocity changes that occur as this fluid flows through the impeller and the associated fixed passageways of the pump; that is, it is a “rotodynamic” pump. All impeller pumps are rotodynamic, including those with radial-flow, mixed-flow, and axial-flow impellers: the term “centrifugal pump” tends to encompass all rotodynamic pumps.

Although the actual flow patterns within a centrifugal pump are three-dimensional and unsteady in varying degrees, it is fairly easy, on a one-dimensional, steady-flow basis, to make the connection between the basic energy transfer and performance relationships and the geometry or what is commonly termed the “hydraulic design” (more properly the “fluid dynamical design”) of impellers and stators or stationary passageways of these machines.

In fact, disciplined one-dimensional thinking and analysis enables one to deduce pump operational characteristics (for example, power and head versus flow rate) at both the optimum or design conditions and off-design conditions. This enables the designer and the user to judge whether a pump and the fluid system in which it is installed will operate smoothly or with instabilities. The user should then be able to understand the offerings of a pump manufacturer, and the designer should be able to provide a machine that optimally fits the user's requirements.

The complexities of the flow in a centrifugal pump command attention when the energy level or power input for a given size becomes relatively large. Fluid phenomena such as recirculation, cavitation, and pressure pulsations become important; “hydraulic” and mechanical interactions—involving stress, vibration, rotor dynamics, and the associated design approaches, as well as the materials used—become critical; and operational limits must be understood and respected.

NOMENCLATURE

NOTE: The units for each quantity defined are as stated in this nomenclature, unless otherwise specifically stated in the text, equations, figures, or tables.

A = area, ft^2 (m^2).

a = constant of the diffuser or volute configuration in Pfleiderer’s slip relation.

a = radius of impeller disk, ft (m), $= r_{1,2}$.

A_p = area of flow passage normal to the flow direction, ft^2 (m^2).

b = width of an impeller or other bladed passage in the meridional plane, ft (m).

NOTE: When dealing with radial thrust, b_s includes also the thickness of the shrouds.

C_p = specific heat of liquid being pumped, $\text{Btu}/(\text{lbm}\cdot\text{degF})$; [$\text{kcal}/(\text{kg}\cdot\text{degC})$].

c or V = absolute velocity, ft/sec (m/s).

D = diameter; unless otherwise subscripted = impeller exit diameter, ft (m).

d = diameter, ft (m).

D_h = hydraulic diameter of flow passage $(= 4A_p/\phi)$, ft (m).

F = thrust force, lbf (N).

g = acceleration due to gravity, $= 32.174 \text{ ft}/\text{sec}^2$ ($9.80665 \text{ m}/\text{s}^2$) at earth sea level.

g_o = constant in Newton’s Second Law, $= 32.174 (\text{lbm}\cdot\text{ft})/(\text{lbf}\cdot\text{sec}^2)$. (There is no SI equivalent; use the dimensionless constant 1 in place of g_o in SI computations.)

$\{g_p\}$ = set of fluid properties associated with gas-handling phenomena.

H = head of liquid column, ft (m) (Eq. 3); can also have the same meaning as the change in head ΔH (that is, the same meaning as “pump head”).

ΔH = change in head across pump or pump stage, also called the “pump head” or “total dynamic head” ft (m).

ΔH = the small reduction in pump head (usually 3%) in testing for NPSHR, ft (m).

$H_e = H_{i,\infty}$ = the ideal head for an infinite number of blades that produce no blockage.

H_i = ideal head $[= H + \Sigma(H_L)]$, ft (m) (Eq. 15b); sometimes called the “input head.”

H_L = head loss, ft (m).

ΣH_L = all losses in the main flow passages from pump inlet to pump outlet, ft (m).

h = static enthalpy in Btu/lbm times $g_o J$, ft^2/sec^2 ; (or in kcal/kg times J , $\text{m}^2/\text{s}^2 = J/\text{kg}$).

- h_{su} or NPSH = net positive suction head, ft (m).
- ID = inner diameter.
- J = the mechanical equivalent of heat, 778 ft-lbf/Btu (4184 N-m/kcal).
- ℓ = blade, vane, or passage arc length, ft (m).
- M or T = torque, lbf-ft (N-m).
- m = distance in streamwise direction in meridional plane (Figure 14), in or ft (m).
- $\dot{m}$ = mass flow rate, lbf-sec/ft (kg/s), = ρQ .
- $MCSF$ or $Q_{\min}$ = minimum continuous stable flow, ft³/sec (m³/s).
- N or n = rotative speed of the impeller, rpm.
- NPSH or h_{su} = net positive suction head, ft (m).
- $NPSHA$ or $NPSH_A$ = available NPSH.
- $NPSHR$ or $NPSH_R$ = required NPSH to prevent significant loss (> 3%) of pump Δp or to protect the pump against cavitation damage, whichever is greater.
- $NPSH_{3\%}$ or $NPSH_{3\%}$ = required NPSH to prevent significant loss (> 3%) of pump Δp ; this is the "performance NPSH" defined in Section 2.1.3.
- n_b or Z_i = number of impeller blades.
- n_q = specific speed in rpm, m³/s, m units (Eq. 38b) = $N_s/51.64$ (Eq. 39c).
- n_v or Z_d = number of vanes in diffuser or stator.
- N_s or $N_{s(US)}$ or n_s = specific speed in rpm, gpm, ft units (Eq. 38a).
- N_{ss} or S = suction specific speed in rpm, gpm, ft units (Eq. 42).
- OD = outer diameter.
- P = total pressure, lbf/ft² (Pa).
- p = pressure, lbf/ft² [Pa (=N/m²)] (= "static pressure").
- Δp = pressure rise, lbf/ft² (Pa).
- p_L = pressure loss, lbf/ft² (Pa).
- $p_{L,i}$ = impeller pressure loss from its inlet to the point of interest, lbf/ft² (Pa).
- $p_{L,i} + \nu_L$ = pressure loss $p_{L,i}$ in impeller plus pressure loss in inlet passage, lbf/ft² (Pa).
- Σp_L = all losses in the main flow passages from pump inlet to pump outlet, lbf/ft² (Pa).
- p_v or p_{op} = vapor pressure of liquid being pumped, lbf/ft² (Pa).
- P_I = power delivered to all fluid flowing through the impeller, ft-lbf/sec (kW).
- P_S = shaft power, ft-lbf/sec (kW).
- ϕ = perimeter of flow passage cross section normal to the flow direction, ft (m).
- Q = volume flow rate or, more conveniently, "flow rate" or "capacity," ft³/sec (m³/s).
- Q_{DR} = flow rate below which discharge recirculation exists, ft³/sec (m³/s).
- Q_L = leakage from impeller exit to inlet, ft³/sec (m³/s).
- $Q_{\min}$ or $MCSF$ = minimum continuous stable flow rate, ft³/sec (m³/s).
- Q_R = flow rate below which recirculation exists, ft³/sec (m³/s).

- Q_{SR} = flow rate below which suction recirculation exists, ft³/sec (m³/s).
- Q3D (quasi-3D) = quasi-three dimensional.
- R = radius of curvature of meridional streamline, ft (m) (Figures 13, 14, and 25).
- r = radial distance from axis of rotation, ft (m).
- r_b = radial distance from axis of rotation to center of circle defining impeller passage width, ft (m) (Figures 13 and 25).
- r_e = maximum value of r within the "eye plane," (Figures 13 and 25.)
- s = width of gap between impeller disk and adjacent casing wall, ft (m).
- $S = N_{ss}$, suction specific speed in rpm, gpm, ft units (Eq. 42).
- sp. gr. = specific gravity, namely, the ratio of liquid density to that of water at 60°F (15.6°C).
- { S } = set of flow properties associated with solids in the pumpage
- T = axial thrust, lbf (N).
- T or T or M = torque, lbf-ft (N-m).
- T = temperature, °F or °R (°C or °K).
- ΔT_c = temperature rise due to compression, °F (°C).
- t = time, sec (s).
- t = blade or vane thickness, ft (m).
- u = internal energy in Btu/lbm multiplied by $g_c J$, ft²/sec²; (or in kcal/kg times J , m²/s²).
- U = tangential speed Ωr of the point on the impeller at radius r , ft/sec (m/s).
- U_e = the value of U at the maximum radial location r_e within the "eye plane."
- V = volume, ft³ (m³).
- V or c = absolute velocity of fluid, ft/sec (m/s).
- V_e = the average value of the meridional velocity component V_m in the eye (= Q/A_e), ft/sec (m/s).
- V_s = slip velocity (Figure 15), ft/sec (m/s).
- W or w = velocity of fluid relative to rotating impeller, ft/sec (m/s).
- W_g = the one-dimensional value of W that would exist if there were no slip.
- w_1 = throat width (Figure 21), ft (m).
- y = transformed distance along blade from trailing edge (Figure 19), in or ft (m).
- z = axial distance in polar coordinate system, ft (m).
- Z or Z_e = elevation coordinate, ft (m).
- Z_d or n_v = number of vanes in diffuser or stator.
- Z_i or n_b = number of impeller blades.
- α = angle of the absolute velocity vector from the circumferential direction.
- β = angle of the relative velocity vector or impeller blade in the plane of the velocity diagram (as seen, for example, in Figure 3) from the circumferential (tangential) direction.

- γ = fluid weight density, lb/ft^3 (N/m^3) = ρg . ($1\text{N} = 1 \text{ kg}\cdot\text{m}/\text{s}^2$).
- δ = clearance, ft (m).
- δ^* = displacement thickness of the boundary layer, ft (m).
- δ_0^* = displacement thickness of the zero-pressure-gradient boundary layer, ft (m), ($\approx 0.002 \times \ell$ for turbulent boundary layers at $\nu = 1$ cs in typical centrifugal pumps).
- ε = absolute roughness height, ft (m).
- ε_2 = fraction of impeller discharge meridional area (that is, the area normal to the velocity component $V_{m,2}$) that is *not* blocked by the thickness of the blades and the boundary layer displacement thickness on blades and on hub and shroud surfaces.
- $\varepsilon_{2,b}$ = fraction of the circumference at the exit of the impeller that is *not* blocked by the thickness of the blades and boundary layer displacement thickness on blades. (See computation in Table 10.)
- $\eta = \eta_p$ = pump efficiency; or a component efficiency (different subscript, Eqs. 8–11).
- θ = rotational polar coordinate or central angle about the impeller axis, radians.

NOTE: In a polar-coordinate description of impeller blades or stationary vanes, θ becomes the construction angle and is usually regarded as positive in the direction of the blade development from inlet to exit of the impeller or other blade row.

- μ = slip factor = V_2/U_2 ($= 1 - h_0$, where h_0 is the slip factor as defined by Busemann¹⁸).
- μ = absolute viscosity, $\text{lb}\cdot\text{sec}/\text{ft}^2$ ($\text{Pa}\cdot\text{s}$ or $\text{N}\cdot\text{s}/\text{m}^2$); often quoted in centipoises, abbreviated to “cp” ($1 \text{ cp} = 0.001 \text{ Pa}\cdot\text{s}$). [μ in cp] = sp. gr. $\times$ (ν in cs).]
- ν = kinematic viscosity ($= \mu/\rho$), ft^2/sec (m^2/s); often quoted in centistokes, abbreviated to “cs” ($1 \text{ cs} = 1 \text{ mm}^2/\text{s}$). [ν in cs] = (μ in cp)/sp. gr.]
- ρ = fluid mass density, $\text{lb}\cdot\text{sec}^2/\text{ft}^4$ (kg/m^3), $= \gamma/g$.
- σ = solidity (Eq. 53).
- σ = Thoma’s cavitation parameter $= h_{sv}/H$.
- T** or **T** or **M** = torque, $\text{lb}\cdot\text{ft}$ ($\text{N}\cdot\text{m}$).
- ϕ = flow coefficient.
- $\phi_e = V_e/U_e$ = impeller inlet or eye flow coefficient.
- ϕ_i (or $\phi_{i,2}$) = impeller exit flow coefficient $= V_{m,2}/U_2$ (Figure 12).
- ψ = head coefficient (Figure 12); stream function (Figure 14).
- ψ_i = ideal head coefficient [$= \psi_{i,2} = V_{e,2}/U_2$ for zero inlet swirl ($V_{\theta,1} = 0$)].
- $\psi_{i,2} = V_{e,2}/U_2$ [$= \psi_i$ for zero inlet swirl ($V_{\theta,1} = 0$)].
- Ω = angular speed of the impeller in radians per second ($1/\text{s}$) $= N\pi/30$.
- Ω_s = universal specific speed (unitless) (Eq. 37) $= N_s/2733$ (Eq. 38a) $= n_q/52.92$ (Eq. 38b).
- Ω_{ss} = universal suction specific speed (unitless) (Eq. 41) $= N_{ss}/2733$ (Eq. 42).
- {2-ph} = set of fluid properties associated with vaporization.

Subscripts

- b = impeller blade.
- BEP = best efficiency point.
- D = drag due to disk friction, bearings, and seals.
- DF = disk friction.
- d = discharge flange or exit (ex) of the pump.
- e = at the “eye” of the impeller. The “eye” is the throat (minimum-diameter point) at the entrance into the impeller and is the area defined by the “eye plane,” which is normal to the axis of rotation. “ e ” can refer more specifically to the shroud or maximum-diameter point within the eye, as with r_e (Figure 13) or U_e . The inlet tips of the impeller blades are generally at or near this location.
- ex = exit of diffuser or the discharge flange or port of the pump (d).
- f = the direction of the flow.
- h = hub.
- i = inner limit of region or gap (Tables 4 and 5)
- i (or ideal) = ideal.
- i (or imp) = impeller.
- in (or s) = pump inlet flange or port.
- I/L = inlet passage; that is, the passage from the pump inlet flange or port to the impeller.
- I = input to fluid.
- L = losses.
- m = “mechanical” (pertaining to efficiency, Eq. 9).
- m = component of velocity in the meridional plane (that is, the axial-radial plane containing the axis of rotation).
- mean = the 50% or rms meridional streamline.
- n = normal or BEP value.
- o = outer limit of region or gap (Tables 4 and 5).
- out = pump outlet flange or port.
- p = pressure side of blade or passage.
- R = value of r at the impeller ring clearance.
- S = shaft.
- s (or in) = suction flange or inlet of the pump.
- SE = shockless entry (that is, inlet velocity vector aligned with blade camber line).
- s/o = shut-off or zero flow rate Q .
- r = in the radial direction.
- rms = the 50% or mean meridional streamline.
- s = suction side of blade or passage; shaft.
- s = same meaning as sh and t .
- sh = shroud (also at the eye plane at inlet—and in general “ t ” at outlet).
- stg = stage.
- T = entry throat of volute or diffuser.

- t = the tip or maximum radial position of the impeller blades at inlet or outlet (same meaning as s and sh).
 t = tongue or cutwater.
 u (see θ , below).
 v = volumetric (pertaining to efficiency, Eq. 11).
 v = volute.
 z = in the axial direction.
 θ or u = component of velocity in the circumferential direction (that is, the tangential direction in the polar view that is perpendicular to the axis of rotation).
 1 = impeller inlet at the blade leading edge—at the mean unless further defined.
 2 = impeller outlet at the blade trailing edge—at the mean unless further defined.
 3 = volute base circle or entrance to diffuser.
 ∞ = for an infinite number of blades that also produce zero blockage of the flow.

Superscripts

$-$ = average value

ENERGY TRANSFER

Hydraulics or fluid dynamics has the primary influence on the geometry of a rotodynamic pump stage—of all the engineering disciplines involved in the design of the machine. It is basic to the energy transfer or pumping process. Staging is also influenced by the other disciplines, especially in high-energy pumps. The basic energy transfer relationships need to be thoroughly understood to achieve a credible design and to understand the operation of these machines. Action of the mechanical input shaft power to effect an increase in the of energy of the pumpage is governed by the first law of thermodynamics. Realization of that energy in terms of pump pressure rise or head involves losses and the second law of thermodynamics.

The First Law of Thermodynamics Fluid flow, whether liquid or gas, through a centrifugal pump is essentially adiabatic, heat transfer being negligible in comparison to the other forms of energy involved in the energy transfer process. (Yet, even if the process were not adiabatic, the density of a liquid is only weakly dependent on temperature.) Further, while the delivery of energy to fluid by rotating blades is inherently unsteady (varying pressure from blade to blade as viewed in an absolute reference frame), the flow across the boundaries of a control volume surrounding the pump is essentially steady, and the first law of thermodynamics for the pump can be expressed in the form of the adiabatic steady-flow energy equation (Eq. 1) as follows:

$$P_s = \dot{m} \left[\left(h + \frac{V^2}{2} + gZ_e \right)_{\text{out}} - \left(h + \frac{V^2}{2} + gZ_e \right)_{\text{in}} \right]$$

where

$$h = u + \frac{p}{\rho} \quad (1)$$

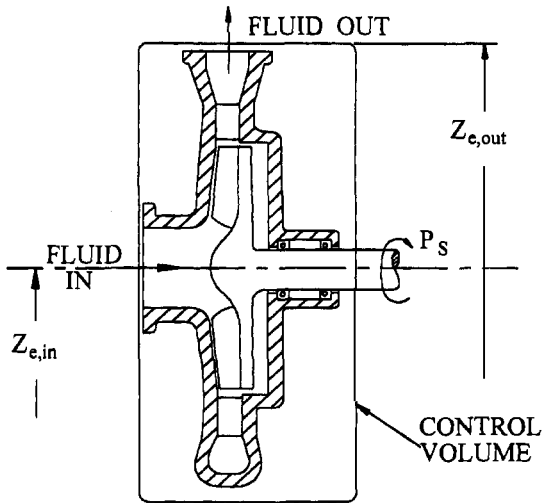


FIGURE 1 Energy transfer in a centrifugal pump

Here, shaft power P_s is transformed into fluid power, which is the mass flow rate $\dot{m}$ times the change in the total enthalpy (which includes static enthalpy, velocity energy per unit mass, and potential energy due to elevation in a gravitational field that produces acceleration at rate g) from inlet to outlet of the control volume (Figure 1).

When dealing with essentially incompressible liquids, the shaft power is commonly expressed in terms of “head” and mass flow rate, as in Eq. 2:

$$\frac{P_s}{\dot{m}} = g\Delta H + \Delta u \quad (2)$$

where

$$H = \frac{p}{\rho g} + \frac{V^2}{2g} + Z_e \quad (3)$$

The change in H is called the “head” ΔH of the pump; and, because H (Eq. 3) includes the *velocity head* $V^2/2g$ and the *elevation head* Z_e at the point of interest, ΔH is often called the “total dynamic head.” ΔH is often abbreviated to simply “ H ” and is the increase in height of a column of liquid that the pump would create if the *static pressure head* $p/\rho g$ and the *velocity head* $V^2/2g$ were converted without loss into elevation head Z_e at their respective locations at the inlet to and outlet from the control volume; that is, both upstream and downstream of the pump.

The Second Law of Thermodynamics: Losses and Efficiency As can be seen from Eq. 2, not all of the mechanical input energy per unit mass (that is, the shaft power per unit of mass flow rate) ends up as useful pump output energy per unit mass $g\Delta H$. Rather, losses produce an internal energy increase Δu (accompanied by a temperature increase) in addition to that due to any heat transfer into the control volume. This fact is due to the second law of thermodynamics and is expressed for pumps in Eq 4:

$$g\Delta H < \frac{P_s}{\dot{m}} \quad \text{or} \quad \eta < 1 \quad (4)$$

where

$$\dot{m} = \rho Q$$

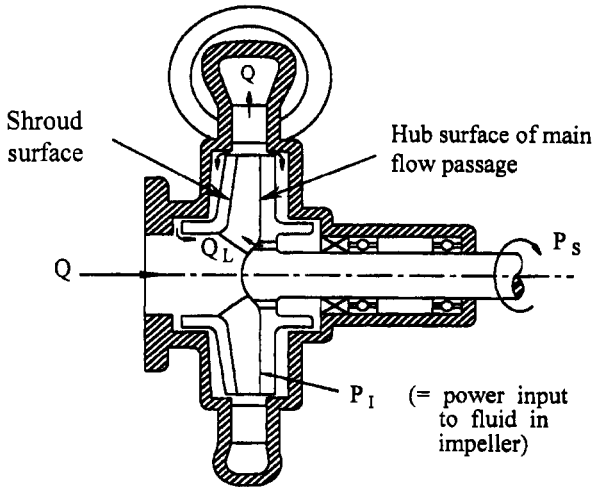


FIGURE 2 Determining component efficiencies. (This is a meridional view.)

The losses in the pump are quantified by the overall efficiency η , which must be less than unity and is expressed in Eq. 5:

$$\eta = \frac{g\Delta H\dot{m}}{P_s} = \text{Overall Pump Efficiency} \quad (5)$$

It should be pointed out here that real liquids undergo some compression—which is accompanied by a reversible increase in the temperature ΔT_c of the liquid—called the “heat of compression.” This portion of the actual total temperature rise ΔT is in addition to that arising from losses and must therefore be taken into account when determining efficiency from measurements of the temperature rise of the pumpage.¹ See the discussion on this subject in Section 2.1.3.

To pinpoint the losses, it is convenient to deal with them in terms of “component efficiencies.” For the typical shrouded- or closed-impeller pump shown in Figure 2, Eq. 5 can be rewritten as follows:

$$\eta = \frac{P_I}{P_s} \times \frac{g\Delta H(\dot{m} + \dot{m}_L)}{P_I} \times \frac{\dot{m}}{\dot{m} + \dot{m}_L} \quad (6)$$

Noting that

$$P_I = g\Delta H_i(\dot{m} + \dot{m}_L) \quad (7)$$

and

$$\dot{m} = \rho Q$$

$$H_i = \text{Ideal Head}$$

one may rewrite Eq. 6 as follows:

$$\eta = \frac{P_I}{P_s} \times \frac{\Delta H}{\Delta H_i} \times \frac{Q}{Q + Q_L} = \eta_m \times \eta_{HY} \times \eta_v \quad (8)$$

where

$$\left. \begin{array}{l} \text{"Mechanical"} \\ \text{Efficiency} \end{array} \right\} \eta_m = \frac{P_I}{P_S} = \frac{P_S - P_D}{P_S} \quad (9)$$

$$\left. \begin{array}{l} \text{Hydraulic} \\ \text{Efficiency} \end{array} \right\} \eta_{HY} = \frac{\Delta H}{\Delta H_i} = \frac{\Delta H_i - \Sigma H_L}{\Delta H_i} \quad (10)$$

$$\left. \begin{array}{l} \text{Volumetric} \\ \text{Efficiency} \end{array} \right\} \eta_v = \frac{Q}{Q + Q_L} \quad (11)$$

Approximate formulas for the three component efficiencies of Eq. 8 will be given further on. Their product yields the overall pump efficiency as defined in Eq. 5, and reflects the following division of the pump losses:

- a. External drags on the rotating element due to i) bearings, ii) seals, and iii) fluid friction on the outside surfaces of the impeller shrouds—called “disk friction”; the total being $P_D = P_S - P_I$. Generally, the major component of P_D is the disk friction, and the “mechanical efficiency” is that portion of the shaft power that is delivered to the fluid flowing through the impeller passages.
- b. Hydraulic losses in the main flow passages of the pump; namely, inlet branch, impeller, diffuser or volute, return passages in multistage pumps, and outlet branch. The energy loss per unit mass is $g\Sigma H_L = g(H_i - \Delta H)$, the ratio of output head ΔH to the input head H_i being the hydraulic efficiency. This is the major focus of the designer for typical centrifugal pump geometries (which are associated with normal “specific speeds”—to be defined later). The other two component efficiencies are then quite high and of relatively little consequence.
- c. External leakages totaling Q_L leaking past the impeller and back into the inlet eye. This leakage has received its share of the full amount of power $P_I = \rho g \Delta H_i (Q + Q_L)$ delivered to all the fluid $(Q + Q_L)$ passing through the impeller. This leakage power is $P_L = \rho g \Delta H_i Q_L$, which is lost as this fluid leaks back to the impeller inlet. The remaining fluid input power is thus $(P_I - P_L) = \rho g \Delta H_i Q$, the ratio of this power to the total (P_I) being the volumetric efficiency.

There are exceptions to this convenient model for dividing up pump losses. The main exception is that if the pump has an open impeller, that is, one without either or both shrouds, that portion of the total leakage Q_L disappears. The leakage now occurs across the blade tips and affects the main flow passage hydraulic losses. The volumetric efficiency is now higher, but the hydraulic efficiency is lower. In that case disk friction is still present, as the impeller still has to drag fluid along the adjacent stationary wall(s). Another exception—for closed impellers—is that disk friction is fundamentally an inefficient pumping action, the fluid being flung radially outward²; and this can result in a slight increase in pump head if the fluid on the outside of an impeller shroud or disk is pumped into the main flow downstream of the impeller.

VELOCITY DIAGRAMS AND HEAD GENERATION

The mechanism of the transfer of shaft torque (or power) to the fluid flowing within the impeller is fundamentally dynamic; that is, it is connected with changes in fluid velocity. This requires the introduction of Newton's second law, which when combined with the first law of thermodynamics, yields Euler's Pump Equation. Fluid velocities at inlet and exit of the impeller are fundamental to this development. Fluid flowing along the blades of an impeller rotating at angular velocity Ω and viewed in the rotating reference frame of that

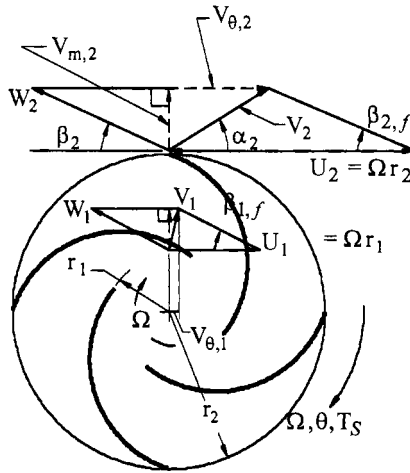


FIGURE 3 Impeller velocity diagrams (1 = inlet; 2 = outlet)

impeller has relative velocity W . Vectorially adding W to impeller blade speed $U = \Omega r$ yields the absolute velocity V , as shown in the velocity diagrams of Figure 3.

Newton's Second Law for Moments of Forces and Euler's Pump Equation Relating impeller torque T to fluid angular momentum per unit mass rV_θ is the convenient way of applying Newton's second law to centrifugal pumps. This is stated as follows for the control volume V that contains the pump impeller:

$$\Sigma T = \int_V [\partial(\rho r V_\theta) / \partial t] dV + \int_S \rho r V_\theta dQ \quad (12)$$

where $\Sigma T = T_S - T_D$ is the summation of torques acting on the impeller; namely, the net torque T_I acting on the fluid flowing through it. The volume integral (first term on the right side) of Eq. 12 is the unsteady term, which is zero for steady operation. It comes into play during changing or transient conditions, such as start-up and shutdown; that is, when the angular momentum per unit volume $\rho r V_\theta$ is changing with time within the impeller control volume V .

The surface integral (second term on the right hand side) of Eq. 12 is the one that pump designers and users are mainly concerned with. Its integration over the exterior surface S of the control volume V is effectively accomplished for most impellers by combining one-dimensional results from inlet to outlet on each of several stream surfaces—imagined to be nested surfaces of revolution bounded by the hub and shroud stream surfaces (indicated in Figure 2). Insight into the significance of this term can be gained by taking the mean value of the integrand in terms of the velocities on a representative stream surface; that is, essentially the surface of revolution lying at an appropriate mean location between hub and shroud. Each of the two velocity diagrams of Figure 3 lies in a plane tangent to this mean stream surface. For flow through an impeller, the torque delivered to the fluid is therefore given by the following relationship involving these average quantities:

$$T_I = (\dot{m} + \dot{m}_L) \times (r_2 V_{\theta,2} - r_1 V_{\theta,1}) \quad (13)$$

or $\times \Omega$:

$$P_I = (\dot{m} + \dot{m}_L) \times (U_2 V_{\theta,2} - U_1 V_{\theta,1}) \quad (14)$$

Eq. 13 says that the torque is equal to the mass flow rate times the change of angular momentum per unit mass $\Delta(rV_\theta)$. This becomes the “power” statement of Eq. 14 when both sides are multiplied by Ω . Following the statement of the second law of thermodynamics in Eq. 4, we now can similarly say that $g\Delta H$ must be less than the power input to the fluid per unit of mass flow rate, namely $\Delta(UV_\theta)$ from Eq. 14. So, we now arrive at Euler’s Pump Equation—expressed three different ways as follows:

$$g\Delta H < \Delta(UV_\theta) \quad (15a)$$

$$g\Delta H_i = \Delta(UV_\theta) \quad (15b)$$

or

$$g\Delta H = \eta_{HY}\Delta(UV_\theta) \quad (15c)$$

The inequality (Eq. 15a) is quantified by Eq. 15b, which follows in view of Eq. 7. Eq. 15c then follows from the definition of hydraulic efficiency (Eq. 10). Euler’s Pump Equation makes one of the most profound statements in the field of engineering, because it determines the major geometrical features of the design of a rotodynamic machine. By reversing the inequality in Eq. 15a, the same principle applies to turbines; hence, the more encompassing title, “Euler’s Pump and Turbine Equation.”

So, to design or analyze a pump, one needs to a) obtain the velocity diagrams that will produce the ideal head at the design flow rate and b) determine how the shape of these diagrams affects the hydraulic efficiency η_{HY} , so as to obtain the desired pump stage head. Step (a) for a given pump is a simple one-dimensional exercise that utilizes the principles of continuity and kinematics (Eqs. 16 and 17) to construct the velocity diagrams for a given total impeller volume flow rate Q and pump rotative speed (Ω or N):

$$\text{Continuity:} \quad Q = 2\pi r b V_m \quad (16)$$

where $W = V_m / \sin \beta_f$

$$\text{Kinematics:} \quad V_\theta = U - W \cos \beta_f \quad (17)$$

Step (b) is in essence the evaluation of the hydraulic losses ΣH_L in Eq. 10, which depend mainly on the relative and absolute velocities, the associated flow passage dimensions, and incidence angles. Eq. 15c then gives the head that the pump stage will generate. Performing steps (a) and (b) at several other flow rates at the same speed enables one to develop the pump performance characteristics.

STATIC PRESSURE GENERATION

The Extended Bernoulli Equation To estimate the losses, it is convenient first to investigate the static pressure and velocity head portions of the total head. Eq. 15c can be written in terms of the total pressure P , which equals $\rho gH - \rho gZ_e$. Similarly, we may speak of hydraulic losses as losses of static pressure Σp_L , which equals $\rho g\Sigma H_L$; so

$$P = P_{in} + \rho(UV_\theta - U_1V_{\theta,1}) - \Sigma p_L - \Delta(\rho gZ_e) \quad (18)$$

where, from Eq. 3, the static (pressure) and dynamic (velocity) components of the total pressure are brought into evidence:

$$P = p + \frac{1}{2}\rho V^2 = \rho gH - \rho gZ_e \quad (19)$$

zero. (This would be a more recognizable form of the extended Bernoulli equation, which differs from the classical Bernoulli equation only by the presence of the loss term.³⁾ A further static pressure increase occurs in the stationary collection system downstream of the impeller (with the attendant losses); namely, in the stationary volute or diffuser. This static rise is generally about a third of that in the impeller, and it is due only to diffusion, that is, the decrease in velocity in that passage. To complete the picture, there is often an increase in the comparatively small velocity in the approach passageway or nozzle or suction branch from the pump inlet port or flange to the impeller eye or blade leading edge. This is accompanied by an attendant small pressure drop.

Internal Static Pressure Distribution If the fluid enters the pump from a stagnant pool, the total pressure at the impeller eye P_1 will be very nearly the static pressure of the upstream pool (plus the pressure equivalent of the elevation of the pool above the eye). This is one reason why the local static pressure p within the pump is often referenced to P_1 , as indicated in the following form of Eq. 21:

$$p - P_1 = \frac{1}{2} \rho(U^2 - W^2) - p_{L,i} - \rho g(Z_e - Z_{e,1}) - \rho U_1 V_{\theta,1} \quad (22)$$

where

$$P = p_1 + \rho \frac{V^2}{2} \quad (23)$$

Figure 5 is an illustration of the internal static pressure development. The difference between P_1 and p_1 is due to the impeller inlet absolute velocity head or dynamic pressure $\rho V_1^2/2$, a much larger difference existing at the impeller exit, namely $\rho V_2^2/2$. Losses in the collector result in the pump or stage outlet total pressure P_{out} being less than P_2 , the rise in total pressure ΔP_{pump} from port to port being $P_{\text{out}} - P_{\text{in}}$. In the figure, P_{in} is very nearly the same as P_1 , the inlet passage loss being comparatively small for most pumps.

NET POSITIVE SUCTION HEAD

Local reduction of the static pressure p to the vapor pressure p_v of the liquid causes vaporization of the liquid and cavitation. Internal pressure drops are due to a) impeller inlet velocity head and inlet passage loss and b) blade loading and loss within the impeller. In order to prevent a substantial decrease of impeller pressure rise, the sum of these pressure drops should not exceed the difference between P_{in} and p_v , the head equivalent of which is called "net positive suction head" or NPSH:

$$\frac{P_{\text{in}} - p_v}{\rho g} = \text{NPSH} \Rightarrow P_{\text{in}} = \rho g \text{NPSH} + p_v \quad (24)$$

Insufficient NPSH leads to cavitation and loss of pump pressure rise. That is because the impeller can become filled with vapor, in which case the density ρ of the fluid within the impeller is then reduced by orders of magnitude. This in turn, as can be seen in Eqs. 18–22, results in essentially zero pump pressure rise; that is, total loss of pump performance.

Eq. 24 substituted into Eq. 22 yields the local static pressure above vapor pressure in terms of the NPSH:

$$p - p_v = \rho g \text{NPSH} + \rho(U^2 - W^2) - p_{L,i+1/L} - \rho g(Z_e - Z_{e,1}) - \rho U_1 V_{\theta,1} \quad (25)$$

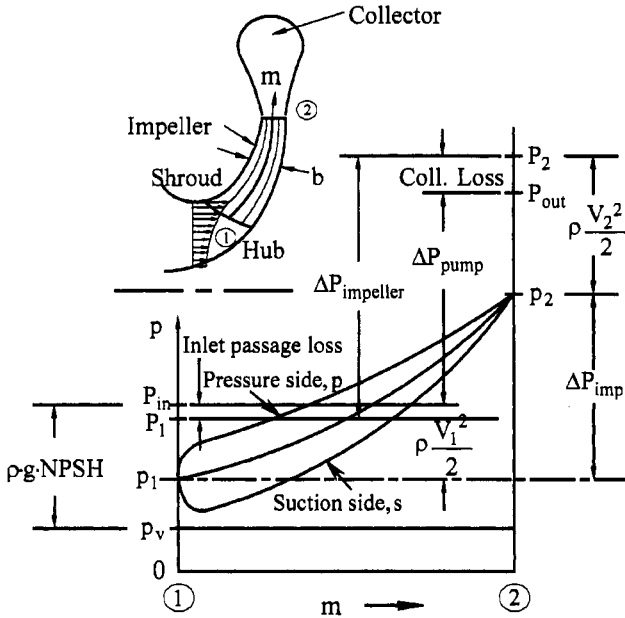
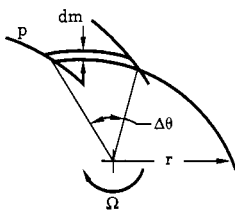


FIGURE 5 Pump stage internal pressure development. Total pressure rise $\Delta P = \rho g \Delta H$

This, together with the foregoing pressure drops, which occur in the inlet region of the pump, is illustrated in Figure 5. The figure contains three plots of p along the representative streamline from 1 to 2, m being distance along this line in the meridional plane. These plots are for the suction side or trailing face of an impeller blade, the pressure side or driving face, and the average or mid-passage position. The middle or average pressure plot is readily described by Eq. 25 in terms of the local average W -distribution. The local blade-to-blade static pressure difference $p_p - p_s$ arises from the torque exerted on a strip of fluid between the blades and approximated here via blade-to-blade average velocity components in Newton's Second Law for Moments of Forces:



$$r(p_p - p_s)b dm = \rho V_m b r \cdot \Delta\theta \cdot d(rV_\theta) \quad (26)$$

$$p_p - p_s = \frac{2\pi}{n_b} \frac{\rho V_m}{\Omega} \frac{d(UV_\theta)}{dm}$$

where, for ease of illustration, the blade-to-blade polar angle difference $\Delta\theta$ is taken equal to $2\pi/n_b$, the actual value of $\Delta\theta$ being slightly less than this due to the thickness of the blades. Thus, for example, too small a number of blades n_b results in a larger value of $p_p - p_s$ and a lower minimum static pressure in the inlet region of the impeller.

CFD codes have been developed that simulate the density reduction in a cavitating impeller and can accurately predict the resulting effect on pump performance.⁴ More rapid and convenient approximations to CFD predictions are available to engineers and

designers in the form of empirical relationships for acceptable levels of NPSH. These will be presented further on, as guidelines for design and performance prediction are developed.

PERFORMANCE CHARACTERISTIC CURVES

Velocity diagrams and ideal head-rise vary with flow rate Q as illustrated in Figure 6 for the typical case of constant rotative speed N or angular speed Ω . Flow patterns in Figure 6b correspond to points on the characteristic curves of Figure 6a. The inlet velocity diagrams (just upstream of the impeller) are shown there for high and low flow rate—with zero swirl being delivered by the inlet passageway to the impeller; that is, $V_{\theta,1} = 0$. The outlet velocity diagrams on Figure 6a are found one-dimensionally, the magnitude of the exit relative velocity vector W_2 varying directly with Q and its direction being nearly tangent to the impeller blade. From these diagrams are found the absolute velocity vector V_2 and its circumferential component $V_{\theta,2}$. Because blade speed U_2 is constant, the resulting plot of the ideal head $\Delta H_i = U_2 V_{\theta,2}/g$ (from Eq. 15b) is a straight line, rising to the point U_2^2/g at zero- Q or “shut-off head.” This is twice the impeller OD tip speed head $U_2^2/2g$. The right-most velocity diagram in Figure 6a has zero $V_{\theta,2}$; however, the maximum or “runout” flow rate happens at lower Q than this. That is because the actual head H is less than ΔH_i due to losses (as seen in Eq. 10), and $\Delta H = 0$ at runout—where overall pump efficiency (Eq. 8) is also zero.

This one-dimensional analysis works well in the vicinity of the best efficiency point (b.e.p. or BEP) and at higher Q because the fluid flows smoothly through the impeller passages as illustrated in Figure 6b for “high Q .” However, it fails at “low Q ,” where recirculating flow develops—indicated by a substantial one-dimensional deceleration or reduction in the fluid velocity relative to those passages—that is, $W_2 \ll W_1$. This is analogous to a diffuser with side walls that diverge too much: the main fluid stream separates from one or both walls and flows along in a narrow portion of the passage in a jet—the rest of the passage being occupied with eddying fluid that can recirculate out of the impeller inlet and exit. Consequently, the real outlet velocity diagram at low Q is the one with the dashed lines and the smaller value of $V_{\theta,2}$, rather than the solid-lined, one-dimensional diagram superimposed on it. This in turn reduces the ideal head at the low- Q point of the curves.

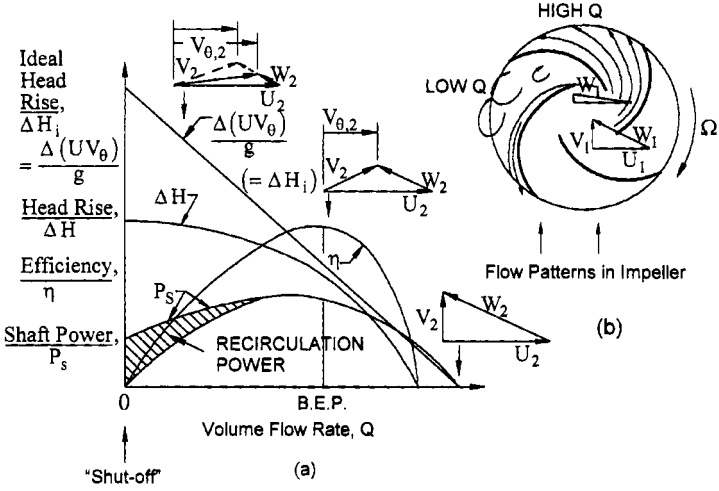


FIGURE 6A and B Characteristic performance curves of a pump stage, related to velocity diagrams

To complicate matters further at low Q , one-dimensional application of this "corrected" outlet velocity diagram via Eq. 14 would produce a pump power consumption curve that passes through the origin of Figure 6a. Such a result (assuming negligible external drag power P_D), is known not to occur in a real pump. Rather, superimposed on the jet flow pattern just described is recirculating fluid that leaves the impeller, gives up its angular momentum to its surroundings, and re-enters the impeller to be re-energized.

In other words, the one-dimensional simplifications mentioned after Eq. 12 do not hold at low Q ; rather, there is an added "recirculation power," which is the UV_ϕ -change experienced by the recirculating fluid integrated over each element of re-entering mass flow rate⁵. The complexity of this recirculation destroys one's ability to interpret pump performance under such conditions by means of velocity diagrams. Instead, a transition is made from empirical correlations for head and power at "shutoff" or zero net flow rate to the high- Q , one-dimensional analysis, enabling one to arrive at the complete set of characteristic curves for efficiency, power, and head illustrated in Figure 6a. In fact, impeller pressure-rise at shutoff is very nearly what would be expected due to the centrifugal effect of the fluid rotating as a solid body, namely $\rho U_\phi^2/2$. The recirculating flow patterns seem to be merely superimposed with little effect on impeller pressure-rise. This recirculation, on the other hand, does produce some additional shutoff pressure rise in the collecting and diffusing passages downstream of the impeller.

SCALING AND SIMILITUDE

When a set of characteristic curves for a given pump stage is known, that machine can be used as a model to satisfy similar conditions of service at higher speed and a different size. Scaling a given geometry to a new size means multiplying every linear dimension of the model by the scale factor, including all clearances and surface roughness elements. The performance of the model is then scaled to correspond to the scaled-up model by requiring similar velocity diagrams (often called "velocity triangles") and assuming that the influences of fluid viscosity and vaporization are negligible. The proportions associated with Eqs. 27, 29, and 32 illustrate this. The blade velocity U (Eq. 30) varies directly with rotative speed N or angular speed Ω —and directly with size, as expressed by the radius r . For the velocity V (or W) to be in proportion to U , the flow rate Q must therefore vary as Ωr^3 ; hence, the "specific flow" Q_s must be constant (Eq. 28). Further, as the head is the product of two velocities, it must vary as $\Omega^2 r^2$; hence, the head coefficient ψ must be constant (Eq. 31). Finally, as power is the product of pressure-rise and flow rate, shaft power P_s must vary as $\rho \Omega^3 r^5$; hence, the power coefficient must be constant.

$$Q = AV \quad \begin{cases} A \propto r_2^2 & \propto D^2 \\ V \propto \Omega r_2 & \propto ND \end{cases}$$

$$\Rightarrow Q \propto ND^3 \quad \text{or} \quad \Omega r_2^3 \quad (27)$$

$$\text{and} \quad \frac{Q}{\Omega r_2^3} = \text{Constant} = Q_s \quad (28)$$

At above Q , $\eta = \text{Constant}$, and

$$g\Delta H = \eta_{HY}\Delta(UV_\phi)$$

$$\Rightarrow \Delta H \propto N^2 D^2 \quad (29)$$

$$U = \Omega r \quad (30)$$

and

$$\psi = \frac{g \Delta H}{U_2^2} = \text{constant} \quad (31)$$

$$P_s = \frac{\rho Q g \Delta H}{\eta}$$

$$\Rightarrow P_s \propto \rho N^3 D^5 \quad (32)$$

and

$$\hat{P}_s = \frac{P_s}{\rho \Omega^3 r_2^5} = \text{constant} \quad (33)$$

Figure 7a is the result of following these similarity rules for a given pump that undergoes a change in speed from full speed to half speed without a change in size. The similar Q at half speed for a given Q at full speed is half that at full speed. At each such half-speed Q -value, the head ΔH is accordingly one-fourth of its full-speed value and the efficiency η is unchanged. One can avoid replotting the characteristic curves in this manner for every change in speed (and size) by expressing them nondimensionally in terms of Q_s , ψ , η , and $\hat{P}_s$. They then all collapse on one another as illustrated in Figure 7b. Note that a change in pump geometry or shape of the hydraulic passageways destroys this similitude and necessarily produces a new set of curves—shaped differently but similar to each other.

Similitude enables the engineer to work from a single dimensionless set of performance curves for a given pump model. This is a practical but special case of the more

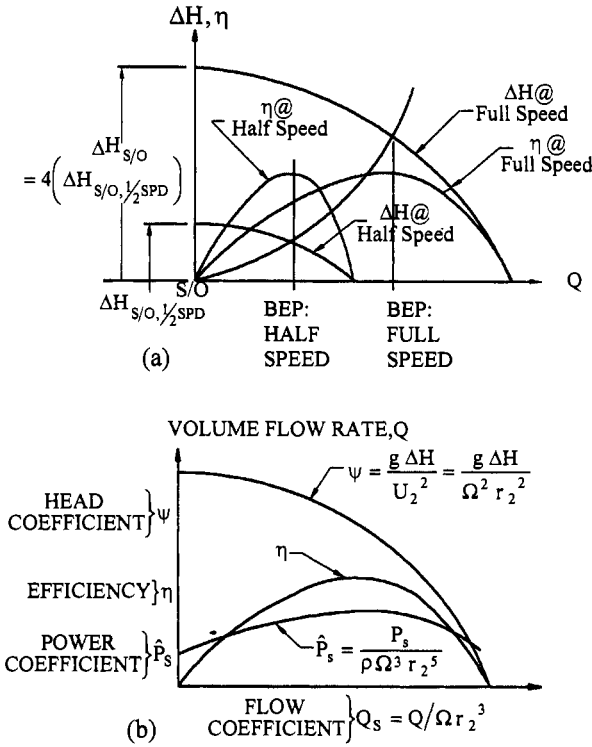


FIGURE 7 Similar performance curves: a) dimensional; b) dimensionless

general statement that pump performance as represented by efficiency, head, and power, is more generally expressed in terms of the complete physical equation as follows:

$$\eta, \Delta H, P_S = \text{fct's.} (Q, r_2, \Omega, \rho, \nu, NPSH, \{2-ph\}, \{g_p\}, \{S\}, \{\ell_i\}) \quad (34)$$

where $\{\ell_i\}$ is the infinite set of lengths that defines the pump stage geometry. A common group of these lengths is illustrated in Figure 8. Dimensionlessly, Eq. 34 becomes

$$\eta, \psi, \hat{P}_S = \text{fct's.} (Q_s, R_e, \tau_2, \{2-\Phi\}, \{\Gamma_p\}, \{\Sigma\}, \{G_i\}) \quad (35)$$

where the dimensionless quantities containing flow rate, viscosity and NPSH are respectively defined as follows:

where	$Q_s = \frac{Q}{\Omega r_2^3}$	$R_e = \frac{\Omega r_2^2}{\nu}$	$\tau_2 = \frac{2gNPSH}{\Omega^2 r_2^2}$
	<u>Specific Flow</u>	<u>Machine Reynolds Number</u>	<u>Cavitation Coefficient</u>

(36)

and

- $\{G_i\} = \{\ell_i/r\}$ defines the dimensionless geometry or shape.
- $\{2-\Phi\}$ = the dimensionless quantities arising from the set of fluid, thermal, vaporization, and heat transfer properties $\{2-ph\}$ that influence the flow of two-phase vapor and liquid. These quantities come into play when the NPSH is low enough for such flow to be extensive enough to influence pump performance.
- $\{\Sigma\}$ = the dimensionless quantities arising from the set of properties $\{S\}$ associated with entrained solids and emulsifying fluids that affect the performance of slurry pumps and emulsion pumps. Similarly, $\{\Gamma_p\}$ arises from $\{g_p\}$ for entrained gases.

SPECIFIC SPEED AND OPTIMUM GEOMETRY

The hydraulic geometry or shape of a pump stage can in principle be chosen for given values of the other independent variables in Eqs. 34 or 35 so as to optimize the resulting performance; for example, to maximize the best efficiency η_{BEP} under certain conditions on the head and power. Two such conditions that are common are a) no positive slope is allowed anywhere along the ΔH -vs.- Q curve of Figure 7 (called the "no drooping nor dip" condition) and b) the maximum power consumption must occur at the BEP (often called the "non-overloading" condition). A fundamental and generally typical pumping situation involves a) negligible influence of viscosity, (that is, high Reynolds number) b) the absence of two-phase fluid effects, (that is, the existence of sufficient NPSH or τ) and c) the absence of entrained gases, solid particles and emulsion-related substances in the fluid. In this situation, Eq. 35 has one remaining significant independent variable; namely, the specific flow Q_s , which in the definitions of Eq. 36 contains the volume flow rate Q , the pump speed Ω , and the characteristic radius r_2 . Most users don't know the size of the pump stage a priori; so, r_2 is eliminated by replacing Q_s in Eq. 35 with a new quantity that is the result of dividing the square root of Q_s by the $\frac{3}{4}$ -power of the head coefficient ψ . Thus, from the definitions just given of Q_s and ψ , one arrives at the *specific speed* Ω_s as the independent variable in terms of which the geometry is optimized³:

$$\Omega_s = \frac{\Omega \sqrt{Q}}{(g\Delta H)^{3/4}} = \frac{\sqrt{Q_s}}{\psi^{3/4}} \quad (37)$$

For convenience, specific speed is usually expressed in terms of the conventional quantities N , Q , and ΔH that correspond to the factors in Eq. 37, which quantities are expressed

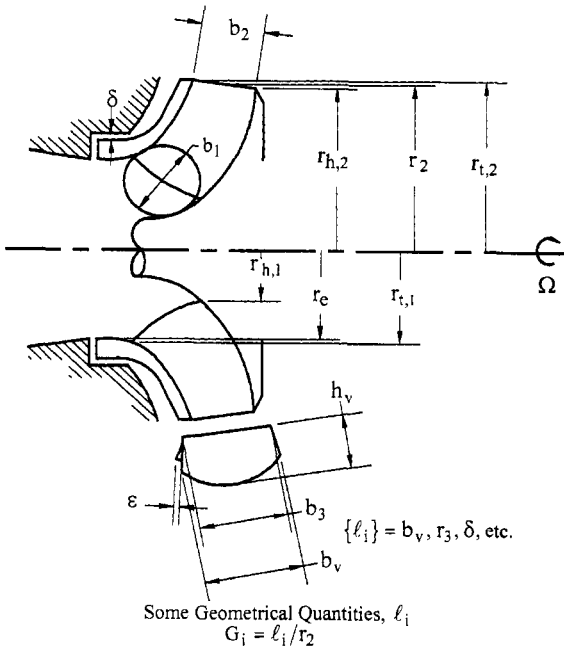


FIGURE 8 Defining the geometry of a pump stage

in the units commonly used commercially. For example, forms found in the United States and in Europe and the relationship of these to the truly unitless “universal specific speed” Ω_s defined in Eq. 37 are as follows:

$$\Omega_s = \frac{N(\text{rpm}) \sqrt{Q(\text{USgpm})/[\Delta H(\text{ft})]^{3/4}}}{2733.016} = \frac{N_{s,(\text{U.S.})}}{2733.016} \quad (38a)$$

$$\Omega_s = \frac{N(\text{rpm}) \sqrt{Q(\text{m}^3/\text{s})/[\Delta H(\text{m})]^{3/4}}}{52.919} = \frac{n_q}{52.919} \quad (38b)$$

Thus,

$$n_q(\text{m}^3/\text{s}, \text{m}) = N_s(\text{USgpm}, \text{ft})/51.64 \quad (38c)$$

Rotor Shape as a Function of Specific Speed Optimization of pump hydraulic geometry in terms of the BEP specific speed has taken place empirically and analytically throughout the history of pump development. An approximate illustration of the results of this process for pump rotors or impellers is shown in Figure 9. Not only does the geometry emerge from the optimization process but also the head, flow, and power coefficients for each shape as well. Approximate values for the optimum BEP head coefficient ψ are shown on the figure. The actual rotor diameter can then be deduced as noted—from the ψ -definition of Eq. 31. The relationship among the various rotors is illustrated in the figure—assuming that they all have the same speed and head; that is, as one moves along

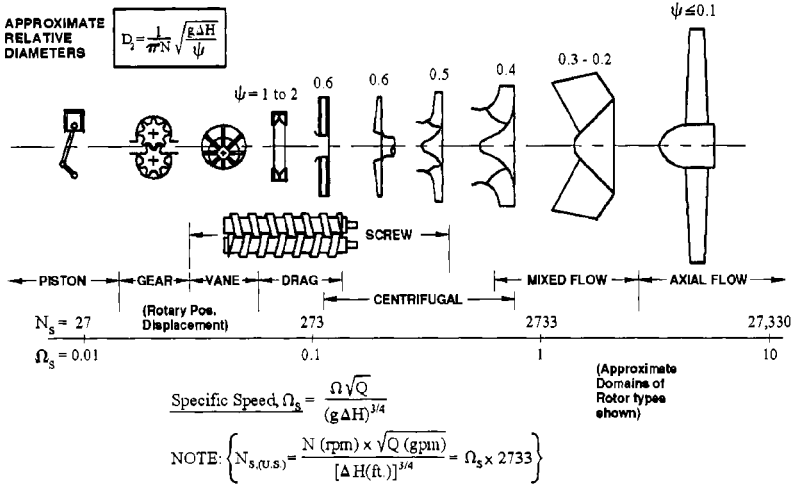


FIGURE 9 Optimum geometry as a function of BEP specific speed (for single-stage rotors)

the abscissa or specific speed axis of Figure 9, only the flow rate is changing as far as the illustrations of the rotors are concerned. As would be expected, therefore, high-specific-speed impellers need to have large passages relative to their overall diameter. Conversely, the passages become smaller as the centrifugal range is traversed toward lower specific speed, the low limit effectively being $\Omega_s = 0.2$ ($N_s \approx 500$), at which point an open impeller of the "Barske" type is illustrated as being typical for this part of the range. Yet centrifugal impellers are found at values of Ω_s as small as 0.1, at which point the impeller is sometimes simply a disk with drilled holes for passages, as illustrated.

For simplicity, the common "double-suction" arrangement, wherein two centrifugal impellers are effectively configured back-to-back, is not shown in Figure 9. Double-suction impellers commonly discharge into a single, surrounding collector designed to accommodate twice the flow rate of one side. Because this collector can be fed by an equivalent single-suction impeller, the specific speed of the whole pump stage is computed using the combined flow rate of the two sides of the double-suction impeller. While this is common practice in the United States, the practice in some other parts of the world is to compute the specific speed of such a pump stage using the flow rate of only one side of the impeller. The former approach is convenient for associating the exit geometrical features and flow behavior of the stage with the magnitude of the specific speed, while the latter favors the suction or inlet characteristics.

At the lower end of the specific speed range shown in Figure 9, rotodynamic pumps (that is, centrifugal pumps, in which category mixed- and axial-flow geometries are generally included) would be too low in efficiency to be practical. Rather, rotary positive displacement pumps take over, there being a transition through the drag pump domain. Sometimes called a regenerative or periphery pump, the drag pump is actually a rotodynamic machine, developing head peripherally around the impeller through successive passes radially through the blades on both sides until a barrier is reached at some point on the periphery, where the fluid is then discharged.⁶

The screw pump, on the other hand, is a truly positive displacement (rotary) machine. It can have two, three, or more meshing screws and can move large quantities of fluid—both single- and multiphase—against a large pressure difference Δp , giving it a specific speed range that extends well into centrifugal pump territory. Screw pump rotors are often configured back-to-back in a double-suction arrangement, as this balances the considerable axial thrust of these rotors. Not shown is the progressive cavity pump, which has a single screw surrounded by an elastomer sealing member.

Lower flow rates are readily accommodated by the vane pump, whereas gear pumps handle a higher range of pressure differences at such flow rates. Finally, extremely high pressures are produced by reciprocating pumps, the specific speed range of which extends off the figure on the left.

Positive displacement pumps appear in Figure 9 in order to provide perspective. The concept of specific speed is not generally applied to these machines, because a given positive displacement pump can have such a wide range of pressure-rise capability at a chosen flow rate and speed as to make it difficult to associate a given rotor geometry with a particular value of specific speed. On the other hand, a unique rotodynamic pump geometry is readily associated with the specific speed of the BEP of such a machine.

Performance of Optimum Geometries Figure 9 enables one to easily identify the pump stage types associated with required pumping tasks in terms of head, flow rate, and rotational speed. Beyond this general picture is the related performance of a real pump geometry in a real fluid. Although, for centrifugal pumps, the specific speed has the major effect on performance, the available NPSH and the viscosity of the pumpage also have an influence. These are evident in the following formal statement of the efficiency of an optimized pump (cf Eq. 35)

$$(\eta_{\max})_{\{G_l\}_{\text{opt}}} = f(\Omega_s, \Omega_{ss}, \{2 \cdot \Phi\}, \{\Gamma_p\}, \{\Sigma\}, R_{e,H,Q}) \quad (39)$$

where the radius r_2 , representing the size, has been eliminated from the other variables in Eq. 35 by introducing the *suction specific speed* Ω_{ss} and the *head-flow Reynolds number* $R_{e,H,Q}$, which are defined in Eqs. 40 and 41:

$$R_{e,H,Q} = \frac{\sqrt{Q} (g \Delta H)^{1/4}}{\nu} = R_e \sqrt{Q_s} \psi^{1/4} \quad (40)$$

$$\Omega_{ss} = \frac{\Omega \sqrt{Q}}{(g \text{ NPSH})^{3/4}} = \frac{\sqrt{Q_s}}{(\tau_2/2)^{3/4}} \quad (41)$$

where, the common form of the suction specific speed, called N_{ss} , is given in commercial U.S. units by (Eq. 42)

$$N_{ss, \{U.S.\}} = \frac{N(\text{rpm}) \sqrt{Q(\text{gpm})}}{[\text{NPSH}(\text{ft})]^{3/4}} = \Omega_{ss} \times 2733.016 \quad (42)$$

Size Effect For sufficiently high NPSH (or sufficiently low suction specific speed) and low viscosity (or high Reynolds number), real pumps also possess a strong size effect on efficiency. This is because, in normal manufacturing processes, the clearances δ preventing internal leakage Q_L (for example, past the impeller sealing rings in Figure 2) do not scale up in proportion to the size (represented by r_2), nor do the surface roughness heights ϵ . Thus, a larger pump tends to be more efficient. Strictly speaking, however, the geometry of the larger pump is not the same as that of the smaller pump, and this forces one to modify Eq. 39 by reintroducing two of the length ratios G_i that were part of the set $\{G_i\}$ in Eq. 35 which characterize the hydraulic shape of the machine. Thus, Eq. 39, revised to reflect these realities, becomes

$$(\eta_{\max})_{\{G_i\}_{\text{opt}}} = f_1 \left(\Omega_s, \Omega_{ss}, R_{e,H,Q}, \{2 \cdot \Phi\}, \{\Gamma_p\}, \{\Sigma\}, \frac{\epsilon}{r_2}, \frac{\delta}{r_2} \right) \quad (43)$$

A study of a large number of commercial centrifugal pumps by H. H. Anderson⁷ has quantified Eq. 43 for such machines. These pumps were all operating in water and had

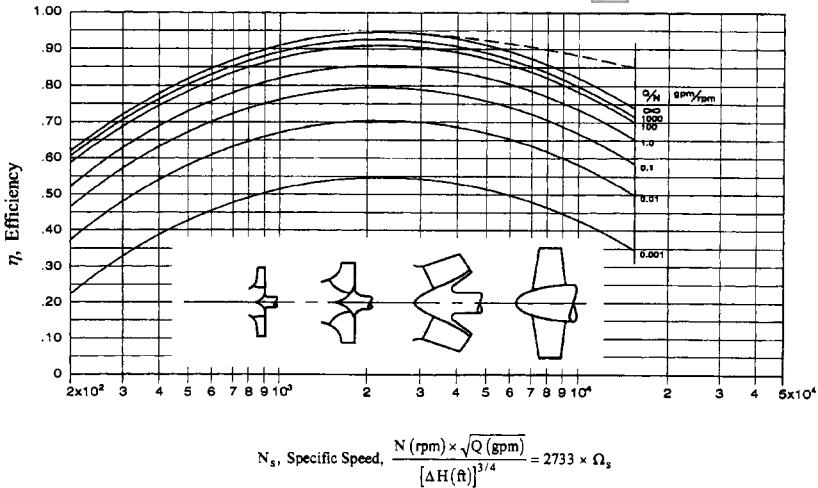


FIGURE 10 Efficiency at the BEP of centrifugal pumps versus specific speed, size, and shape—adapted from Anderson⁷ for $X = 1$

Note: Actual experience for $N_s > 2286$ shows higher efficiency, as indicated by the dashed line. $\text{m}^3/\text{h} = 0.227 \times \text{gpm}$.

sufficient NPSH for performance not to be influenced by Ω_s . The results are given by Eq. 44, which is plotted in Figure 10 for the quantity $X = 1$:

$$\eta = 0.94 - 0.08955 \times \left[\frac{Q(\text{gpm})}{N(\text{rpm})} \times X \right]^{-0.21333} - 0.29 \times \left[\log_{10} \left(\frac{2286}{N_s} \right) \right]^2 \quad (44)$$

where

$$X = \left[\frac{140}{\epsilon(\mu\text{-in.})} \right]^2 = \left[\frac{3.56}{\epsilon(\mu\text{m})} \right]^2$$

Eq. 44 is a combination of separate relationships developed by Anderson for efficiency and speed as functions of flow rate⁷. Included is a correction for specific speed that is too conservative for $N_{s(\text{US})} > 2286$ or Ω_s greater than about unity. With this qualification, Figure 10 is a useful representation for centrifugal pumps and is often as far as many users go in determining the performance of these machines.

Viscosity Effects Centrifugal pump geometries have not generally been optimized versus Reynolds number—often because the effect on hydraulic shape is not very great except for the highest viscosities of the pumpage, and a given application can sometimes experience a substantial range of viscosity. Studies of conventional centrifugal pumps over a range of Reynolds number have been combined in nomographic charts in the Hydraulic Institute Standards, which yield correction factors to the head, efficiency, and flow rate of the BEP of a water (low viscosity) pump in order to obtain the BEP of that pump when operating at higher viscosity⁸. Figure 11 is a presentation of these correction factors in terms of the head-flow Reynolds number. Values of the head correction factor C_H are also given for flow rates other than that of the BEP, enabling one to produce a completely corrected curve of head vs. flow rate. Strictly speaking, in view of Eq. 43, each pump geometry has a unique set of such correction factors, yet the data presented in Figure 11 have been widely utilized as reasonably representative of conventional centrifugal pumps.

NPSH Effects In many cases, the available NPSH is low enough, or the suction-specific speed Ω_s at which the pump stage must operate is high enough for significant two-phase activity to exist within the impeller. This is to be expected in centrifugal impellers of water

Viscous Fluid Effects on Pumps

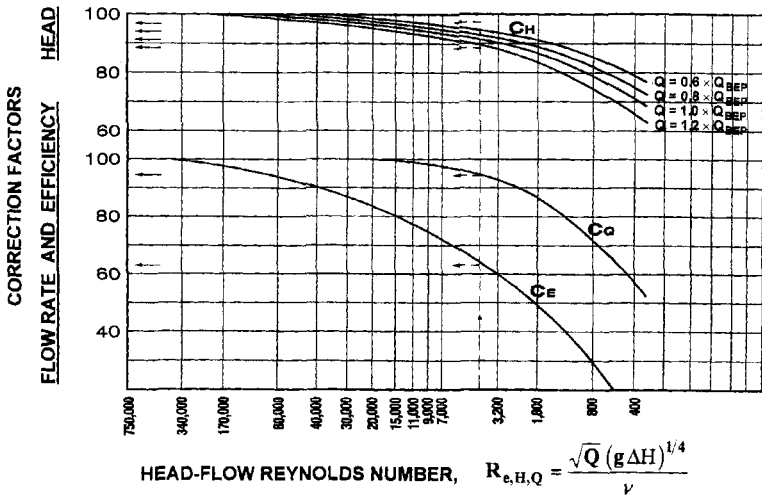


FIGURE 11 Viscous fluid effects on centrifugal pumps—adapted from Hydraulic Institute ANSI/HI 2000 Edition Pump Standards, Reference 8. Values of the correction factors C_Q , C_H , and C_η are each shown as the percent of the water flow rate, efficiency, and head, respectively, that is realized for operation of the same pump in a high-viscosity fluid (low Reynolds number). Except as noted for the C_H curves, these are BEP quantities. The BEP values of Q and H in water operation, together with the higher viscosity of interest, are used in forming the Reynolds number.

pumps if the available Ω_{ss} is greater than about 3 to 4 (or $N_{ss,(U.S.)} = 8,000$ to 11,000). In such a case, Ω_{ss} and the vaporization quantities $\{2\cdot\Phi\}$ in Eq. 35 dictate a profound change in the impeller geometry into that of an inducer. The inducer has an entering or “eye” diameter that is significantly enlarged—together with tightly wrapped helical blading. Often the inducer is a separate stage that pressurizes the two-phase fluid as needed to provide a sufficiently low value of Ω_{ss} at the entrance of the more typical impeller blading that is immediately downstream of the inducer. If the two-phase fluid is near its thermodynamic critical point, the $\{2\cdot\Phi\}$ operate to greatly reduce the amount of two-phase activity within the pump. (At the critical point, the liquid and gas phases are identical, and therefore both have the same specific volume.) An example is the pumping of liquid hydrogen, for which an inducer is unnecessary until much higher values of Ω_{ss} are reached. Moreover, inducers—typically limited to Ω_{ss} -values of about 10 ($N_{ss,(U.S.)} = 27,000$) in water—can, at sufficiently low tip speed, operate at zero NPSH, which corresponds to an infinite value of Ω_{ss} ⁹.

Pumping Entrained Gas In addition to the liquid’s own vapor (which is the gas involved in the NPSH-effects discussion), many pumping applications deal with a different gas; that is, a different substance from the liquid being pumped. The effects of this gas on performance arise from a) the volume flow rate of the gas at the inlet, b) the pressure ratio of the pump, which determines how far into the impeller this gas volume persists; that is, how much it gets compressed, and c) how much of the gas dissolves in the liquid as the pressure increases within the pump, which depends on both the solubility and the degree of agitation of the fluid produced by the pump. The set of fluid properties associated with these gas-handling phenomena are represented by $\{g_p\}$ in Eq. 34, the dimensionless form (Eqs. 35 and 43) of this set being $\{\Gamma_p\}$. Generally, for typical commercial centrifugal pumps, the performance under such conditions usually manifests itself as a loss of pressure rise, which is reasonably stable up to an inlet volume flow rate fraction of gas to liquid of 0.04 to 0.07¹⁰. Inducers can handle larger inlet volume fractions of gas, and, under Dalton’s law for partial pressures, the liquid’s own vapor also occupies the volume of the gas bubbles.

Single and multistage centrifugal pumps have been built that handle far greater gas volume than these single-stage values^{11,12}; moreover, multiphase rotary positive displacement screw pumps can handle gas volume fractions up to 1 (100% gas)¹².

Effects of Slurries and Emulsions Finally there is the influence of the dimensionless quantities (Σ) in Eq. 43. Impeller and casing design are altered so as to reduce wear-producing velocities if the pumpage is a slurry of solids contained in a carrier liquid. (Slurry pumps are usually single-stage machines with a collector or volute casing surrounding the impeller.) This usually means a smaller impeller eye diameter (which, as can be seen in Figure 6, reduces the inlet relative velocity W_{1i}) and a larger radial distance from the impeller to the surrounding volute because the circumferential velocity component V_θ of the fluid emerging from the impeller (also seen in Figure 6) slows down with increasing radial position and is then lower in the volute passageway¹³. Performance also is altered, depending on the composition and concentration of the slurry. These are complicated non-Newtonian flows and are covered in detail elsewhere in this book in conjunction with a thorough treatment on solids-handling pumps. Emulsions are another example of such flows, many of which are destroyed by excessive local shear in the fluid. For this reason, screw pumps are sometimes utilized for emulsions rather than oversized, slow-running centrifugal pumps. Except for thin layers of the fluid at the clearances, most of the flow in a screw pump experiences very little shear in comparison to the flow through a centrifugal pump.

Electromagnetic Effects Not appearing in Eq. 43 are quantities associated with electromagnetic phenomena. For example, electric current flowing radially outward through fluid contained in an axially directed magnetic field is capable of producing a rotating flow. Called a hydromagnetic pump, this device is therefore "centrifugal," yet it has no moving parts. Such pumps have been used for liquid metals and could be made reasonably efficient for any pumpage with high conductivity.

DESIGN PROCEDURES

Establishing the Pump Configuration The first step in designing a pump is to determine the type and number of stages that are needed to meet the given set of operating conditions, usually Q , p_1 , p_2 , available NPSH (= $NPSHA$) and specific gravity of the fluid. If the pump must meet several such sets of operating conditions, one set has to be chosen for the BEP or design point so all the others are satisfied, if possible. Making the proper choice of this BEP may require some iteration: first making a trial choice, doing a preliminary design, and determining the corresponding pump performance characteristic curves, and then repeating these steps if necessary.

Pump rotative speed N (rpm) must be chosen in order to proceed. Selection of the highest practical rpm is desirable because it yields the smallest size and therefore usually the lowest cost and easiest containment of system pressure. If, for the chosen number of stages, the stage specific speed is too low, Figure 10 indicates that efficiency is generally improved with greater speed. The maximum possible rpm is that which yields a value for the suction specific speed Ω_{ss} (Eq. 41) or N_{ss} (Eq. 42) that the first stage of the pump can accommodate, where NPSH is found respectively from Eq. 41 or Eq. 42 as follows:

$$NPSH = \frac{1}{g} \left(\frac{\Omega}{\Omega_{ss}} \right)^{4/3} Q^{2/3} \text{ or } NPSH(ft) = \left(\frac{N(rpm)}{N_{ss}(U.S.)} \right)^{4/3} [Q(gpm)]^{2/3} \quad (45)$$

Typically, the Ω_{ss} -capability of an impeller does not exceed a value that is somewhere in the range 3 to 4.5 ($N_{ss} = 8,200$ to $12,300$), depending on minimum (off-design) flow rate requirements, and it is typically 10 (27,000) or less for inducers. Generally, the supplier furnishes a pump that has a value of NPSH-capability (or " $NPSHR$ ") smaller than the stated $NPSHA$. This difference or " $NPSH$ -margin" is desirable if there is any uncertainty as to the true value of $NPSHA$. It is essential for high values of Δp_{stg} ; (see discussion of

"high-energy pumps" further on in this section). Often a higher speed can be employed if a double-suction impeller (entry of fluid from both sides) can be used, as then only half of the given Q enters each side of this impeller, and only that half can be used in the Ω_s -equation applying to that fluid and impeller type.

The double-suction configuration is popular for large, single-stage pumps because the axial thrust is nominally zero. It is also found as the first stage in some multistage pumps, in which case the arrangement of the remaining impellers can be "back-to-back" (half of them facing in one direction and half opposite) to achieve axial thrust balance. On the other hand, the interstage flow passages are simpler if these impellers are all facing in the same direction—in which case the thrust is opposed by a balancing drum or disk as described in Section 2.2.1. Forming the specific speed from N , Q , and ΔH (from Eq. 3) and referring to Figure 9 for the kind of impeller (which then gives one an idea of the other hydraulic components (inlet passage, diffuser and collector, and so on) and Figure 10 for the expected efficiency, one can decide whether the pump would perform better and still meet any installation and size restrictions with more than the one stage implied by this first N_s -calculation.

If the user requires the performance characteristics of a positive displacement pump—such as a wide range of pressure-rise at a nearly constant flow rate Q —selecting such a pump may be possible if the specific speed is not too high. (See Figure 9.) The choice of speed for positive displacement pumps is sometimes determined by mechanical considerations rather than suction capability.

Sizing the Pump The next step is to determine the approximate size of a pump or pump installation. Beginning with the impeller, the agent of the energy transfer to the fluid in a pump, one utilizes the results of the pump hydraulic geometry optimization process. This also yields the proportions of the velocity diagrams that correspond to the geometry associated with the desired value of specific speed. These in turn lead to two major sizing factors for an impeller, which are plotted for typical commercial pumps in Figure 12; namely the head coefficient ψ (originally defined in Eq. 31) and the outlet flow

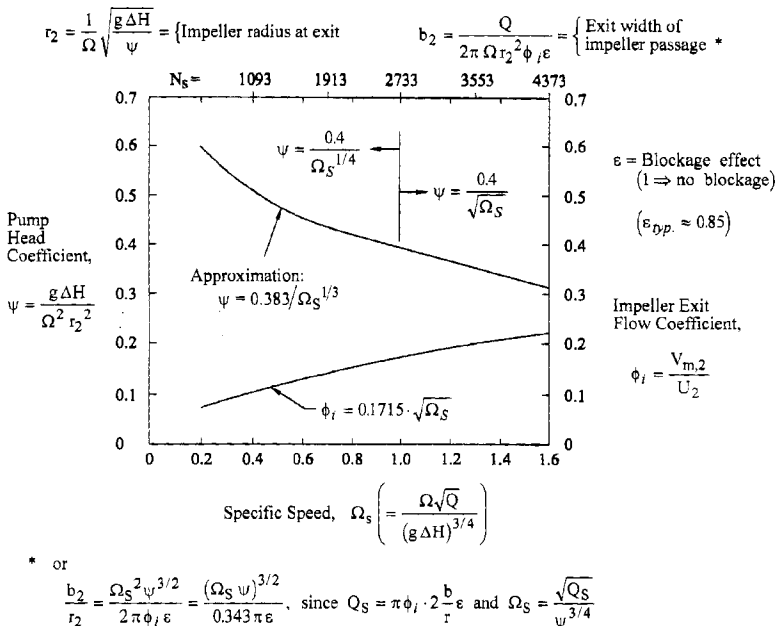


FIGURE 12 Sizing factors for the diameter and width of typical impellers—adapted from Stepanoff⁸

coefficient ϕ_i defined on this figure⁵. Equations in the figure show how these factors are used to determine the exit radius r_2 (or diameter D_2) and the width b_2 (Figure 8) respectively. The flow coefficient ϕ_i is obviously a velocity component ratio. For the head coefficient of a pump with zero inlet prewhirl to the impeller (that is, $V_{\theta,1} = 0$), the relevant velocity ratio (see Figure 3) is $V_{\theta,2}/U_2$, which in view of Eqs. 15b and 31 is equal to the ideal head coefficient ψ_i , as this corresponds to the ideal head H_i . Referring to Eq. 15c, it is obvious that the actual head coefficient $\psi = \eta_{HY} \times \psi_i$, or, in the (common) case of zero prewhirl, $\psi_i = V_{\theta,2}/U_2$.

Overall pump dimensions are conveniently viewed in terms of factors times the impeller diameter. The overall diameter exceeds that of the impeller due to the surrounding casing. For single stage pumps, this casing will include a volute and perhaps a set of diffuser vanes immediately surrounding the impeller; so, the casing diameter can be 50% greater than D_2 or more. On multistage pumps (many with radial-outflow diffusers and return vanes), this excess is often less than 50%.

Single-stage pump axial length includes provision for inlet passageways and bearing housings and is therefore approximately the same as the overall pump diameter. On the other hand, the axial length of a stage in a multistage pump is often less than half of the impeller diameter. Minimizing this "stage length" (and diameter) is a goal of competitive pump designers in the quest to create a machine of light weight and low cost. But too small a stage length is accompanied by inferior hydraulic performance because not enough room is provided for the passages around and within the impeller to turn the fluid with minimum loss. Further, the bearing housings and the suction and discharge "heads" or end pieces must be considered in arriving at the overall length of a multistage pump.

This sizing discussion so far has focused on pumps with a radial-outflow geometry. For the axial-flow geometries of higher specific speeds, the situation is simpler. The diameters of the propeller (Figure 9) and any stationary vanes downstream (or upstream) are essentially the same. The approach and discharge piping is of about the same size. A simple guideline for the maximum radius of the propeller is found from the following approximate extrapolation of the ψ -curve of Figure 12:

$$\psi_t = \frac{0.53}{\Omega_s} \quad (46)$$

where the outer radius of the propeller $r_{t,2}$ should be substituted for the mean exit radius r_2 in the ψ -definition. Thus the ψ -curve is in reality a ψ_t -curve for non-radial-flow pumps. For small departures from radial flow, as illustrated in Figure 8, the ψ -curve of Figure 12 can be used either way.

These guidelines are often all that pump users or fluid system designers need to plan their installations. To get beyond this overview, one must pursue the hydraulic design in detail—as afforded by the following development and examples.

Designing the Impeller Determination of the geometrical features of the impeller is generally accomplished in the following order: a) the "eye" radius r_e , b) the exit radius r_2 or $r_{t,2}$, and c) the exit width b_2 or, in the case of mixed- and axial-flow impellers, the hub exit radius $r_{h,2}$ —all of which form the starting point for d) shaping the hub and shroud profiles (Figure 13); and, finally, e) construction of the blades.

a) The eye. The inlet radius of the impeller eye r_e (Figure 13) is nearly the same as $r_{t,1}$, which is the diameter of the tips of the impeller blades at the inlet. This emerges after the eye flow coefficient $\phi_e = V_e/U_e$ [the ratio of the one-dimensional axial velocity entering the eye (Figure 8) to the tangential speed of the impeller eye $U_e = \Omega r_e$] is known:

$$\phi_e = \frac{Q/A_e}{\Omega r_e} = \frac{\bar{V}_e}{U_e} \quad (47)$$

$$A_e = \pi(r_e^2 - r_s^2) \quad (48)$$

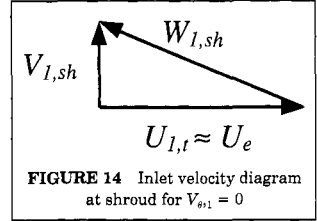
TABLE 1 NPSH correlations for breakdown of pressure rise due to two-phase activity in pump

$$\tau = \frac{NPSH}{\frac{U_e^2}{2g}} \quad \Omega_{ss} = \frac{\sqrt{\pi\phi_e \left(1 - \frac{r_s^2}{r_e^2}\right)}}{\left(\frac{\tau}{2}\right)^{3/4}}$$

a) Impellers (Gongwer¹⁴)

$$NPSH_{3\%} = k_1 \frac{V_e^2}{2g} + k_2 \frac{W_{1,sh}^2}{2g} = \left(\frac{V_{1,sh}}{V_e}\right)^n \frac{V_e^2}{2g} + k_2 \frac{W_{1,sh}^2}{2g}$$

$$\text{For } V_{\theta,1} = 0: \tau_{3\%} = \frac{NPSH_{3\%}}{\frac{U_e^2}{2g}} = (k_1 + k_2)\phi_e^2 + k_2$$

**FIGURE 14** Inlet velocity diagram at shroud for $V_{\theta,1} = 0$

where,

for large pumps (Gongwer¹⁴), $k_1 = 1.4$, and $k_2 = 0.085$

and, for typical pumps, $k_1 \approx 1.69$, and $k_2 \approx 0.102$

b) Inducers (curve fit to the data of Stripling and Acosta¹⁵)

$$\tau = 0.02 + 0.02 \left[\log_{10} \left(\frac{\phi_e \sin \beta_{b,1,sh}}{1 + \cos \beta_{b,1,sh}} \right) + 3 \right]^{3.5}$$

eye radius r_e can be combined to yield the ratio of eye size to that of the OD for centrifugal impellers; viz.:

$$\frac{r_e}{r_2} = \frac{D_e}{D_2} = \frac{\Omega_s^{2/3} \sqrt{\psi}}{\left[\pi\phi_e \left(1 - \frac{r_s^2}{r_e^2}\right) \right]^{1/3}} \quad (51)$$

Except for inducers, which have very low values of ϕ_e , this ratio is affected mainly by specific speed as illustrated in Figure 9.

c) The exit width b_2 . The equation for exit passage width b_2 in Figure 12 can be used for radial-outflow and mixed-flow impellers, r_2 being located halfway across the passage. This involves the exit flow coefficient or meridional velocity ratio $\phi_i = V_{m,2}/U_2$, the lower curve of Figure 12 being for typical values of this quantity. The “openness” factor ε allows for blockage due to blade thickness and to the buildup of boundary layers on the surfaces of the passageways (blades and hub and shroud). The value of ε is generally between 0.8 and 0.9, the higher figure applying to larger machines. For axial-flow impellers or propellers and inducers, a choice of the hub-to-tip radius ratio at the exit defines the passage width instead. This ratio decreases with specific speed from about $\frac{2}{3}$ at the right end of Figure 12 to $\frac{1}{3}$ or less at the highest specific speeds.

d) Hub and shroud profiles. With the eye and the outlet sizing established, the two are connected by specifying the hub and shroud profiles. Some texts illustrate the variation

of hub and shroud profiles with specific speed¹⁶. Although these are excellent guidelines coming from experience, what follows is the approach one would take to synthesize these shapes on the basis of fundamental fluid dynamical considerations, at the same time taking experience into account. Referring to Figure 13, an acceptable geometry can be achieved by following these guidelines:

- i. Maintaining the meridional flow area $2\pi r_{b,1}b_1$ at the blade leading edge at about the same as it is at the eye, namely $\pi(r_e^2 - r_s^2)$, or slightly larger to compensate for blade blockage; but then gradually increasing it versus meridional distance to the generally larger value already established at the exit, namely $2\pi r_2b_2$.
- ii. Choosing the minimum radius of curvature R_{sh} of the shroud to be about half the radial opening at the eye. This avoids excessive local velocity $V_{1,sh}$ at the blade leading edge, (Figure 14 in Table 1.) This has two consequences. Shaping the impeller blades to match a widely varying meridional approach velocity can complicate the construction of these blades. Also, on first-stage impellers, if $V_{1,sh}$ is too great, the local pressure at that location will be closer to the vapor pressure, increasing the required NPSH (or $NPSH_{3\%}$). This is due to a larger resulting value of the empirical factor k_1 , presented in Table 1 as the n^{th} power of the velocity ratio $V_{1,sh}/V_e$. $n \approx 2$ is expected from Bernoulli considerations, and if $V_{1,sh}/V_e = 1.3$, the value of 1.69 emerges for k_1 as shown in Table 1. $V_{1,sh}/V_e = 1.25$ is a typical choice of designers for setting the impeller blades at inlet. A CFD solution of the combined inlet passage and impeller blading would show the actual resulting value of this velocity ratio, which might then lead to a change in the impeller blading or radius of curvature R_{sh} . The powerful effect on this velocity ratio of the sharpness of the turn that the flow negotiates in coming into the impeller eye and turning toward the radial direction can be seen from the following finite-difference form of the differential equation governing axisymmetric inviscid flow in the impeller eye region:

$$\frac{\Delta V_m}{V_e} = \frac{b_e}{R}$$

where R is the average radius of curvature of the flow in the meridional plane of Figure 13, ΔV_m is the difference in V_m from hub to shroud over a distance b_e perpendicular to the direction of flow that moves at an average meridional velocity V_e . Thus a sharp turn with its small value of R produces a large value of $\Delta V_m/V_e$, which in turn corresponds to a large value of the velocity ratio $V_{1,sh}/V_e$.

- iii. Shaping the hub profile compatibly with the guidelines as stated earlier. This is best done after making an initial estimate of the shroud profile as outlined previously. The distribution of meridional flow area from the eye to the exit should then be specified. From this, a hub profile will emerge. As seen in Figure 13, the hub of a radial-outflow impeller becomes essentially radial over the outer portion of its extent. If this does not result from this procedure, appropriate adjustments can be made to the shroud profile and the process repeated.
- iv. For a high-specific-speed, axial-flow impeller, or inducer, the hub profile is often a cone or a reverse curve between a smaller radial location at inlet to a larger one at outlet, the latter radius decreasing with increasing specific speed as mentioned earlier. A cone or cylinder for the shroud profile is often found in such machines.

e) Construction of the blades. The blades are designed by i) selecting the locus of the leading and trailing edges in the meridional plane, ii) establishing the surfaces of revolution (streamwise lines in the meridional plane) from inlet to outlet along which the construction proceeds, iii) selecting the inlet angles, iv) selecting the outlet angles, v) establishing the number of blades, and vi) obtaining the blade coordinates from inlet to outlet:

- i. *Leading and trailing edge loci.* If every point along the leading and trailing edges is revolved about the axis of rotation so as to lie in one meridional plane, the loci of these

edges appear as shown in Figure 13. The outer or shroud end of the blade leading edge is positioned at or near the minimum radial location; that is, at or near the eye plane, whereas the inner or hub end is typically well back and largely around the corner along the hub profile. These locations are desirable; first, at the shroud, because the absolute velocity V (typically $= V_m$) approaching the blade begins to decelerate beyond the eye plane, so starting the blade ahead of this decelerating region tends to prevent separation of the fluid from the shroud surface due to the pumping action in the blade channels¹⁷; and secondly, being far enough along the hub in the streamwise direction to avoid impractical blade shapes (excessive twist, rake, and so on) that would make both the construction and the flow inefficient. The locus of the blade trailing edges is normally straight in the meridional plane and is axial in orientation for most centrifugal pumps. At the higher specific speeds, this locus becomes more and more slanted until it takes on the nearly radial orientation it has for a propeller (Figure 9).

- ii. *Surfaces of revolution for blade construction.* Developing the coordinates of the blades along three streamwise surfaces of revolution—the hub, mean, and shroud, whose intersections with the meridional plane appear as streamwise lines in that plane—usually provides a sufficient framework for shaping the blades of an impeller. However, for high specific-speed impellers, where the passage width in the meridional plane b (Figure 13) is large (about equal to or greater than the meridional distance from leading to trailing edge), definition along two intermediate surfaces of revolution is also needed to achieve a satisfactory design.

The “mean” line is one that is representative of the flow from a one-dimensional standpoint as well as for the construction of the blades. Precisely, this is the mass-averaged or “50%” streamline (that is, the streamline for $\psi = 0.5$ in Figure 14)—which evenly divides the mass flow¹⁷. This line is reasonably and conveniently approximated by the “rms streamline,” that is, the line that would result in a uniform meridional velocity distribution from hub to shroud and therefore equal areas $2\pi r \Delta n$ normal to the meridional velocity component V_m . In this case, Δn ($= \Delta b$) is the spacing between the rms streamline and the hub or shroud line. This would put each point on the mean line at the root mean square radial position along a true normal to the meridional streamlines; hence, the “rms” terminology.

- iii. *Inlet blade angles.* The blade angles are set to match the inlet flow field. This is done where each of the previously chosen surfaces of revolution (that intersect the meridional plane in the streamwise lines just described) crosses the chosen locus of the blade leading edges in the meridional plane. At each such crossing point, an inlet velocity diagram of the type shown in Figure 3 is plotted in a plane tangent to the surface of revolution at that point. (Figure 3, representing a purely radial-flow configuration, is a view of such a plane, as the surfaces of revolution are then simply disks.) Each such velocity diagram or triangle contains a specific value of the angle $\beta_{r,1}$ between the relative velocity vector W_1 and the local blade speed vector $U_1 = \Omega r_1$.

The corresponding blade angle $\beta_{b,1}$ between the mean camber line of the blade and the circumferential direction is set equal to $\beta_{r,1}$ or slightly higher than this to allow for the higher $V_{m,1}$ caused by non-zero blade thickness at the leading edge and to allow for higher flow rates that may be called for at off-design conditions. To construct the triangle, one first plots U_1 and then $V_{m,1}$, which is taken from a CFD or other analysis of the meridional velocity variation across the inlet eye as discussed above under *d) Hub and shroud profiles* or is chosen as the mean value $Q/2\pi r_{b,1} b_1$ (Figure 13) at the rms streamline. It is adjusted from experience at the shroud and hub. Likewise, if any prewhirl $V_{\theta,1}$ is delivered to the impeller, it must be taken into account as illustrated in Figure 3.

- iv. *Outlet blade angles.* Whereas the inlet velocity diagrams enable the designer to correctly set the blades to receive the incoming fluid with minimum loss, the outlet velocity diagram displays the evidence—through the magnitude of the circumferential velocity component $V_{\theta,2}$ that the intended head will be delivered by the pump in accordance with Eq. 15c. As shown in Figure 3, $V_{\theta,2}$ is determined—for the given impeller tip speed U_2 —by the exit relative flow angle $\beta_{r,2}$ in conjunction with the exit meridional

velocity component $V_{m,2}$. This value of V_m is somewhat larger than that given by Eq. 16 because of a) blockage due to blade thickness and boundary layer displacement thickness and b) the presence of any leakage flow Q_L (Figure 2 and Eq. 11) that may also be flowing through the impeller exit plane or Station 2.

Well inward of the exit plane, the direction of the one-dimensional relative velocity vector W can be assumed to be parallel to the blade surface; however, in the last third of the passage, the blade-to-blade distribution of the local relative velocity changes due to the unloading of the blades at the exit. This produces a deviation of the direction of W_2 from that of the blade. This deviation, called "slip" in centrifugal pumps, results in less energy being delivered to the fluid by the impeller than would be the case if there were "perfect guidance" such as would occur with an infinite number of blades. Accordingly, in the outlet velocity diagram of Figure 15, the relative flow angle β_f is less than the blade angle β_b . This deviation is quantified by the "slip velocity" V_s . The magnitude of V_s depends on the distribution of loading along the blades from inlet to exit and therefore on the geometry of the flow passages and the number of blades. (Without slip, W_2 is the same as the "geometric" relative velocity $W_{g,2}$ shown in the figure.) The slip factor $\mu = V_s/U_2$ —typically between 0.1 and 0.2—was determined theoretically by Busemann for frictionless flow through impellers with zero-thickness, logarithmic-spiral blades (constant- β from inlet to exit) and a two-dimensional, radial-flow geometry with parallel hub and shroud¹⁸. Applicability of this theory to typical impellers, despite the differences in geometry and the real fluid effects, was found to be good by Wiesner, who represented Busemann's results by the following convenient approximation¹⁹:

$$\left. \begin{aligned} \mu &\equiv \frac{V_s}{U_2} = \frac{\sqrt{\sin \beta_2}}{\bar{n}_b^{0.7}} \text{ if } \frac{r_1}{r_2} < \varepsilon_{limit}, \text{ where } \varepsilon_{limit} = \frac{1}{e^{\left(\frac{8.16 \sin \beta_2}{\bar{n}_b}\right)}} \\ \mu &\equiv \frac{V_s}{U_2} = 1 - \left\{ \left(1 - \frac{\sqrt{\sin \beta_2}}{\bar{n}_b^{0.7}} \right) \times \left[1 - \left(\frac{\frac{r_1}{r_2} - \varepsilon_{limit}}{1 - \varepsilon_{limit}} \right)^3 \right] \right\} \text{ if } \frac{r_1}{r_2} > \varepsilon_{limit} \end{aligned} \right\} \quad (52)$$

A broader, empirical slip correlation for pumps was developed by Pfeleiderer, taking into account impeller geometry and blade loading, as well as the influence of the downstream collecting system (volute or diffuser)²⁰. Pfeleiderer computes the slip velocity as the product of a slip factor p and the impeller exit tangential velocity $V_{a,2}$, where p is computed as shown in Table 2. This table also contains a simple example; namely, a

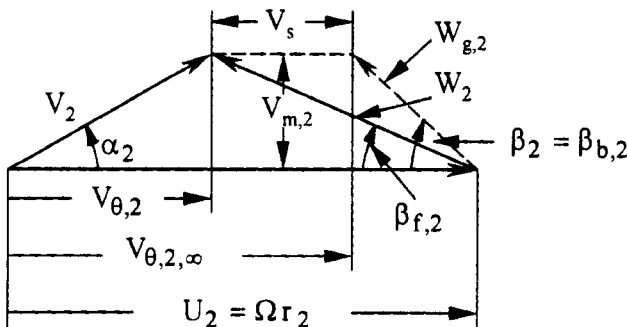


FIGURE 15 Impeller outlet velocity diagram

TABLE 2 Pfleiderer's slip formula

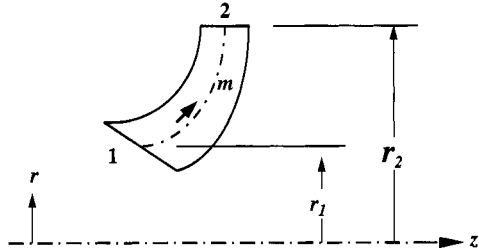
Slip velocity, $V_s = pV_{\theta,2}$

$$\text{Thus, } \frac{V_{\theta,\infty,2}}{V_{\theta,2}} = 1 + p$$

$$\text{where } p = \frac{\psi' r_2^2}{\int_1^2 n_b r dm}$$

and where $\psi' = a \times (1 + \sin \beta_{b,2})$

$$\text{and } a = \left\{ \begin{array}{l} 0.6 \text{ for vaned diffuser;} \\ 0.65 \text{ to } 0.85, \text{ volute;} \\ 0.85 \text{ to } 1.0, \text{ vaneless diffuser.} \end{array} \right\}$$



Example: $r_2 = 1$; $r_1 = 0.4$; $V_{\theta,2} = \frac{U_2}{2}$; $n_b = 7$; $\beta_{b,2} = 20$ degrees; $a = 0.8$ (volute)

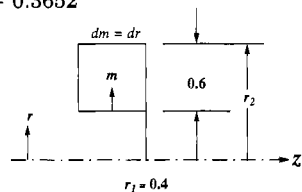
Results:

$$\psi' = 0.8 \times (1 + 0.342)$$

$$\int_1^2 n_b r dm = n_b \times \frac{1}{2} \times (r_2^2 - r_1^2) = n_b \times \frac{r_2 + r_1}{2} \times (r_2 - r_1)$$

$$p = \frac{V_s}{V_{\theta,2}} = \frac{\psi' r_2^2}{\int_1^2 n_b r dm} = \frac{0.8 \times (1 + 0.342) \times 1^2}{7 \times \frac{1 + 0.4}{2} \times 0.6} = 0.3652$$

$$\mu = \frac{V_s}{U_2} = p \times \frac{V_{\theta,2}}{U_2} = 0.3652 \times \frac{1}{2} = 0.1826$$



radial flow impeller of a volute pump, for which the resulting value of μ is 0.1826—versus 0.1498 via Eq. 52; however, in this case the latter result is low by about 15%. A study of the Busemann plots in Wiesner's paper yields $\mu = 0.18$. Yet, if this had been a vaned-diffuser pump, Pfleiderer would have predicted $\mu = 0.1468$ for the same impeller, as it would have delivered more $V_{\theta,2}$ for the same $\beta_{b,2}$, $W_{g,2}$, and, therefore, $V_{\theta,2,\infty}$ (Figure 15). This stems from the factor "a" in Table 2 having the value 0.6 (for a vaned diffuser) instead of 0.8 (for a volute). So by this combination of circumstances—and in this example—Eq. 52 describes the slip of a diffuser pump impeller. But, despite the simplicity of Eq. 52, Pfleiderer's method (Table 2) would appear to be a more rational, comprehensive, and satisfying method for estimating slip in real pumps.

So, to find the outlet blade angle, the designer begins by deciding upon the required value of $V_{\theta,2}$; finds the exit flow angle and other elements of the diagram assuming the existence of slip. Next, the designer computes the slip and then obtains the value of

the outlet blade angle $\beta_{b,2}$. The process is iterative because the forementioned blockage depends on the blade angle as well as the thickness.

- v. *Number of blades.* The choice of the number of impeller blades is influenced by a) interaction of the flow and pressure fields of the impeller and adjacent vaned structures such as the volute tongues or diffuser vanes and b) the need to maintain smooth, attached—and therefore efficient—fluid flow within the impeller passages. The effect of the number of blades on the interaction phenomenon is addressed in the latter part of this section under the topic of high-energy pumps, where this issue becomes critical. Smooth, attached flow is assured if the product of the number of blades and their total arc length ℓ along a given meridional streamline, as illustrated in Figure 16, is of sufficient magnitude. Divided by a representative circumference on that streamline, usually that of the impeller outer diameter (OD), this product is called the solidity σ :

$$\sigma = \frac{n_b \times \ell}{2\pi r_2} \quad (53)$$

If splitter blades are used, the value of n_b changes from impeller inlet to outlet (1 to 2). In that case, the numerator of Eq. 53 is replaced by the implied sum of the arc lengths of all the blades in the impeller.

In practice, solidity varies from about 1.8 at low specific speed ($\Omega_s < 0.4$ or $N_s < 1093$) to slightly less than unity at $\Omega_s = 3$ ($N_s = 8199$). For example, Dicus' curve²¹ is useful for $\Omega_s > 1$ ($N_s > 2733$). This limits the relative velocity reduction that occurs on the blade surfaces. Illustrated in Figure 17, this reduction or diffusion arises from the loading on the blades expressed in terms of the blade-to-blade relative velocity difference ΔW_b :

$$W_s - W_p (= \Delta W_b) = \frac{2\pi}{\Omega n_b} \times \frac{V_{m,o}}{\bar{W}} \frac{d(UV_\theta)}{dm} \quad (54)$$

$V_{m,o}$ is the local meridional velocity component neglecting blockage. (One-dimensionally, $V_{m,o}$ is the value of V_m found from Eq. 16, where the radius r is that from the axis of rotation to the center of the circle of diameter b in Figure 8, which in turn lies on an

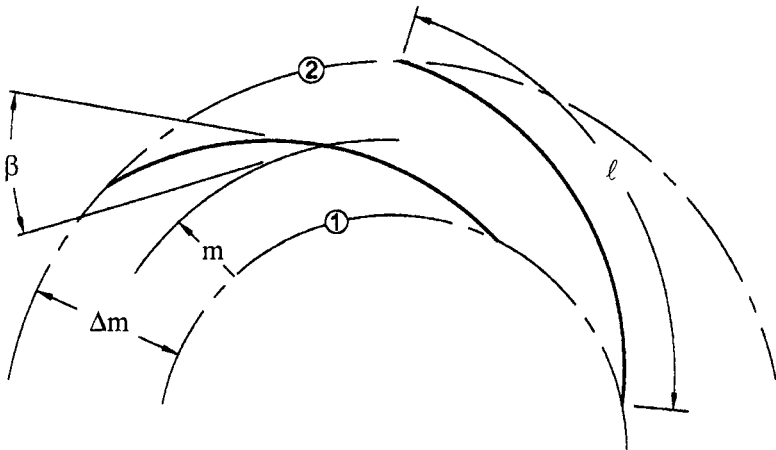


FIGURE 16 Intersection of impeller blades with mean surface of revolution

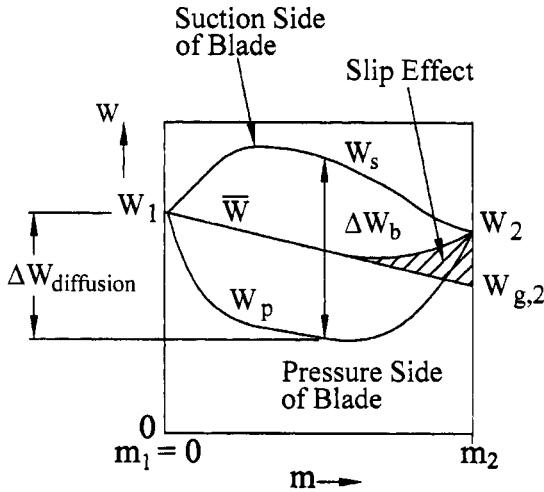


FIGURE 17 Relative velocity distributions

imaginary line in Figure 8 that is normal to the hub, shroud, and intermediate stream surfaces.) Here, ΔW_b emerges by applying Bernoulli equation [Eq. 21 with no change in radius (that is, no change in U) or loss as one traverses from pressure side to suction side of the passage] to the static pressure difference $p_p - p_s$ arising from the delivery of angular momentum to the fluid (Eq. 26). This in turn results from the application of the shaft torque to the blades. It is also assumed in the derivation of Eq. 54 that the *blade-to-blade average relative velocity W lies halfway between the surface velocities W_s and W_p* (which would exist just outside the boundary layers on the blades,) as illustrated in Figure 17. This is a good assumption for efficient flow well within a bladed channel²². ΔW_b is inversely proportional to the solidity because, on the average, from inlet to outlet, Eq. 54 becomes

$$\overline{W_s - W_p} = \overline{\Delta W_b} \approx \left(\frac{2\pi r_2 \overline{\sin \beta}}{n_b \Delta m} \right) \Delta \left(\frac{r}{r_2} V_\theta \right) \quad (55)$$

where it can be seen from Figure 16 and Eq. 53 that the fraction involving the number of blades n_b is the reciprocal of the solidity σ because

$$\ell = \Delta m / \overline{\sin \beta} \quad (56)$$

If splitter blades are used, an appropriate average number of blades is substituted for n_b in Eq. 53.

For unconventional impeller geometries, the foregoing solidity guidelines may be inadequate to assure efficient flow. For any geometry, though, the concept of a diffusion factor D , utilized by NACA researchers²³ to assess stationary cascades of airfoils can be employed. In view of Eqs. 53–56, their equation for D takes the following form for both axial- and non-axial-flow geometries, rotating or not:

$$D = 1 - \frac{W_2}{W_1} + \frac{\Delta \left(\frac{r}{r_2} V_\theta \right)}{2\sigma W_1} \quad (57)$$

This can be deduced from Figure 17 as follows:

$$D = \frac{W_1 - W_2}{W_1} + \frac{\overline{\Delta W_b}}{2W_1} \left(\approx -\frac{\Delta W_{\text{diffusion}}}{W_1} \right) \quad (58)$$

Then, Eq. 57 is obtained through the definitions of the average value of ΔW_b (Eq. 55 with Eq. 56) and σ (Eq. 53). NACA researchers found that losses increase rapidly if $D > 0.6$. However, many centrifugal pump impellers have virtually the same value of relative velocity W at inlet and at outlet—along the rms streamline (Figure 17), so D from Eq. 57 is less than 0.6 on the rms streamline and even negative along the hub streamline. This situation was encountered in accelerating (turbine) cascades and led to the use of local diffusion factors, one for each side of the blade, namely D_p and D_s . Here, inspection of Figure 17 and Eq. 58 leads to

$$D_p = 1 - \frac{W_{p, \min}}{W_1} \quad ; \quad D_s = 1 - \frac{W_2}{W_{s, \max}} \quad (59a \text{ and } b)$$

where the 0.6 limit applies individually to D_p and D_s . Eqs. 59a and 59b, therefore, constitute a more useful form of the diffusion factor concept for assessing the blade loading and the choice of the number of blades in centrifugal pump impellers²⁴.

Finally, the total blade length or number of blades, should not exceed that necessary to limit the diffusion as just described, as this adds unnecessary skin friction drag, which causes a reduction in efficiency. Thus the solidity values given in conjunction with Eq. 53 should not be appreciably exceeded, unless blade load needs to be reduced to lower levels, as with inducers to limit cavitation⁹ or impellers for pumps that must produce lower levels of pressure pulsations.

- vi. *Development of the blade shape.* Blades are developed by defining the intersection of the mean blade surface (really an imaginary surface) or camber line on one or more nested surfaces of revolution. Two such surfaces are formed by the hub and shroud profiles. If the blade shape is two-dimensional (that is, the same shape at all axial positions z), the mean blade surface is completely defined by constructing it on only one such surface of revolution. Generally, however, the shape is three-dimensional and is a fit to the shapes constructed on two or more of these surfaces of revolution; namely, the hub and shroud and usually at least one surface between them. After this final shape is known, half of the blade thickness is added to each side. (Sometimes the full blade thickness is added to one side only, meaning that the constructed surface just mentioned ends up—usually—as the pressure side of the finished blade rather than the mean or “camber” surface. The effective blade angles are then slightly different from those of the pressure side used in the construction process.) The construction along a mean surface of revolution is illustrated in Figure 18. The distribution of the local blade angle β (or more precisely, β_b) is found first by either the “point-by-point” method or the *conformal transformation* method—both of which yield the polar coordinates of the blade, r , θ , and z . These coordinates also depend on the chosen shapes of the intersections of the surfaces of revolution with the meridional plane; that is, the hub, shroud, and mean meridional “streamline” or rms line, as in Figure 18c, and the fact that, on the surface of revolution Figure 18a, $\tan \beta = \text{arc } bc / \text{arc } ac = dm/dy$. The elemental tangential length $dy (= \text{arc } ac)$ is the same on both the surface of revolution (Figure 18a) and in the polar view (Figure 18b). From Figure 18b, it is seen that $dy = r d\theta$, so the “wrap” angle θ is found from

$$\tan \beta_b = \frac{dm}{rd\theta} \quad (60)$$

and r and z are found from the fact that the coordinate m along each of the construction surfaces is a function of r and z (Figure 18c).

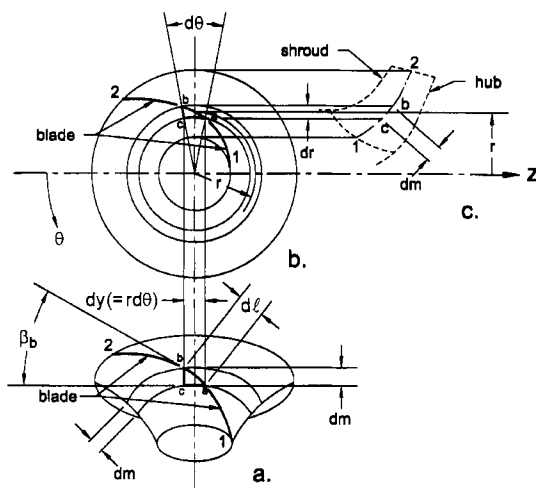


FIGURE 18 Blade construction: a) view of construction surface of revolution; b) polar view; c) meridional view

If the blade is two-dimensional, its mean surface consists of a series of straight-line axial elements, each having a unique r and θ at all z . Such a blade is typical of low-specific-speed, radial-flow impellers, and can be easily constructed by the "point-by-point" method. Here, one specifies the distribution of W_θ —often linear as in Figure 17—after determining the hub and shroud profiles and the corresponding distribution of V_m ¹⁶. In effect, one obtains the distribution of the blade angle β_b by constructing a velocity diagram like the one in Figure 15 at every m -location from inlet (1) to outlet (2) in Figure 18c, dealing only with the "geometric" or nondeviated velocities, in order to get a smooth variation of the blade angle β_b vs m . Allowance is made for blockage due to the thickness of the blades and the displacement thickness of the boundary layers in the passage. The resulting wrap angle θ for each m -point—as well as the corresponding r and z —is then found from Eq. 60. (For convenience in designing the blades, the construction angle θ is often taken as positive as one advances from impeller inlet to exit. For most impellers, this turns out to be opposite to the direction of rotation; and θ is taken in the direction of rotation for most other purposes of pump design and analysis.) As discussed previously in Paragraph iv and illustrated in Figure 17, the actual flow will deviate from the resulting blade via the "slip" phenomenon.

The point-by-point method allows the designer to exercise control over the relative velocity distributions on the blade surfaces (Eq. 54 and Figure 17) via specification of the distribution of W_θ or other velocity component in Figure 15; for example, V_a . This becomes more important if an unconventional impeller geometry is being developed¹⁷.

The point-by-point method can also be used for three-dimensional blades. A simple approach in this respect would be to use this method to determine the blade shape along the rms- or 50%-streamline (that is, on the mean surface of revolution depicted in Figure 18). The shapes on the other streamlines, generally the hub and the shroud, can also be found by this method. The resulting overall blade shape, however, is subject to the condition that the resulting wrap $\theta_2 - \theta_1$ cannot greatly differ on all streamlines without the blade taking on a shape that is difficult to manufacture and which may turn out to be structurally unsound or create additional flow losses. This is because the final blade shape is the result of stacking the shapes that have been established on the nested stream surfaces defined by these meridional streamlines. Blade forces due to twists arising from this stacking could modify the expected flow and cause unexpected diffusion losses.

One way to generate blade shapes along the hub and shroud that have the same (or nearly the same) wrap as that obtained from point-by-point construction of the blade on the mean surface of revolution is to establish the desired inlet and outlet blade angles β_s on each such surface and then mathematically fit a smooth shape $y(m)$ to these end and wrap conditions, where y is the tangential coordinate seen in Figure 18 and defined in Figure 19. A conformal representation of the shapes of the blades resulting from such a procedure on each of the three surfaces is seen in Figure 19. These shapes are sometimes called “grid-lines” or simply “grids”—from the description of the graphical procedure that relates these shapes in the conformal representation to those on the actual, physical surfaces⁵. In such a representation, the blade angles are the same as they are on the physical surface of revolution because $\tan \beta = dm/dy$ and $dy = rd\theta$, also yielding Eq. 60.

If the associated distributions of W_g and V_m are smooth, one can expect to have a satisfactory result if these conformal representations are also smooth. Thus, many skilled designers bypass the computations just described for the point-by-point method and use the *conformal transformation* method of blade design. Here, one simply establishes the grid-line shapes by eye in the conformal plane of Figure 19, specifying the blade angles β at inlet and outlet by the previous procedures as the starting point for drawing each grid-line. This conformal blade shape is then transformed onto the physical surface, the differential tangential distance dy becoming $rd\theta$ on the physical surface (Figure 18) and the differential meridional distance dm being identical in both the conformal and physical representations. If the resulting blade shape appears to be unsatisfactory, the designer repeats this process, possibly first altering the hub and shroud profiles or the blade leading and trailing edge locations on these profiles and recomputing the β s.

Designing the Collector The fluid emerging from the impeller is conducted to the pump discharge port or entry to another stage by the collecting configuration, which can employ one or more of the following elements in combination: a) *volute*s, which can be used for designs of all specific speeds, b) *diffuser* or *stator vanes*, which are often more economical of space in high-specific-speed single-stage pumps and in multistage pumps, and for the latter, c) *return* or *crossover passages*, which bring the fluid from the volute or diffuser to the eye of the next-stage impeller. Generally, the most efficient impeller has

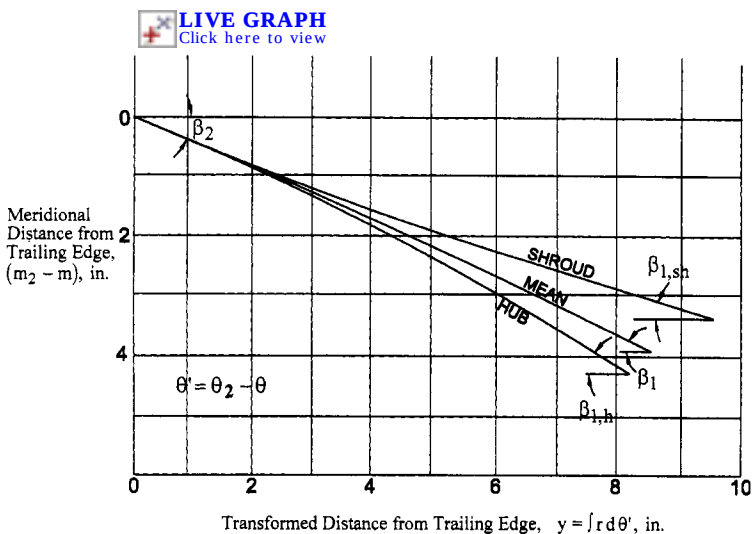


FIGURE 19 Conformal transformation of blade shape: “grid-lines”

a steady internal relative flow field as it rotates in proximity to these configurations. This is assured by all of these elements because they are designed to maintain uniform static pressure around the impeller periphery—at least at the design point or BEP. An exception to this rule is the concentric, “doughnut”-type, “circular-volute” collector, which is used on small pumps or in special instances where the uniform pressure condition is desired at zero flow rate.

The proximity of stationary vanes in these collecting configurations to the impeller must be considered in their design. Called “Gap B,” the meridional clearance between the exit of the impeller blades and adjacent vanes ranges from 4 to 15% of the impeller radius, volutes having higher values in this range than diffusers, and pumps of higher energy level requiring the larger values. If these gaps are too small, the interactions of the pressure fields of the adjacent blade and vane rows passing each other can cause vibration and structural failure of impeller blades, diffuser vanes, and volute tongues.

a) Volute. A volute is built by distributing its cross-sectional area on a “base circle” that touches the tongue or “cutwater” and is meridionally removed from the impeller exit by Gap B. (For radial-discharge impellers, as in Figure 20, this is a radial gap, and the base circle has radius r_3 .) Beginning at the tongue, the cross-sectional area A , of the volute passage is zero, but it increases with angle θ in the direction of rotation, ending up at area A_T in the “throat” T , as depicted in Figure 20. Worster demonstrated that the desired peripheral uniformity of static pressure can be achieved if the product rV_θ is constant everywhere in the volute²⁵. One-dimensionally, this means that $r_T V_{\theta,T} = r_2 V_{\theta,2}$; and, if the velocity V_T is essentially tangential (in the θ -direction), $r_T V_T \approx r_2 V_{\theta,2}$. The diffusion or reduction of the velocity V from the impeller periphery at r_2 to the larger r_T of the throat produces a static pressure increase above that at the impeller exit; however, friction losses in the volute would cause a reduction in static pressure around the impeller at r_2 from tongue to throat unless the throat area $A_T = Q/V_T$ is slightly enlarged, creating a little more diffusion to compensate for this loss. Thus, in practice, at the BEP,

$$r_T V_T \approx (0.9 \text{ to } 0.95) \times r_2 V_{\theta,2} \quad (61)$$

At off-BEP conditions, the volute will be either too large or too small and Eq. 61 will not be satisfied. When the flow coefficient (or Q/N) drops below the BEP value, there will be excessive diffusion and an increase of static pressure around the volute from the zero area point around to the maximum area point at the throat. Proceeding around further, past the throat, a sudden drop in pressure occurs across the tongue to bring the pressure back to what it was at the starting point²⁵. The opposite situation occurs above BEP.

Each of these off-BEP circumferential static pressure distributions is properly viewed as the consequence of a mismatch between the head-versus-flow characteristics of the impeller and volute²⁶. For the impeller, there is the falling, straight, H_i -versus- Q line or “impeller line” of Figure 6, whereas the volute characteristic or “casing line” would be a straight line starting at the origin of Figure 6 and crossing the impeller line at the match point, which is generally at or close to the BEP flow rate. This casing line is straight because the throat velocity V_T varies directly with flow rate Q and, through Eq. 61, directly with the ideal head H_i —because $\Omega \times r_2 V_{\theta,2} = H_i$ (Eq. 15b for $V_{\theta,1} = 0$). In other words, the same volute could be optimum at a different value of Q if it were paired with another impeller whose H_i -versus- Q line crossed this same casing line at that different Q .

To essentially eliminate the consequent radial thrust on the impellers of large pumps at off-BEP conditions, a double volute is used; that is, there are two throats, 180° apart, there being either two discharge ports or a connecting “back channel” to carry the fluid from one of the throats around to join the flow emerging from the other—to form a single discharge port.

The value of the volute cross-sectional area A_v at a given polar position θ_v can be found for the portion of the total pump flow rate Q being carried in the volute at that θ_v -position together with the condition $rV_\theta = \text{constant}$ versus radius. This will produce a distribution $A_v(\theta)$ that is slightly below a straight-line variation versus θ_v from zero to A_T . Often, the practice is to use the latter straight-line design because this produces larger values of A_v .

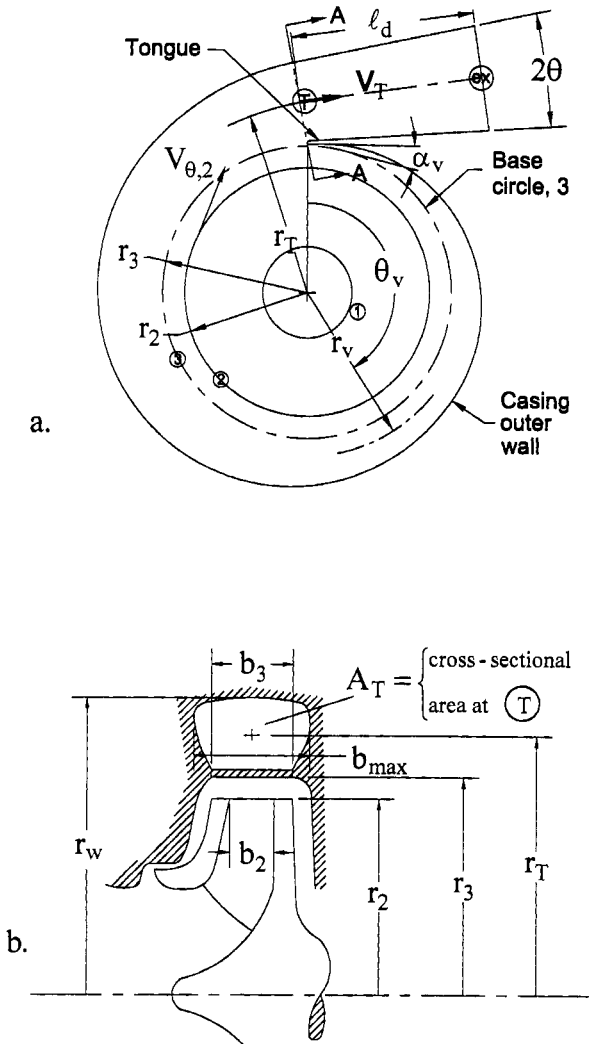


FIGURE 20 Volute casing: a) polar view; b) meridional view including Section A-A of throat T

where the hydraulic radius of the volute is small, thus compensating for the greater friction loss in that region through lower velocity—particularly for the smaller pump sizes. The cross-sectional shape of the volute is dictated by the need to make a minimum-loss transition from a small area at the beginning of the volute where the height (as can be deduced from Figure 20b) is much smaller than the width b_3 to the throat, for which the height (to the outer casing wall at r_w) and the width b_{max} are more nearly equal. Too small an aspect ratio (height/width) decreases the hydraulic diameter too much and increases the loss. There is another transition from the throat through an essentially conical diffuser (which may negotiate a turn) to a larger, circular exit port. This diffuser can be designed with the help of charts of flow elements and will normally have a 7° angle of divergence and a discharge area up to twice that of the throat A_T ²⁷. Thus, there is a substantial diffusion from

the impeller periphery to the pump or stage exit port. This generally produces a static pressure rise in the collection system that is 20 to 25% of that of the whole stage.

b) Vaned diffusers. A vaned diffuser is rotationally symmetric and, if properly applied, produces minimal radial thrust over the whole flow rate range of a pump. Although diffusion can be accomplished in a radial outflow configuration without vanes due to the essential constancy of the angular momentum per unit mass rV_θ , one rarely finds a pump with a vaneless diffuser, partly because so much radial distance is needed to effect the reduction of tangential velocity required, as well as the still larger volute needed on single-stage pumps to collect the fluid at the exit of such a diffuser. Also, the absolute flow angle α_2 (Figure 15) of the fluid leaving the impeller is usually too small to satisfy the conditions for stall-free flow in a vaneless diffuser²⁸. A vaned diffuser, on the other hand, can accomplish the reduction of velocity in a shorter radial distance. Also it can diffuse axially and, to a degree, even with radially inward flow.

Vaned diffusers are similar to multiple volutes in concept, except they are subject to off-design flow instabilities if not shaped correctly. Width b_3 is usually slightly greater than b_2 in order to accommodate discrepancies in the axial positions of impellers that feed them. With reference to Figure 21, "Gap B" ($= r_3 - r_2$) is in effect a short vaneless diffuser, and by the time the fluid has reached the throat (the dashed line at Station 4), it has gained a substantial portion of the static pressure recovery that takes place via diffusion from Station 2 to Station "ex." This "prediffusion" is enhanced by the fact that the throat area at Station 4 ($= b_3 w_1$ per passage for parallel-walled radial-flow diffusers) is larger than it is for volutes, the following relation applying to diffusers²⁹:

$$r_T V_T \cong 0.8 r_2 V_{\theta_2} \quad (62)$$

Therefore, more diffusion than would result from applying Eq. 61 occurs in a vaned diffuser, the skin friction loss due to an otherwise higher velocity at the throat being offset by an efficient reduction of the velocity up to that point and a lower velocity onward.

The fully vaned portion from throat to exit (Figure 21), which performs most of the rest of the diffusion and associated static pressure recovery of the stage, is designed to perform efficiently and maintain stable flow. For typical radial-flow geometries with parallel walls, the vanes can be of constant thickness and comparatively thin or can thicken up to form "islands." The latter approach usually produces a channel that is two-dimensional with straight sides diverging at an included angle, length-to-entrance width ℓ/w_1 , and area ratio A_{ex}/A_T in a combination that avoids appreciable stall³⁰. A typical combination is an included angle of $11\frac{1}{2}$ degrees, $\ell/w_1 = 4$, and $A_{ex}/A_T = 1.8$, which also applies for vanes of constant thickness, as illustrated in Figure 21. Constant thickness vanes have curvature. This modifies the performance somewhat^{31,32}, but it allows a smaller overall radius ratio of the diffuser, r_{ex}/r_3 .

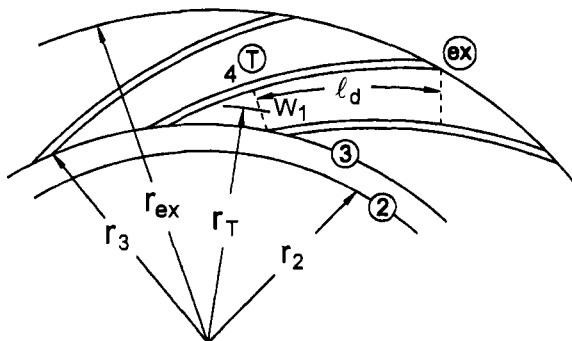


FIGURE 21 Vaned diffuser

Also, this ratio r_{n_3}/r_3 will be smaller as the number of vanes n_v increases. The best experience seems to be with diffusers that have only a few more vanes than the number of impeller blades n_b (n_b rarely exceeds 7 in traditional commercial pumps). For pumps of higher energy levels (or high head per stage, as discussed further on in connection with high-energy pumps), it is important that n_v be chosen so as to avoid a difference of 0 or 1 between n_b and n_v or their multiples—up to at least the third multiple or “order” of each. A difference of 2 should also be avoided for at least the lower orders³³.

At off-design flow-coefficients (or off-design flow rate at a constant speed), the angle α of the absolute velocity vector V (Figure 15) approaching the diffuser will vary; yet, for typical stages, a wide range of flow coefficient is possible without damaging instabilities, even at high energy levels. This is likely the case because α is rather small at the design point or BEP (except for designs having high specific speed), so variations of the angle that occur with flow changes are within the unstalled performance range of the diffuser vane system.

c) Return passages. Conducting the relatively low-velocity fluid from the diffuser to the eye of the next impeller in a multistage pump is accomplished with return vanes or passages that also deswirl the fluid wholly or partially. Except for development of stall in the diffuser, these passages will not see a changing angle of the approaching velocity vector because the diffuser feeding them is also a stationary element. In radial-flow pumps, there is a sharp turn in the meridional plane in order to redirect the fluid radially inward. The fluid, still possessing a circumferential component of velocity that is greater than the meridional component, actually sees a much gentler turn. However, downstream of this point, a sharp turn of the blades is invariably a feature of a return passage; and this, together with the need to ensure undistorted flow into the following impeller, often dictates that the vane system accelerate the fluid as it approaches the eye. Although losses in the return passages—being related to the low velocity within them—have a minor effect on the overall stage efficiency, the design of such passages must ensure unstalled flow into the impeller in order to avoid the negative impact of a distorted inlet flow on the efficiency and to promote pulsation-free operation of the impeller.

A variety of return-passage geometries exist, some of which are presented in the literature^{29,34}. The continuous-vane type is integral with the upstream diffuser, thereby eliminating the entry losses into yet another vane system after the diffuser^{3,34}. Improvements in manufacturing technology have made this potentially more efficient approach more viable for radial machinery. The continuous-vane concept is standard practice in the design of mixed-flow “bowl”-type pumps²¹. The diffusing stator vane row that receives the fluid from the impeller of an axial-flow pump—being an axial-flow element itself—possesses the return feature already. Diffusion in axial-flow stators is typically accomplished by a reduction in velocity of about 30 to 40%. The actual value is governed by an acceptable level of the diffusion factor, Eq. 57. (A similar reduction in relative velocity is needed for an axial-flow impeller to generate static pressure, as can be seen from Eq. 21. By comparison, centrifugal impellers, on the rms streamline, usually have W_2 about equal to W_1 —as seen in Figure 17.)

Axial-Flow Pumps The preceding development, though general, is applicable mainly to centrifugal and mixed-flow pumps. In that procedure, the impellers have appreciable solidity, and original blade shapes are constructed from the viewpoint of one- or two-dimensional channel flow. The collectors are often volutes or non-axial-flow vane systems. Performance is not known a priori and so must be estimated, as outlined further on. On the other hand, the extensive two-dimensional, experimental, axial-flow cascade data amassed by NACA researchers²³ and others enables the designer to adopt existing airfoil blade shapes and so predict the performance with greater confidence. The procedure for utilizing these shapes and the corresponding experimental results has long been the basis for designing axial-flow compressors for gas-turbine engines and is clearly described by Hill and Peterson³⁵. This approach is widely used, especially for high-specific-speed, low-solidity axial-flow propeller pumps—in designing both rotating and stationary blade rows. Insights for propeller pump design and performance characteristics can be found in Stepanoff.⁶

An exception to this axial-flow pump design approach is the case of inducers. Although they are axial-flow pumps, they have high solidity and are usually designed as channel-flow machines. The design philosophy outlined in the preceding paragraphs is applicable,

except that the blades usually approximate constant- or variable-pitch helices. Performance prediction is generally accomplished via one-dimensional calculations and the correlations described in the following paragraphs and, increasingly, via CFD analysis.

PREDICTING THE PERFORMANCE CURVES

The choices made in the foregoing design procedures can and should be verified analytically, the objectives being first to generate the performance characteristic curves for head and power at constant speed and second to ensure stable behavior of the various systems in which the pump is to be applied. For the first objective, the solution involves analytical or empirical approaches: a) at non-recirculating flow conditions; that is, from flow rates Q somewhat below Q_{BEP} out to the maximum "runout" flow rate, b) at shut-off ($Q = 0$) and low flow, or c) the complete set of curves for a given pump predicted by means of computational fluid dynamics (CFD).

Generating Performance Curves The fluid dynamical limitation on the deceleration of the relative velocity W determines the shape of the head-versus-flow curves. This is inherent in the choice made for the head coefficient ψ in Figure 12, which sizes the impeller and is illustrated in Figure 22. The typical situation of zero (or nearly so) inlet whirl $V_{s,1} = 0$ means that the ideal head coefficient ψ_i equals the most significant ratio of the outlet velocity diagram because from Eqs. 15 and 31 (with $V_{s,1} = 0$):

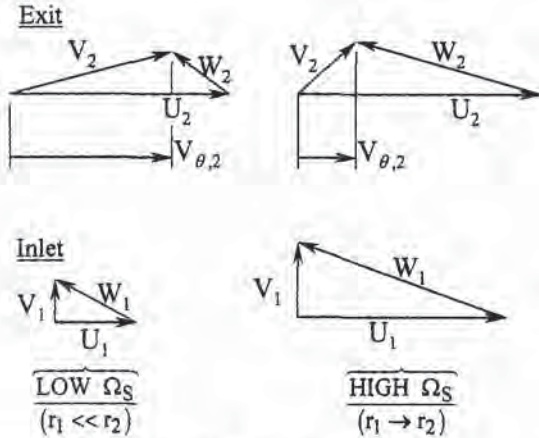
$$\psi_{\text{ideal}} = \frac{U_2^2 (V_{s,2}/U_2)}{U_2^2} \quad (63)$$

$$\psi = \eta_{HY} \psi_{\text{ideal}} = \eta_{HY} \frac{V_{s,2}}{U_2} \quad (64)$$

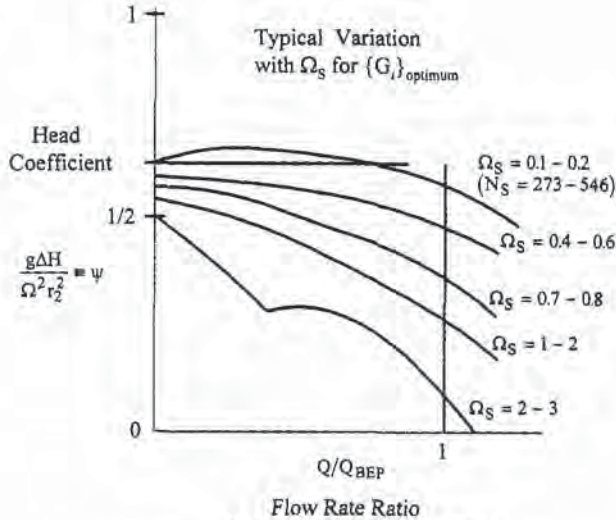
Figure 22 illustrates how specific speed Ω_s affects the BEP value of ψ , and therefore ψ . Overall, only a small reduction of W occurs in most impellers. So, at low Ω_s , the low value of W_1 associated with the small eye relative to the maximum diameter (Figure 9) enables the outlet velocity diagram (Figure 22a) to have a high value of $V_{s,2}/U_2$. On the other hand, this ratio drops as Ω_s increases and the eye grows to be as large as the maximum diameter of the wheel. Figure 22b is the result because the value of ψ at shut-off (about $\frac{1}{2}$) is not based on the one-dimensional concept of velocity diagrams but primarily on the pressure generated by solid body rotation of stalled (though recirculating) fluid contained within the impeller. The BEP values of ψ in Figure 22b are consistent with Figure 12 and illustrate why a high-specific-speed impeller has such a substantial "rise to shut-off" of the head curve. This is dramatically illustrated in Figures 8–10 of Section 2.1.3 in which the head curves are normalized to that of the BEP⁸⁶.

a) Non-recirculating flows. The BEP efficiency and head can be determined from correlations for typical pumps or from computation of the losses. Fluid dynamic procedures described in this section can be used to determine the shapes of the head and power curves at all flow rates to runout, using the BEP as an anchor point for such computations. For pumps designed conventionally, beginning with Figure 12, Anderson's overall (BEP) efficiency correlation (Eq. 44) as modified in Figure 10 is useful. Other similar charts, especially Figure 6 in Section 2.1.3, are in widespread use. The breakdown of the losses involved, as expressed by Eqs. 8–11, is quantified through the development of the three component efficiencies η_{HY} , η_m , and η_v in Table 3. All three decrease with decreasing specific speed—as might be expected from the charts just mentioned.

This can be seen in the η_{HY} -expression (a) of Table 3 because ψ_i is greater at low Ω_s as discussed relative to Figure 22. Jekat's η_{HY} -expression (b) of the table works surprisingly well, largely because of the flow effect in Figure 10 (explained there as the "size effect" of



a) Velocity diagrams at BEP



b) Head-versus-flow curves.

FIGURE 22 Performance versus specific speed: a) velocity diagrams at BEP, b) head-versus flow curves.

larger relative roughness and clearances in smaller pumps) and because low Ω_s tends to go hand-in-hand with low flow rate Q .

To compute η_{HY} at $Q \neq Q_{BEP}$ (and, if required, at $Q = Q_{BEP}$ as well), it is necessary to go deeper into the prediction of η_{HY} by developing expressions for the losses noted in Eq. 21, which are basically expansions of the expression for the collector loss coefficient ζ_c ²⁶ and for the impeller loss expression (c) of Table 3³⁷. In this expression, the incidence loss coefficient k can be obtained from cascade data or developed as a combination of a turning

and a sudden expansion loss^{b,9,27}. The "pipe-type friction factor" f can be increased to include secondary flow and diffusion losses due to blade loading (or turning³⁸ of the absolute velocity vector V). The resulting f -value can thus be twice the usual pipe value associated with the skin friction losses in the passage. (The pipe value of f is found from the well-known pipe friction chart—Figure 31 in Section 11.1—by substituting a

TABLE 3 Component efficiency expressions developed

A) Hydraulic Efficiency

Breaking up the main flow losses into impeller (including inlet)

and collector losses $\sum_i L_i = L_{\text{imp}} + L_c$ and defining $L_c = \zeta_c \frac{V_2^2}{2}$, where ζ_c

is the collector loss coefficient or fraction of $V_2^2/2$ not converted into static pressure rise in the collector, we have (Eq. 10)

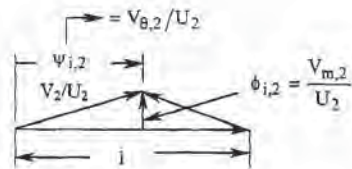
$$\eta_{HY} = 1 - \frac{\sum L_i}{\Delta(UV_\theta)} = 1 - \underbrace{\frac{L_{\text{imp}}}{\Delta(UV_\theta)}}_{\eta_I} - \underbrace{\frac{\zeta_c V_2^2/2}{\Delta(UV_\theta)}}_{\zeta_c \left(\frac{\psi_{i,2}}{2} + \frac{\phi_{i,2}^2}{2\psi_{i,2}} \right)}$$

or, for no inlet whirl ($V_{\theta,1}=0$) $\psi_{i,2} = \psi_{i,1} : 1231$

$$\eta_{HY} = \eta_I - \zeta_c \left(\frac{\psi_{i,2}}{2} + \frac{\phi_{i,2}^2}{2\psi_{i,2}} \right) \quad (a)$$

Impeller hydraulic efficiency: $(\eta_I \approx 0.90 - 0.95)$

Collector loss: $(\psi_{i,2}/2 \text{ predominates})$
 $(\zeta_c \approx 0.2 - 0.3)$



Practical experience (roughness ϵ constant for all sizes) yields the formula by W.K. Jekat:

$$\eta_{HY} \approx 1 - 0.8/[Q(\text{gpm})]^{0.25} \quad (b)$$

$$\text{or} \quad \eta_{HY} \approx 1 - 0.071/[Q(\text{m}^3/\text{s})]^{0.25}$$

at all specific speeds.

However a deeper analysis treats losses L_{imp} as follows:

$$L_{\text{imp}} = k \frac{W_1^2}{2} + f \frac{\overline{W^2}}{2} \frac{\ell}{D_h} \quad (c)$$

where D_h = passage hydraulic diameter,

ℓ = passage length

f = pipe-type friction factor

k = incidence loss coefficient

$\overline{W^2}$ = the average square of the relative velocity within the passage along the rms or mean meridional streamline

W_1 = relative velocity just upstream of the impeller blades on the rms streamline

TABLE 3 Component efficiency expressions developed (*Continued*)**B) Mechanical Efficiency**

From Eq. 9,

$$\eta_m = \frac{P_s - (P_{DF} + P_{B\&S})}{P_s} = 1 - \frac{P_{DF} + P_{B\&S}}{P_s} \quad (d)$$

$$P_{DF} = \frac{C_m}{2} \rho \Omega^3 r_2^5 \quad (e)$$

Bearing &
Seals**Disk Friction**

Where (for two sides of disk) $C_m \approx 0.085 \left/ \left(\frac{\Omega r_{t,2}^2}{\nu} \right)^{0.2} \right.$ Typically $C_m = 0.004$
(at $Re = 4.3 \times 10^6$) (f)

$\uparrow$
 Re

Shaft Power

$$P_s = \frac{\rho g \Delta H}{\eta_p} = \frac{\psi Q_s \rho \Omega^3 r_{t,2}^5}{\eta_p}$$

$$Q_s = \frac{Q}{\Omega r_{t,2}^5} = \Omega_s^2 \psi^{3/2} \Rightarrow P_s = \frac{\Omega_s^2 \psi^{5/2}}{\eta_p} \rho \Omega^3 r_{t,2}^5 \quad (g)$$

Combining the usually smaller $P_{B\&S}$ with P_{DF} , Eq. (d) (with e, f and g) becomes

$$\eta_m = 1 - \frac{(C_m/2) \times \eta_p}{\Omega_s^2 \psi^{5/2}} \quad (h)$$

C) Volumetric Efficiency

From Eq. 11,

$$\eta_v = \frac{Q}{Q + Q_L} \approx \frac{1}{1 + 5 \left[\frac{\frac{\delta}{r_R} \left(\frac{r_e}{r_2} \right)^2}{\Omega_s^2 \psi} \right]} \quad (j)$$

The clearance ratio δ/r_R is typically 0.001 to 0.002

[Leakage back to impeller inlet across two rings (front and back) essentially doubles the leakage loss, changing the value 5 in Equation (j) to 10.]

representative average passage hydraulic diameter $D_h = 4A_p/\pi$ for the pipe diameter d .) A further increase in this f -value occurs if the impeller is missing one or both rotating shrouds; that is, it is a semi- or fully-open impeller with blade tip leakage losses appearing in the main flow stream³⁹. Multiphase flows in pumps often are accompanied by greater than normal hydraulic losses; for example, increasing the concentration of solids in the carrier liquid flowing through a slurry pump increases the f -value still further⁴⁰ (see Section 4.2).

Quasi three-dimensional (Q3D) analysis⁴¹ affords an assessment of the secondary flow and diffusion losses and gives results similar to inviscid three-dimensional (3D) flow analysis. Q3D analysis starts by solving the 2D meridional (hub-to-shroud) flow field. This is followed by a series of 2D blade-to-blade inviscid solutions⁴² or approximations thereto²², each on a surface of revolution generated by one of the meridional streamlines of the hub-to-shroud 2D solution and producing results like that of Figure 17. From this, one computes the diffusion factors (Eqs. 57–59) and decides whether the diffusion losses are significant—in which case a redesign is in order.

CFD Analysis finally enables one to make an accurate and complete hydraulic performance evaluation of the impeller and accompanying stationary flow elements. This can be done at all flow rates and NPSH-conditions for any fluid via a corresponding series of fully viscous, single- or multi-phase solutions of the Reynolds-averaged Navier-Stokes equations (as described in Section 2.1.2).

Mechanical efficiency η_m , as stated earlier, is largely the result of impeller disk friction. If the drag of bearings and seals is added, as in Eq. (d) of Table 3, the moment coefficient C_m in the disk friction formula (e) can be increased over known disk friction values^{43,44} to include these effects. (On the other hand, the drag power loss of shaft seals, though usually quite small, is generally directly proportional to speed. Such losses can therefore be significant in small pumps running at lower-than-normal speeds.) The C_m -expression given in Formula (f) reflects this adjustment and includes the drag on both sides of a smooth impeller at the usually high values of Reynolds number R_e for low-viscosity liquids and for a typical clearance ratio $s/a = 0.05$, where a is the disk radius. This works well for most impellers: The drag at the ring fits roughly compensates for the fact that the impeller eye has been cut out of the disk, and so on. (There is very little influence on C_m of the gap width s between impeller shroud and casing wall, C_m being proportional to $(s/a)^{0.1}$ in general.⁴⁵ For very small s/a , C_m instead grows as s/a decreases. See Refs. 43 and 44 for formulas that apply to the entire range of Reynolds number R_e and clearance ratio s/a .)

The value of C_m can be even larger for semi- or fully-open impellers, if the neighboring fluid is rotating faster relative to the wall—as is the case with radial-bladed open impellers. The fluid between a shrouded impeller and adjacent wall, on the other hand, rotates at half speed⁴⁶. (In cases where the impeller surface and adjacent wall are both rough, C_m is larger than just discussed⁴⁴.) Finally, notice in Eq. (h) that very low specific speed Ω_s produces a dramatically low value of η_m . This drives ψ to the larger values of Figure 12 at low Ω_s —also dictated by the W-deceleration considerations per Figure 22. Overall there is a benefit, despite possibly lower η_{HY} [Eq. (a)] due to the consequently greater ψ_i and collector loss.

Volumetric efficiency η_v applies to leakage across impeller shroud rings or “neck rings” and balancing drums. Eq. (j) in Table 3 is an approximation for the leakage across a typical ring of a closed-impeller pump, assuming orifice-type flow at a discharge coefficient of $\frac{1}{2}$. Results can vary from this approximation of the more exact method reported by Stepanoff.⁵ Referring to Figure 2, leakage Q_L occurs at $r = r_R$, (r_R being approximately 1.2 times r_i under a pressure difference across the ring of about $\frac{2}{3}$ that of the pump stage. If the shroud is removed and the open blades are fitted closely to the adjacent wall, as with open impellers, the consequent leakage from one impeller passage to the next across the blade tips does not affect η_v , and Eq. (j) should be modified accordingly. Rather, the tip leakage causes a hydraulic efficiency loss as previously discussed. Finally, as with η_m , Eq. (j) indicates that low- Ω_s pumps have low η_v .

At flow rates Q other than Q_{BEP} , the analytical methods described previously for computing the hydraulic efficiency are utilized, together with computation of the inlet and outlet velocity diagrams, which yield the ideal head and power curves as illustrated in Figure 6. In this procedure, the slip velocity V_s (Figure 15) applies to the BEP and, at other flow rates, the exit relative flow angle β_{r2} can be assumed constant. This accords with the fact that V_s for the narrower active jet at low flow rates must be smaller. A blockage model for the thickening wakes and narrower active jets that develop as Q is decreased can be introduced to compute the one-dimensional velocity diagrams, but ignoring this at non-recirculating flow rates appears not to be serious in determining the shapes of the head and power curves.

b) Shut-off and low flow. The foregoing analyses apply over that portion of the flow rate range that does not involve recirculation, as illustrated in Figure 6. The complexity of recirculation had not been readily handled analytically prior to the use of CFD, and this has forced pump designers to estimate the low-flow end of the H - Q curve with the help of empirical correlations. Insightful fluid dynamical reasoning about the physics of the flow led to useful expressions for the head developed and the power consumed at shut-off. Shut-off, then, in addition to the BEP, becomes the other anchor point of the head and power curves; and this—together with the shapes established for these curves at the higher flow rates—gives the analyst an idea of the intervening shapes.

Shut-off head $H_{s/o}$ can be viewed as the sum of two effects occurring at $Q = 0$, each being represented by a term in this equation:

$$H_{s/o} = \frac{k_{imp} \times (U_{t,2}^2 - U_{h,1}^2)}{2g} + \frac{k_{ex} \times U_2^2}{2g} \quad (65)$$

or

$$\psi_{s/o} = \frac{k_{imp} \times \left(\frac{r_{t,2}^2}{r_2^2} - \frac{r_{h,1}^2}{r_2^2} \right)}{2} - \frac{k_{ex}}{2} \quad (66)$$

where the first term is the centrifugal effect of essentially solid body rotation of the fluid confined within the impeller; and the second term is the pitot effect of the recirculating fluid from the impeller that impinges against the volute or diffuser throats which in turn are connected through stagnant fluid to the exit port of the pump. While the factors k_{imp} and k_{ex} associated with these effects vary with the hydraulic configuration, the values involved can be estimated as follows: $k_{imp} \approx 1$, as the radial equation of motion³ would indicate for fluid rotating at Ωr everywhere within the blades, i.e., for $r_{h,1} < r < r_{t,2}$ (Figure 8). Thus, as indicated by Eqs. 65 and 66, increasing the minimum radius of the blades at inlet $r_{h,1}$ tends to reduce the shut-off head. However, the presence or absence of fluid swirl in the region upstream of the impeller blades at shut-off has been found experimentally to affect the value of k_{imp} in surprising ways—sometimes increasing it above unity in such a way as to minimize the effect on shut-off head of any non-zero value of $r_{h,1}$. The value of k_{ex} depends on $(r_3 - r_2)/r_2$, or “Gap B” and other features of the impeller exit and collector geometry. It is usually in the range 0.2 ± 0.1 , any change in the geometry that increases the shut-off power coefficient (see below) raising k_{ex} by driving more recirculating flow from the impeller against the volute or diffuser throats. Thus the shut-off head coefficient $\psi_{s/o}$ (Eq. 66) for typical radial-flow pumps generally exceeds $\frac{1}{2}$, the value of 0.585 being advanced by Stepanoff⁵. Estimates for $\psi_{s/o}$ are also indicated in Figure 22b.

Shut-off power consumption $P_{s/o}$ includes disk, bearing, and seal drag power P_D and that which drives the recirculation P_{recirc} . The latter is generally dominant by far. From similarity arguments (Eq. 33), the shut-off power coefficient

$$\hat{P}_{s/o} = \frac{P_{s/o}}{\rho \Omega^3 r_2^5} \quad (67)$$

is a constant for a given pump geometry. Mockridge, in a discussion attached to an ASME paper by Stepanoff, reasoned that a wider impeller (larger b_2 at the same diameter D_2) would recirculate more fluid at shut-off and therefore have a higher value of this coefficient. His correlation is shown in Figure 23 and is probably the most significant quantitative result available for predicting the performance of centrifugal pumps at shut-off conditions short of implementing a full CFD solution of the impeller and accompanying stationary flow elements³⁶.

The design task therefore resolves itself into an iteration between an efficient geometry-generating scheme and a rapid CFD flow and performance analysis of the geometry resulting from each iteration⁴⁵. This is especially useful if a nontraditional geometry is

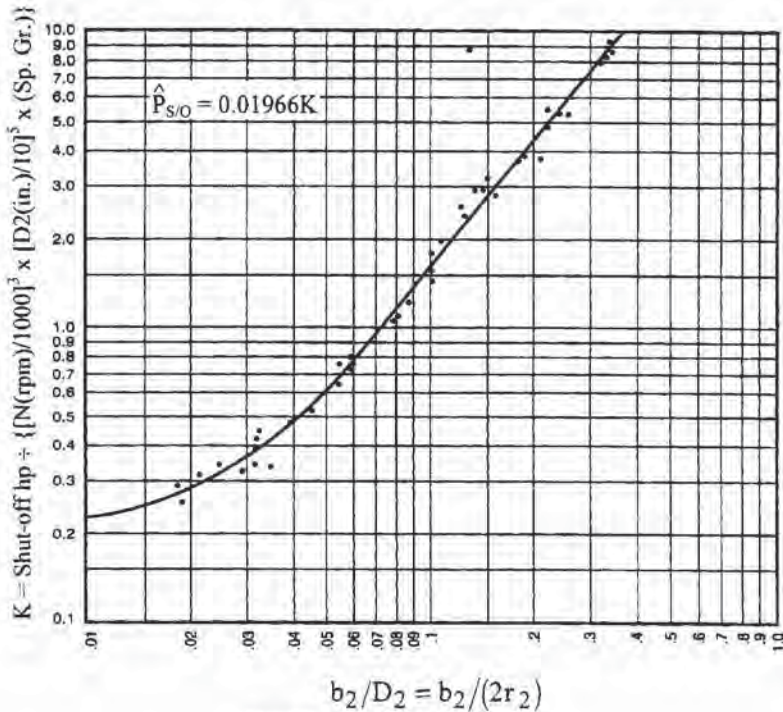


FIGURE 23 Shut-off power coefficient

involved, or if an efficient design is sought that will produce a desired performance curve shape. Nevertheless, many turbomachinery designers can make more rapid and valid judgments about their respective classes of machines through the time-honored iteration between a proprietary direct or inverse design and a convenient inviscid performance prediction scheme.^{41,46} "Inverse" means that the blade shape emerges from an initial specification of certain desired fluid velocity component distributions.¹⁷ This concept has been expanded through more formal geometry optimization employing neural networks.⁴⁷ Thus, designers have developed reliable diffusion criteria (computed, for example, from Eqs. 59a and 59b) for interpreting the acceptability of the free-stream relative velocity distributions W_s and W_p on the blade surfaces (Figure 17) produced by the Q3D blade-to-blade solutions⁴². Because CFD codes solve the actual viscous flow field, the boundary condition on the blade surface is zero relative velocity. This can be at least partly overcome by displaying the CFD-distributions of pressures on the blade surfaces, the interpretation of which would require knowledge of the corresponding criteria for these pressures⁴⁸. Also, the velocities at the edge of the boundary layer could be extracted from the CFD solution and displayed in familiar terms. A useful design approach for the present may therefore be to a) produce the final design by the more traditional methods and b) predict the performance curves via CFD⁴⁹.

Predicting Axial Thrust The prediction of pump performance is not truly complete without the corresponding prediction of the hydrodynamic axial and radial thrust that the impeller(s) can be expected to encounter. A comprehensive treatment of radial thrust appears in Section 2.1.3, and a review of axial thrust and thrust balancing devices is covered in Section 2.2.1. However, obscure flow phenomena can profoundly affect the radial

distributions of pressure on the outside surfaces of a shrouded impeller that give rise to the net axial thrust. These phenomena become even more complex when discharge recirculation occurs and can cause adverse mechanical response in high-energy pumps, as will be explained further on. As a basis for tackling such problems, the fundamentals of axial thrust are presented in Table 4 for shrouded centrifugal impellers that have leaking fluid flowing in the gaps between the impeller shrouds and the adjacent casing walls. The positive direction of the thrust T is taken toward the suction or eye of the single-suction impeller shown. The incoming axial momentum $\rho Q V_{s,1}$ is generally quite small for radial impellers and has been omitted from the Table. It serves, however, to reduce T .

The centrifugal effect of the fluid spinning in the sidewall gaps causes a reduction in static pressure from the outer periphery (OD) of the impeller to the sealing ring, and this is quantified in the expressions given for the swirl velocity component V_θ . These

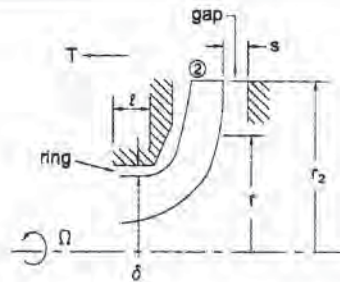
TABLE 4 Leakage effects on axial thrusts

$$\text{Thrust} = (T_B - T_F) n_{\text{stgs.}} - T_{s,lv}$$

$$\text{where } T = \int_{r_{\min}}^{r_{\max}} 2\pi r p dr$$

$$\text{where } p = p_2 - \int^2 \frac{\rho}{g} \frac{V_{\theta, \text{gap}}^2}{r} dr$$

and p and $V_{\theta, \text{gap}}$ are determined as follows in the absence of vanes in the gap:



A) For zero leakage flow, fluid outside impeller rotates at

half speed; i.e., at $\frac{\Omega r}{2}$; namely, $V_{\theta, \text{gap}} = \frac{\Omega r}{2}$; so that

$$p_o - p_i = \int_1^o \rho \frac{V_{\theta, \text{gap}}^2}{r} d\tau = \frac{\rho \Omega^2}{4} \int_1^o r d\tau = \rho \frac{\Omega^2 (\tau_o^2 - \tau_i^2)}{8}$$

or

$$p_o - p_i = \frac{\rho}{8} \Omega^2 \tau_o^2 \left[1 - \left(\frac{r_i}{r_o} \right)^2 \right]$$

$$\left[\begin{array}{l} \text{Note: This is } 1/4 \text{ of what } (p_o - p_i) \text{ would be if} \\ V_{\theta, \text{gap}} = \Omega r \text{ (full wheel speed - as on the front)} \\ \text{side of a semi-open impeller} \\ \text{then, one gets } (p_o - p_i) = \frac{\rho}{2} \Omega^2 \tau_o^2 \left[1 - \left(\frac{r_i}{r_o} \right)^2 \right] \end{array} \right]$$

B) For non-zero leakage

1) Inflow: $V_{\theta, \text{gap}}$ averages to be > half speed

$\Rightarrow$ greater $(p_o - p_i)$ than at half speed (or zero leakage)

2) Outflow: $V_{\theta, \text{gap}}$ averages to be < half speed

$\Rightarrow$ nearly constant pressure from $(p_o - p_i)$

TABLE 4 Leakage effects on axial thrusts (*Continued*)

Specifically,

$$\text{where } V_{\theta, \text{gap}} = \begin{cases} \Omega r_2 \times f_{\text{inflow}} \left(\frac{r}{r_2}, \Phi_{\text{lkg}} \right) \times f_{\text{width}} \\ \text{or} \\ \Omega r \times f_{\text{outflow}} (\Phi_{\text{lkg}}) \times f_{\text{width}} \end{cases}$$

$$\text{where } \Phi_{\text{lkg}} = \frac{Q_{\text{leakage}}}{\Omega r_2^3} \Rightarrow \begin{cases} \text{Interaction of} \\ \text{leakage and} \\ \text{pressure at ring} \end{cases}$$

$$\text{where } Q_{\text{leakage}} = f_L(\ell, \delta, \Omega, v, \Delta p_{\text{ring}})$$

$$f_{\text{inflow}} = \left(0.5 - \frac{450}{\pi} |\Phi_{\text{lkg}}| \right) \frac{r}{r_2} + \frac{450}{\pi} |\Phi_{\text{lkg}}|$$

$$f_{\text{outflow}} = 0.5 - 100 |\Phi_{\text{lkg}}|$$

$$f_{\text{width}} = \frac{\sqrt{A^2 + A} - A}{0.5}$$

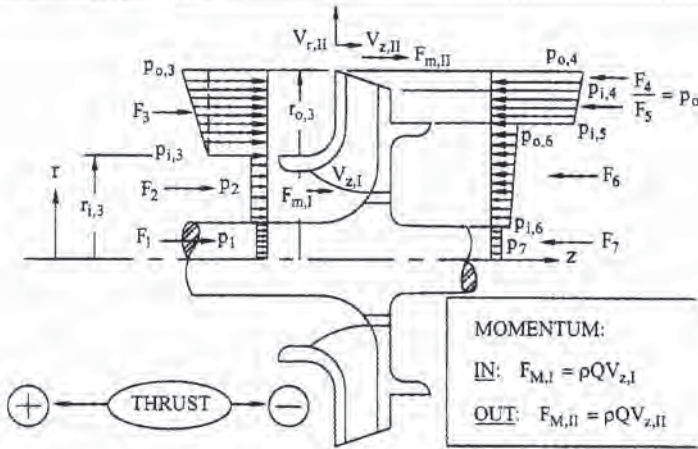
$$\text{where } A = 1/[5 s/r_2]$$

expressions are curve fits to experimental data for the leakage flowing radially inward on either or both sides of the impeller⁵⁰ and for outflowing leakage as occurs on the back side (away from the “front” or suction side) of multistage pump impellers due to the higher pressure arising from the diffusing system downstream of each impeller⁵¹. In the absence of leakage, the fluid in the sidewall gap rotates at about half the local impeller wheel speed; that is, $V_g = \Omega r/2$, and this half speed is typical of the gap flow near the impeller OD, even in the presence of leakage. The greater the inflow leakage, the lower the pressure becomes at the entrance to the ring clearance. The major effect is that of swirl or the tangential component of velocity V_g , which varies inversely with radius; although casing wall drag tends to slow it toward half speed. More leakage flow is less influenced by this drag and so experiences a greater centrifugal effect. This in turn means more pressure drop from OD to ring. (The leakage rate, of course, is affected, the solutions for both leakage and pressure distribution being linked and usually requiring iteration.) The opposite effect happens for outflow on the back side of multistage pump impellers. Here the fluid enters the sidewall gap at a small radius (see Section 2.2.1) and so with negligible swirl. It flows outward without picking up much swirl, especially if there is substantial radial outflow leakage, which means the centrifugal effect is small, yielding a nearly constant pressure versus radius. The overall result is more net thrust toward suction than might be expected from a cursory look at the pressure-loaded surfaces.

If wear ring clearances increase during the life of the pump, the net thrust of multistage pump impellers increases. Likewise, unequal ring wear leads to uncertain changes in the thrust of a “balanced” single- or double-suction impeller with inflow to wear rings on both sides. Similarly, these theories can be applied to balancing drums and other such devices described in Section 2.2.1.

Integration of the pressure equation in Table 4 becomes a chore unless the whole theory is computerized. A quick estimate of the thrust is possible, however, if the distributions of pressure in the separable domains of the surfaces are assumed to be linear; in that case, the integration is simple and yields the closed-form results of Table 5. This table also

TABLE 5 Approximate axial thrust calculation



$$\text{THRUST} = F_7 + F_6 + F_5 + F_4 - F_3 - F_2 - F_1 - F_{M,I} + F_{M,II}$$

$$\text{For Const. - } p \text{ Forces: } F_j = p_j \times \pi (r_{j,o}^2 - r_{j,i}^2) \quad j = 1, \dots, 7$$

$i = \text{inner}; o = \text{outer}$

$$\text{For Linear } p(r) \text{ Forces: } F_j = F_{j,\text{const.}} + F_{j,\text{var}}$$

$$\text{where } F_{j,\text{var}} = \frac{1}{3} \pi r_{j,o}^2 \left[2 - \frac{r_{j,i}}{r_{j,o}} - \left(\frac{r_{j,i}}{r_{j,o}} \right)^2 \right] \times (p_{j,o} - p_{j,i})$$

Example:

$$F_3 = p_{i,3} \times \pi (r_{o,3}^2 - r_{i,3}^2) + \frac{1}{3} \pi r_{o,3}^2 \left[2 - \frac{r_{i,3}}{r_{o,3}} - \left(\frac{r_{i,3}}{r_{o,3}} \right)^2 \right] \times (p_{o,3} - p_{i,3})$$

indicates how to account for each element of the thrust, including the axial momentum terms, which become significant for higher-specific-speed mixed-flow impellers. In all cases, in order to proceed with the calculation, the static pressure at the impeller OD must be known, as it is from the boundary condition imposed at the OD that the rest of the pressure distribution emerges. Even for substantial leakage, the pressure drop of the fluid entering the sidewall gaps from the impeller exit is small or negligible; therefore, the impeller pressure essentially applies in the gap (at the OD) as well. The foregoing methods of predicting pump head also yield the impeller OD pressure, which is usually between 75 and 80% of the stage pressure rise above inlet.

Thrust computations can therefore be coupled with the head-curve prediction scheme being employed for the pump, thereby yielding predicted thrust curves together with the predictions of hydraulic performance.

Ensuring Stable Performance The ability for a pump to run smoothly with minimal pressure-rise and flow-rate excursions is dependent on the shape of the pump head-flow

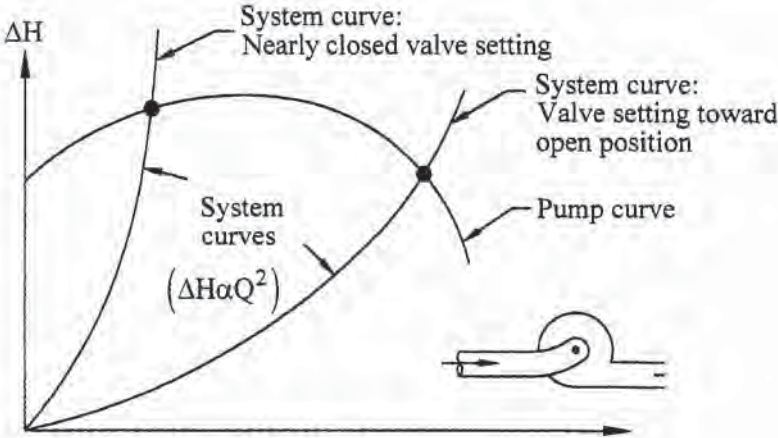
performance curve and the characteristics of the system in which it operates. There are two types of pump-system instability; namely, a) static instability, which can be ascertained by studying the pump and system head curves, and b) dynamic instability, which requires more detailed knowledge of the system⁶². (In addition to these system-related instabilities, there is the unsteady behavior of the separated and recirculating flows that occur when a pump operates at a flow rate substantially below the BEP. Called hydraulic instability, this becomes important in higher-energy applications and is therefore discussed later.)

a) Static stability and instability. Figure 24 illustrates two pumping systems; namely, a) a piping system in which the flow is turbulent and largely independent of Reynolds number; so, the head drop ΔH through it is proportional to Q^2 , and b) two reservoirs with a constant difference ΔH between the two liquid surfaces and comparatively negligible head loss in the pipes connecting them. In each case, the pump is designed to produce head ΔH , as required to deliver the desired flow rate Q . The influence of the pump head curve shape is immediately appreciated in Case (b): the curve “droops” as Q is reduced to shut-off, thereby producing two vastly different flow rates at the same head. In fact, however, the pump will not operate at the lower- Q intersection point of the two curves. The pump shut-off head is less than ΔH , so it will produce no positive flow rate. Instead, as discussed in Section 2.1.3, fluid will flow backwards through the pump. Further, if circumstances could allow operation at this lower- Q point, even a vanishingly small increase of Q would cause a further, divergent increase because the head of the pump exceeds the ΔH of the system. Likewise, a small decrease leads to even lesser Q because the system ΔH exceeds that of the pump. This is called “static instability.” Conversely, the higher- Q point of Figure 24b is “statically stable,” small departures in Q being suppressed by algebraically opposite signs of the difference between the system and pump heads. Both intersection points of Figure 24a are seen by this type of analysis to be statically stable. If the operator increases the frictional resistance by closing up a valve in the piping system, the operating point simply moves to the left on the curve and remains stable. Thus, it is concluded that if the slope of the pump H -versus- Q curve is *less* than that of the system, operation will be *statically stable*—and vice versa.

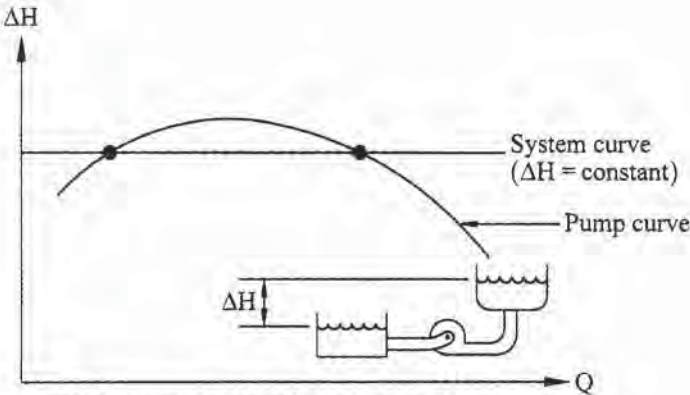
Most pumping systems are combinations of the “pure friction” type of Figure 24a and the “purely elevation” type of Figure 24b. In this case of static stability, the drooping pump head curve presents no problem. Theoretically, it is possible to have a pump head curve with a kink that could have a more positive slope than that of a system curve, which might intersect it at such a kink. The high- Ω , curve of Figure 22b depicts a kink or dip, which is due to stalled flow within a mixed- or axial-flow impeller that is not sufficiently confined by the impeller to be maintained in solid body rotation. (It is the centrifugal effect of such rotation that maintains the pressure rise in radial-flow impellers despite the stalling.) That particular pump, if applied to a system that never intersects the pump head curve at the kink, would never experience static instability. On the other hand, the designer may want to take on the challenge of designing a machine without such a kink. The previously mentioned design procedure utilizing CFD in both the impeller and the diffuser to check whether the kink is gone would be a way of tackling this problem⁶³.

In-depth discussion of the variety of systems that can be encountered, including multiple systems and parallel operation of multiple pumps, can be found in Sections 2.1.3, 11.1, and 11.2. Purely from a static stability standpoint, most of these situations demand a substantially negative slope of the head curve throughout the range of flow rate Q —or what is commonly specified as a pump with head “continuously rising to shut-off” (This “rise in head” versus a drop in Q should not be confused with the pump developed head H , which is also properly termed the “head rise” ΔH , produced by the pump at a given value of Q .)

b) Dynamic instability. If the system has appreciable capacitance, operation may not be stable if the slope of the pump head curve is positive or even zero⁶⁴. This is true even though the slope of the pump head curve is less than that of the system head curve as required above for static stability—as with the lower- Q intersection point of Figure 24a. Dynamic instability can be manifested as pump surge, a phenomenon wherein the flow rate oscillates and can even be alternately positive and negative through the pump⁶⁴. This



a) System with frictional resistance only



b) System with elevation change only

FIGURE 24A and B Pump-system stability

is characteristic of “soft” systems that contain vessels with free surfaces and, therefore, appreciable capacitance. Two-phase flow increases the capacitance of a system and can cause dynamic instability. For example, fluctuating vapor volume within the propellant pump inducers can contribute to the dynamic instability of a rocket propulsion system⁵⁵.

On the other hand, a “hard” piping system with no capacitance is theoretically capable of accommodating a pump that has a flat or drooping head curve and that operates on that flat or drooping section of the curve. Low-specific-speed pumps can have drooping curves (Figure 22b), especially if designed with a high head coefficient ψ at the BEP. Figures 5 and 7 of Section 2.1.3 depict flat and drooping curves of low- Ω , radial-bladed pumps, a type that is widely used in low-cost, small sizes. If properly applied, such machines will operate with stability.

However, to avoid uncertainties, a conservative approach that guarantees both static and dynamic stability for the widest range of applications is to design all pumps without

flat spots or droops in the head curves. In practice, this approach requires that the pump be designed to have at least 15 to 20% more head ΔH at shutoff than at BEP and, therefore, for constant-speed pumps, a value of the shutoff head coefficient $\psi_{s/0}$, which is that much greater than the BEP head coefficient ψ . Thus, it may happen that choosing ψ from Figure 12 to size the impeller conflicts with this "rise-to-shutoff" design requirement, a situation more likely to occur with low-specific-speed pumps. If so, a lower value of ψ than that of Figure 12 would need to be chosen for the BEP.

DESIGN EXAMPLE

To illustrate the application of the preceding design information, the basic hydraulic design requirements are presented in Table 6 for a single-stage centrifugal water pump with a volute collector. The chosen conditions compute to a universal specific speed Ω_s of 1 ($N_s = 2733$, and $n_q = 52.9$ from Eqs. 38a and 38b). The pump is to be designed for a suction specific speed Ω_{ss} of 4.5 ($N_{ss} = S = 12,300$ by Eqs. 41 and 42), therefore, a "performance-NPSH" (see NPSH discussion in Section 2.1.3) or $NPSH_{3\%}$ of 14 ft (4.27 m). As such, this pump is readily applicable to taking water from an atmospheric reservoir with some suction lift. Although acceptable for 1780 rpm specified for this pump, this suction-specific speed Ω_{ss} capability of 4.5 is regarded as high for pumps with more head (energy level) than this one (see the energy-level discussion later).

Impeller Design

SUCTION NOZZLE Beginning upstream, an ideal, axial-flow approach passage is assumed—this is known as an "end-suction" configuration. The suction-approach passage or suction branch (not shown) is simply a conical nozzle that increases the velocity of the fluid from the suction port to the impeller eye by about 50% in an axial distance of about half the diameter of the eye. This helps to ensure the existence of uniform flow at the eye. Too short a nozzle would mean excessive local meridional (axial in this geometry) velocity at the impeller shroud and possible separation.

IMPELLER INLET Beginning with Ω_{ss} of 4.5, computations for this example are carried out in Table 7 for the impeller eye geometry. The choice of the $NPSH_{3\%}$ correlation of Table 1

TABLE 6 Given conditions for design example (single stage, end-suction volute pump)

Speed	= 1780 RPM	
Flow Rate	= 2500 gpm (0.1577 m ³ /s)	
Head	= 104 ft. (31.7 m)	Fluid: Water $\left\{ \begin{array}{l} \text{sp. gr.} = 1 \\ \nu = 1 \text{ cs.} \end{array} \right.$
Specific Speed	$\Omega_s = 1; N_s = \frac{1780 \sqrt{2500}}{(104)^{3/4}} = 2733; n_q = \frac{1780 \sqrt{0.1577}}{(31.7)^{3/4}} = 52.9$	
Suction Specific Speed	$\Omega_{ss} = 4.5 \quad (S = 12,300)$	
NPSH	$= \left\{ \frac{1780}{12,300} \right\}^{4/3} (2500)^{2/3} = 14 \text{ ft. (4.27 m)}$	

TABLE 7 Impeller inlet (design example)

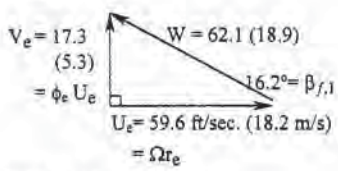
$$\Omega_{ss} = 4.5 = \frac{\sqrt{\pi \phi_e \left(1 - \frac{r_s^2}{r_e^2}\right)}}{(\tau/2)^{3/4}} \quad (\text{from Eq. 50})$$

For $\tau = (1.69 + 0.102) \phi_e^2 + 0.102$
 $\phi_e = 0.29 = \text{eye flow coefficient}$

Eye dia/2 = eye radius:

$$r_e = \sqrt[3]{\frac{Q}{\Omega \pi \phi_e \left(1 - \frac{r_s^2}{r_e^2}\right)}} = \sqrt[3]{\frac{2500 \text{ gpm} / 449 \frac{\text{gpm}}{\text{ft}^3/\text{sec}}}{\frac{\pi}{30} \times 1780 \times \frac{\pi}{\text{sec}} \times 0.29}} \quad (\text{from Eq. 49})$$

$[r_s = 0 \text{ (no shaft in eye)}]$
 $r_e = 0.3201 \text{ ft. (0.0976 m)}$
 $r_e = 3.84 \text{ in. (97.6 mm)}$



(See Fig. 25): $R_{sh} = \frac{r_e - r_s}{2} = 1.92 \text{ in. (48.8 mm)}$

Other dimensional data shown on Fig. 25.

is used, with $k_1 = 1.69$ and $k_2 = 0.102$. The local maximum velocity at the eye $V_{1,sh} (= V_{m,1,sh})$ is assumed to be 25% greater than the one-dimensional average $V_e = Q/A_e$. This is typical of this end-suction configuration. Moreover, the value of k_1 should be more than adequate for this value of $V_{1,sh}$.

The Ω_{ss} and τ - ϕ_e relationships yield the eye flow coefficient ϕ_e , which in turn sizes the eye. Had the NPSH been significantly greater than the 14 ft specified in Table 6, these relationships would have produced a value of ϕ_e that is larger than 0.3, which in the discussion following Eq. 50 was said to be the upper limit of normal design practice. In that event, ϕ_e would have been chosen independently of Ω_{ss} , and the eye radius r_e found immediately from Eq. 49. In this case, $\phi_e = 0.29$ implies a τ -value of 0.253, which is typical. However, lower ϕ_e - and τ -values are common, especially for the case of a shaft through the eye, because this tends to maintain the level of Ω_{ss} in the face of the r_e -effect in Eq. 50. The nominal velocity diagram at the eye—substantially the shroud-end or tip of the blade leading edge—shows V_e rather than $V_{1,sh}$ for the meridional component of velocity and so is not the actual velocity diagram at that location. Rather, this triangle serves to identify the geometry through the basic ratio $\phi_e = V_e/U_e$ —without having to deal with the uncertain choice of $V_{1,sh}/V_e$. Moreover, ϕ_e is the tangent of the tip relative flow angle $\beta_{f,1}$ as it would be for a uniform axial velocity profile in the eye.

With the eye radius r_e established, the local shroud radius of curvature R_{sh} follows from the guidelines associated with Figure 13. The geometry established so far is illustrated in Figure 25. Before the full picture shown there can be established, the outlet must be sized.

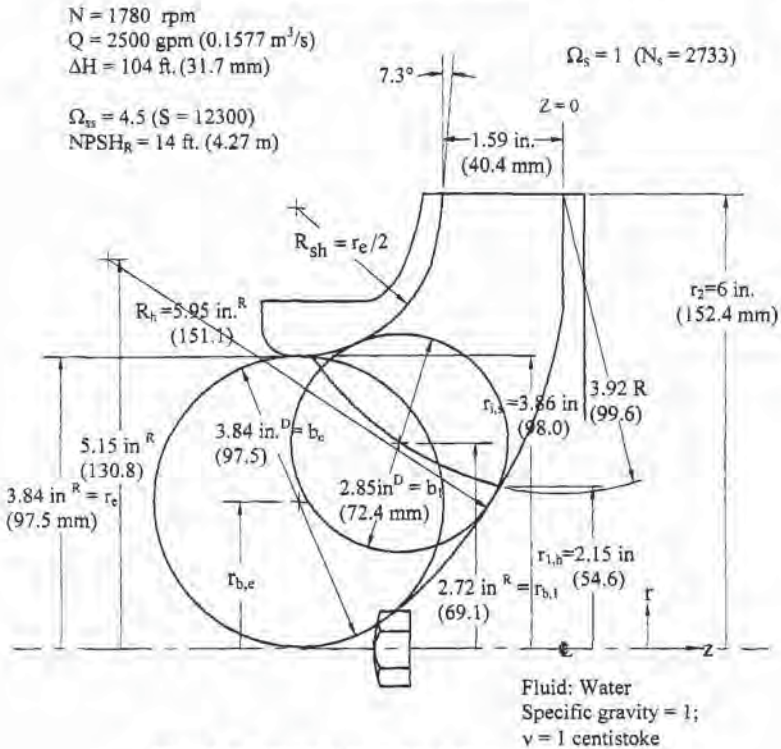


FIGURE 25 Impeller hub and shroud profiles (design example)

IMPELLER OUTLET The computations in Table 8 for the impeller exit begin with the choice of the typical value of $22\frac{1}{2}$ degrees for the outlet blade angle $\beta_{b,2}$. This enables the head coefficient ψ to be chosen under the guidance of the upper curve in Figure 12. The value 0.385 is selected, and this yields the impeller diameter of 12 in. (304.8 mm). The other curve in Figure 12 is for outlet flow coefficient $\phi_{i,2}$, which conveniently equals 0.1715 for $\Omega_s = 1$. This leads to the exit width b_2 after adding in the leakage and the blockage of blades and boundary layers per the computations of Tables 9 and 10.

However, Anderson⁷ points out that what matters for centrifugal pump performance is neither the blade angle nor the exit width individually, but the impeller outlet relative area $2\pi r_2 b_2 \sin \beta_{b,2}$. Choosing a higher blade angle is possible if b_2 is correspondingly reduced (and $\phi_{i,2}$ increased) so as to maintain this area and therefore the relative velocity W . Figure 15 shows that $V_{s,2}$ is thereby essentially unchanged; this in turn preserves the impeller head.

EFFICIENCIES Anderson's overall pump efficiency correlation (Figure 10 and Eq. 44) and the component efficiency expressions of Table 3 lead to the results of Table 9. These give an indication of the relative magnitudes of the losses and are as follows:

$$\text{Overall efficiency } \eta_p = 0.8550$$

$$\text{Mechanical efficiency } \eta_m = 0.9814$$

$$\text{Volumetric efficiency } \eta_v = 0.9833 \text{ (leakage across front and back rings)}$$

$$\text{Hydraulic efficiency } \eta_{HY} = 0.8860$$

TABLE 8 Impeller outlet (design example)

From Figure 12

$$\beta_{b,2} = 22 \frac{1}{2}^\circ \rightarrow \phi_2 = \frac{V_{m,2}}{U_2} = 0.1715 \quad (= \phi_1)$$

$$\rightarrow \psi = \frac{g_0 \Delta H}{U_2^2} = 0.385$$

$$(U_2 = \Omega r_2) = \frac{\pi}{30} \times 1780 \times \frac{6}{12} = 93.2 \frac{\text{ft.}}{\text{sec.}} (28.4 \text{ m/s})$$

$$r_2 = \sqrt{\frac{g \Delta H}{\Omega^2 \psi}} = 12 \sqrt{\frac{32.174 \times 104}{\frac{\pi^2}{30^2} \times (1780)^2 \times 0.385}}$$

$$= 6 \text{ in. (152.4 mm) [12 in. (304.8 mm) dia.]}$$

$$\text{Width: } b_2 : A_2' = \pi D_2 b_2'$$

$$b_2 = b_2' + 2\delta_o^* + \text{Leakage Effect}$$

Solve after obtaining volumetric efficiency and getting approximate passage length (to compute δ_o^*)

η_{HY} is at this point simply deduced from the others, beginning with Anderson's correlation. Although it is confirmed by Jekat's correlation in the table, it can be found in a detailed computation of the hydraulic losses via one-dimensional methods. This will be carried out further on to obtain the performance characteristic curves. Meanwhile, this initial computation enables the determination of $V_{s,2}$ at the end of Table 9, which, along with $V_{m,2}$ from Table 8, is a major element of the outlet velocity diagram of Figure 26.

BLOCKAGE AND WIDTH AT IMPELLER EXIT With the leakage and exit blade angle information, Table 10 contains the computations of the blockage and the exit width b_2 . This entails the choice of the number of blades, the blade thickness t (2% of the impeller diameter and typically assumed to exist at the exit and elsewhere on the blades except near the leading edges where typically half that value is chosen), and the approximate blade length ℓ (assuming the mean-streamline blade angle to be constant at $22\frac{1}{2}$ degrees). The boundary layer blockage is computed from the following approximations:

- Adverse pressure gradients on the blades lead to a boundary layer displacement thickness δ^* of twice the zero-pressure gradient value δ_o^* on each blade surface.
- Secondary flows scrub the boundary layers from the hub and shroud surfaces; so, δ^* is assumed to be equal to δ_o^* on those surfaces.
- $\delta_o^* = 0.0462 \ell^{0.8} \nu^{0.2} / W^{0.2}$ for flat-plate, turbulent flow⁵⁶, and is approximated in this example for low viscosity by a linear growth with length along the blade.

The resulting thickness of the boundary blockage is 0.0732 in (1.86 mm) on the blades, which themselves have a thickness of 0.24 in (6.1 mm). Because these thicknesses are inclined at the $22\frac{1}{2}$ -degrees outlet angle, the actual circumferential blockage is $(1 - e_{g,b}) = (1 - 0.870)$ or 13% of $2\pi r_2$. In particular, $(0.24 + 0.0732)/\sin(22\frac{1}{2} \text{ degrees}) = 0.82$ in or

TABLE 9 Component efficiencies (design example)

$$\text{Efficiency: } \eta_p = 0.94 - 0.08955 \left(\frac{2500}{1780} \right)^{-0.21333} - 0.29 \left[\log_{10} \left(\frac{2286}{2733} \right) \right]^2 = 0.8550 \quad (\text{Eq. 44})$$

$$\text{Mech. Eff'y: } \eta_m = 1 - \frac{0.002 \times 0.8550}{1^2 \times (0.385)^{5/2}} = 0.9814 \quad [P_D = 1.43 \text{ hp. (1.07 kW)}] \quad (\text{from Table 3})$$

$$\text{Volumetric Efficiency: } \eta_v = \frac{1}{1 + 5 \frac{0.0016 \times (3.84/6)^2}{1^2 \times 0.385} \times 2} = 0.9833 \quad (\text{from Table 3})$$

$$\text{Hydraulic Efficiency: } \eta_{HY} = \frac{\eta_p}{\eta_v \eta_m} = \frac{0.8550}{0.9650} = 0.886 \quad \text{vs.} \quad (\text{Eq. 8})$$

(Jekat's formula in Table 3):

$$\eta_{HY} \approx 1 - \frac{0.8}{(2500)^{0.25}} = 0.887$$

$$\Rightarrow V_{B,2} = \frac{32.174 \times 104}{93.2 \times 0.886} = 40.5 \text{ ft/sec.}$$

$$\Rightarrow \psi_i = \frac{40.5}{93.2} = 0.435$$

$(6.1 + 1.86)/\sin(22\frac{1}{2} \text{ degrees}) = 20.8 \text{ mm per blade}$, which for all six blades is 0.130 times the circumference of 37.7 in (958 mm). The width b_2 is computed to maintain $V_{m,2}$ at the chosen value of $0.1715 \times U_2$, while accommodating this blockage and that of the sidewall boundary layers. Altogether, it can be computed from these data that 85% of the meridional exit area $2\pi r_2 b_2$ is estimated to remain open for the one-dimensional flow of $(Q + Q_L)$ at velocity $V_{m,2}$, the boundary layers causing $\frac{1}{5}$ of the blockage. This is quite typical. An openness of 90% is possible for larger impellers.

HUB-SHROUD PROFILES It is now possible, through the guidelines outlined in the discussion of Figure 13, to finish plotting the hub and shroud profiles in Figure 25, which are also seen in Figure 26c. At this point, the envelope of the leading edges of the blades is approximated by a circular arc—later modified somewhat in the construction of the blades. The arithmetic average radius of the meridional passage at the leading edges is found with the circle of diameter 2.85 in (72.4 mm) to be 2.72 in (69.1 mm). The resulting line passing through the center of the circle is normal to both hub and shroud and is approximated by the dashed straight-line leading-edge quasi-normal shown in the Figure 26c.

INLET VELOCITY DIAGRAMS The rms radial point on this quasi-normal line of Figure 26c crosses the leading edges of the blades at $r_{i, \text{mean}} = r_{i, \text{rms}} = 2.91 \text{ in (73.9 mm)}$ and is slightly larger than the arithmetic average radius of 2.72 in (69.1 mm). It is this latter radius that must be used in computing the one-dimensional meridional velocity. Once obtained, it is applicable to the rms location (the location of the “mean” or “rms” inlet velocity diagram).

TABLE 10 Impeller blade blockage and width at exit (design example)

Blade no. $\left\{ \begin{array}{l} n_b: \text{Assume} = 6 \\ \text{Check Solidity:} \end{array} \right.$	$\ell = \frac{\Delta m}{\sin \beta} \approx \frac{3.5 \text{ in.}}{\sin 22 \frac{1}{2}^\circ} = 9.15 \text{ in. (232.4 mm)}$ <p>(Eq. 56)</p> $\sigma = \frac{6 \times 9.15}{\pi \times 12} = 1.46 \rightarrow \text{OK}$ <p>(Eq. 53)</p>
Blade thickness $\left\{ \begin{array}{l} t = 0.04 \text{ in. (6.1 mm)} \end{array} \right.$	

$\epsilon_{2,b} \text{ (blades only)} = \frac{6(0.0732 + 0.240)}{\pi \times 12 \times \sin 22 \frac{1}{2}^\circ} = 0.870$	$\delta_o^* \approx 0.002 \times \ell \approx 0.002 \times 9.15 \text{ in. (232.4 mm)}$
	$2\delta \approx 4 \delta_o^* \approx 0.0732 \text{ in. (1.86 mm) on blades}$
	$2\delta \approx 2 \delta_o^* \approx 0.0366 \text{ in. (0.93 mm) on hub and shroud}$

gpm & ft/sec to inches
 η_v

$$b_2 = 0.0366 + \frac{2500 \times 0.3208 / 0.9840}{\frac{0.1715 \times 93.2}{V_{m,2}} \times \frac{0.870}{\epsilon_{2,b}} \times \pi \times 2 \times 6} = 1.59 \text{ in. (40.4 mm)}$$

(Since

$$b_2 = 2\delta_o^* + \frac{Q + Q_L}{V_{m,2} \times \epsilon_{2,b} \times \pi \times 2 \times r_2})$$

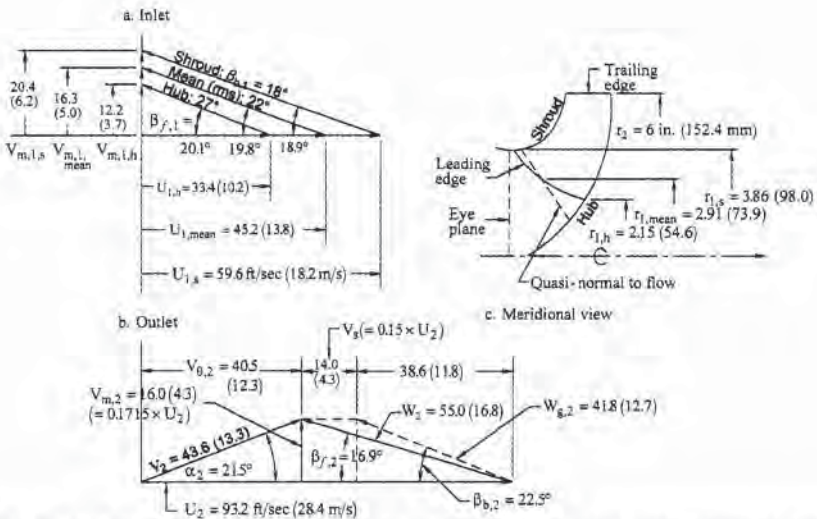


FIGURE 26A through C Impeller velocity diagrams for design example: a) inlet; b) outlet; c) meridional view

This diagram is one of three triangles shown for the inlet in Figure 26a, the other two being located at the hub and shroud locations of the blade leading edges. Notice that $V_{m,1}$ for the rms triangle is 16.3 ft/sec (5.0 m/s), which is slightly less than the eye velocity V_e of 17.3 (5.3). Allowing for blade blockage, this would bring the blocked meridional velocity V_m within the blading closer to V_e , the objective being to keep V_m constant in the inlet region and turn into the radial direction. The other triangles correspond to the radial locations of the blade at hub and shroud, as illustrated in Figures 26c and 27, and assume that $V_{m,1,h} = 1.25$ times $V_{m,1}$, and $V_{m,1,h} = 0.75 V_{m,1}$. A full Q3D or CFD solution would determine these velocities more accurately; however, the design usually proceeds in this way—largely because the hub blade angle is usually a good deal larger than hub flow angle. Efforts to match the hub flow angle more closely entail special blading that is beneficial for high-energy pumps but has little effect otherwise. The blade angle at the shroud is slightly lower than the flow angle (by about 1 degree). This slightly negative incidence (to the mean camber line of the blade) is actually ideal for efficient flow and minimum cavitation. The largest values of U and W exist at the shroud, as can be seen for the shroud inlet triangle, making it important to have the best match at that point. Two or three degrees of positive incidence is quite common at the mean or rms radial location and allows for blockage by the blades that does not increase the relative velocity W as the fluid enters the impeller. (Actually a drop upon entrance to the blading in this mean W of 10% or more is often found in radial-flow impellers. Such blading tends to minimize the sum of impeller incidence and passage friction losses at the BEP flow rate and enables lower NPSH-capability at flow rates above BEP.)

OUTLET VELOCITY DIAGRAM The outlet velocity components having been found in Tables 8 and 9, the slip velocity V_s must still be found in order to obtain the complete outlet velocity diagram shown in Figure 26b. This slip is computed by Pfleiderer's method (Table 2), which utilizes the $r(m)$ shape of the mean meridional streamline illustrated in Figure 27, V_s emerging as 15% of U_2 . The "a" factor for influence of the collector geometry was taken in the middle of the range for volutes at 0.75; (see Table 2). Wiesner's Eq. 52 yields 17.65%. This would mean 6% less $V_{u,2}$ and head. However, this discrepancy is not unexpected, and in view of the earlier discussions on slip, the Pfleiderer result is chosen as more realistic.

Nevertheless, uncertainty in the slip is the Achilles heel of the one-dimensional analysis method. For this reason, most analysts "calibrate" their codes by deducing the slip from test results and applying it to impellers of similar geometry. For Pfleiderer's method, this would be done by calibrating the "a" factor. CFD solutions are now the best approach to overcoming this difficulty.

As has been emphasized heretofore, this outlet velocity diagram contains the basic information about the performance and design of the pump. It supplies the boundary conditions for the volute design, but first the impeller blading that must produce it will be established and evaluated.

IMPELLER BLADING As indicated in Figure 26, the blade angles at both inlet and outlet have been chosen at hub, mean, and shroud (with the same $22\frac{1}{2}$ degrees all across the trailing edge being assumed, although some would specify a little variation). Fitting a reasonable blade shape between these end conditions can be done in the conformal plane (illustrated in Figure 19). These shapes, when transformed as described earlier, yield the hub, mean, and shroud blade shapes identified in the polar view of Figure 28. In actuality, the inverse "point-by-point" method was used, specifying W_r as indicated in Figure 29 and, at each of the 21 stations along the mean line of Figure 27, developing the velocity diagrams in the manner employed to arrive at Figure 26b; obtaining β_i at each station; and developing the *mean-streamline blade shape* of the polar view with Eq. 60. Involved in this procedure—which was computerized—was the calculation at each station of the blockage and of the local slip velocity, the latter being estimated as a fraction of the discharge slip velocity V_s that increases rapidly to unity as the exit is approached.

Note that the meridional area is needed at each station in order to compute V_m . As indicated in Figure 27, this is approximated here as the area of the frustum of a cone defined by the dashed lines or "quasi-normals," which are as nearly perpendicular to both hub and shroud as possible. Except for some machines with meridionally curved passages and a

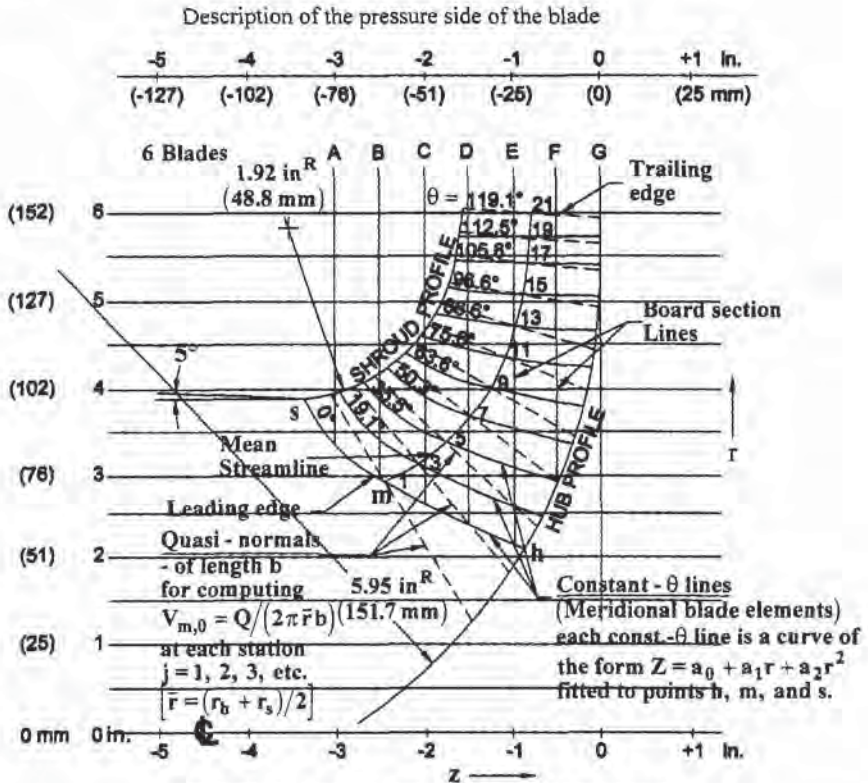


FIGURE 27 Impeller blading—meridional view (design example)

large passage width-to-length ratio causing the true normals to be strongly curved, the quasi-normal approach works well.

To establish the *blade shapes along the hub and shroud*, the corresponding “grid lines” of Figure 19 were found from polynomials satisfying the end conditions of blade angle and location, the y -position of the inlet end of the grid line being iterated until the desired blade wrap was obtained. In this way, different wrap angles $\Delta\theta (= \theta_2 - \theta_1)$ can be imposed, but a tolerable resulting blade shape requires that these $\Delta\theta$ s be not much different from the $\Delta\theta (= 119.1$ degrees) obtained on the mean line from the method just described. In this example, it was possible to maintain $\Delta\theta$ the same on all three construction lines. Alternatively and perhaps more consistently, the inverse approach could be applied at hub and shroud as well as on the mean line.

Regardless of which procedure is used to generate the hub, mean and shroud blade shapes—the conformal transformation method, the point-by-point method, or some combination—the *shape of the blade everywhere else* still needs to be established. To do this, the shapes of the constant- θ lines in Figure 27—called “blade elements”—must be specified. This can be done mathematically as indicated on the figure or by eye (the latter approach has been widely used). Note that each constant- θ line actually lies in a different meridional plane. Thus, Figure 27 depicts a superposition of all 21 meridional planes, each containing an intersection of the blade surface and appropriately identified as having the θ -value noted on the figure. The resulting blade shape is illustrated in the polar plots of Figure 28.

DIFFUSION ASSESSMENT Figure 29 is a presentation of the free-stream relative velocity distributions at the blade surfaces and halfway between. The rapid approximate method of

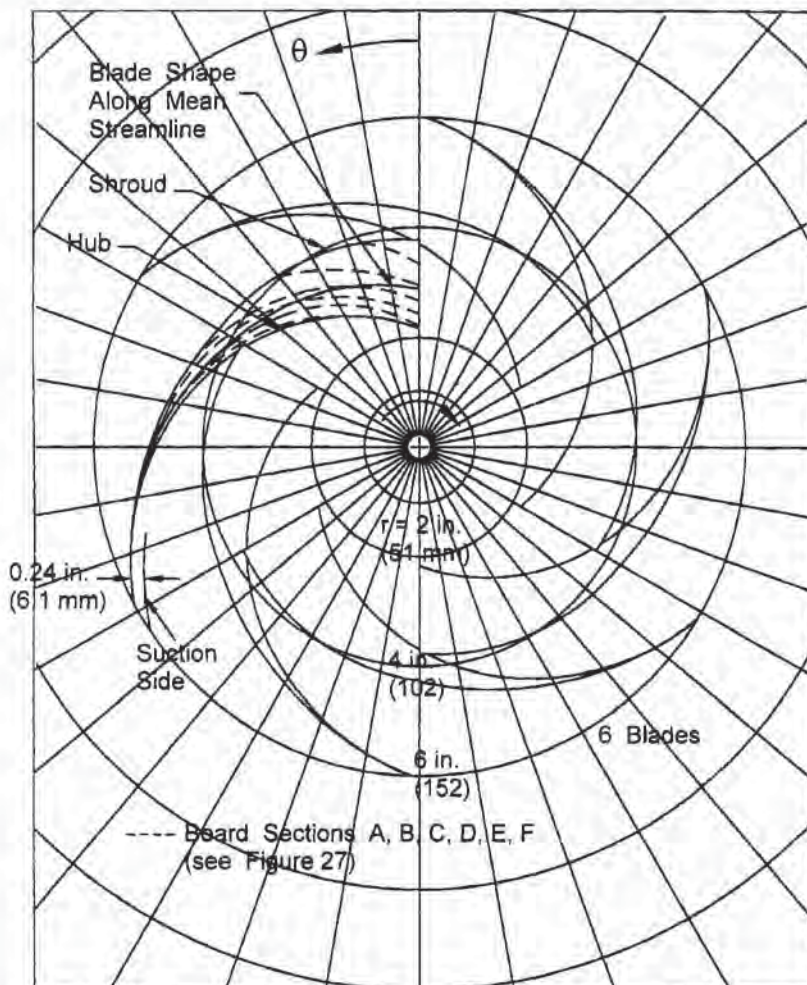


FIGURE 28 Impeller blading—polar view, showing pressure sides of the blades (design example)

Stanitz (Eq. 54) was utilized to obtain the surface W 's, the mean values of W on the mean (rms) streamline coming from the local velocity diagrams developed at each station²². The mean W 's at hub and shroud were found at the ends of the respective constant- θ lines under the assumption that the ideal head or UV_θ -product is constant along each of these lines from hub to shroud. This is usually a fair approximation; however, a Q3D or CFD analysis would yield a closer estimate of these mean W -distributions⁴¹. Nevertheless, the diffusion factors computed from the plotted surface velocities by means of Eqs. 59 are shown on the figure—these values are well below the 0.6 limit. This would indicate that the resulting solidity of 1.48 (originally estimated at 1.46 in Table 10) is adequate, and that the correct number of blades was chosen.

Notice also on Figure 29 that the exit value of W is significantly larger than W_g , which is consistent with the outlet velocity diagram of Figure 26b. Further, a slight jump in the mean W at the inlet above the upstream value $W_{1,mean}$ is indicated to occur one-dimensionally across the leading edge. As mentioned in the discussion following Figure 26,

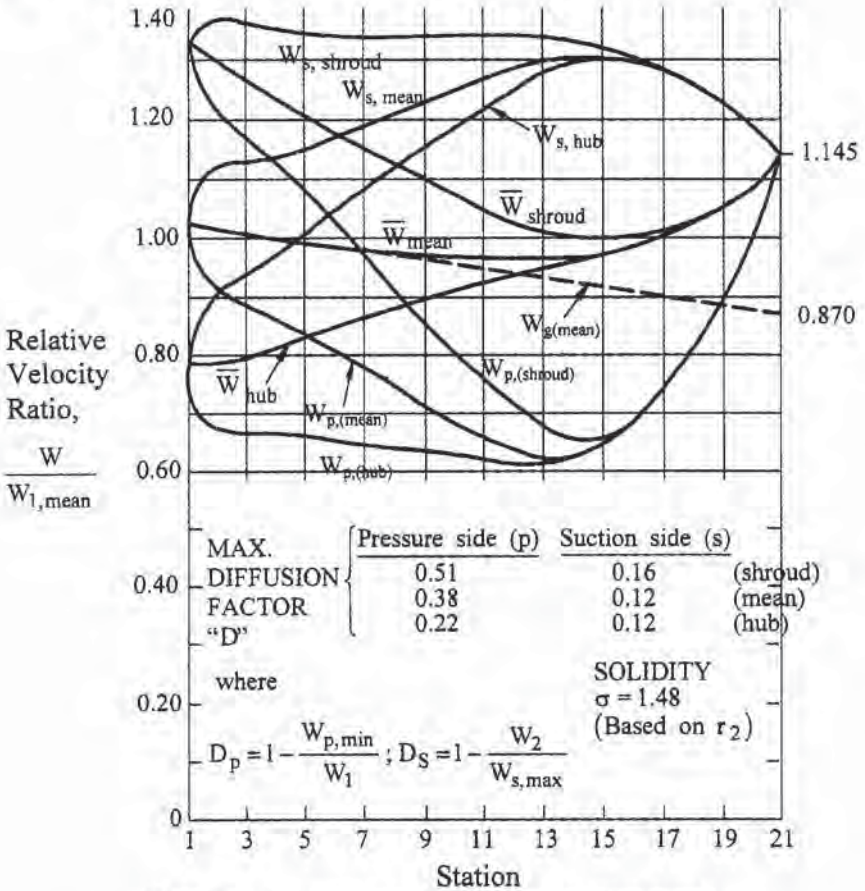


FIGURE 29 Blade surface relative velocity distributions (design example)

the actual practice of many designers is to create a drop of this mean W . However, in this example, as the higher value of W is not sustained, it is unlikely that the jump actually occurs in the 3D velocity field. Moreover, details near leading and trailing edges are not well handled by Eq. 54—a CFD or 2D blade-to-blade solution is more desirable for this type of diffusion assessment⁴². Nevertheless, the design guidelines given here were developed in the light of applicable one-dimensional velocity diagrams, computations, and experimental experience. Thus, if the flow rate for which the curves of Figure 29 were generated (namely the specified design point Q of Table 6) had been 5% lower, the same impeller would have exhibited a small drop in the value of the mean W at Station 1.

Volute Design With the exit velocity V_2 of 43.6 ft/sec (13.3 m/s) and the tangential velocity component $V_{\theta,2}$ of 40.5 ft/sec (12.3 m/s) from Figure 26b, the volute design process is started in Table 11 using Eq. 61. This entails a choice of the radial “Gap B” of 6% of the impeller radius r_2 and a tongue leading edge thickness t_t equal to 70% of the impeller blade thickness. These choices are not critical for a pump of low energy level such as this is, as the stresses imposed by the flow on structural elements are small. The throat area A_T is then found to be 32.60 in² (21,032 mm²) and the throat velocity V_T computes to 24.60 ft/sec (7.50 m/s), which is 56% of V_2 and so represents considerable diffusion from impeller exit

to throat. (The throat area A_T is the fundamental feature of a volute and acts in concert with the impeller exit area, as emphasized by Anderson⁷ and Worster²⁵ and described in the earlier part of this section on *Designing the Collector* in terms of the intersection of the casing and impeller lines²⁶.)

The circumferential distribution of the volute cross-sectional area A_v versus polar angle θ_v from the tongue (Figure 20a) is developed in the latter part of Table 11, a final listing being presented for a) linear A_v versus θ_v and b) constant one-dimensional angular momentum $r_v V_v$.

TABLE 11 Volute casing (design example)

Refer to Fig. 20

A) Throat area A_T

Simultaneously solve for A_T from

$$a) \quad r_T \frac{Q}{A_T} = 0.95 r_2 V_{\theta,2} \quad (\text{Eq. 61, using 0.95}) \quad \text{since } A_T = \frac{Q}{V_T}$$

and

$$b) \quad r_T = r_3 + \frac{1}{2} \sqrt{A_T} + t_1 \quad (\text{Fig. 20, assuming a square throat})$$

Eliminating r_T in combining (a) and (b) yields a quadratic equation in $\sqrt{A_T}$:

$$c) \quad A_T \times \left(\frac{0.95 r_2 V_{\theta,2}}{Q} \right) - \frac{1}{2} \sqrt{A_T} - (r_3 + t_1) = 0$$

Taking $r_3 = 1.06 r_2$ (i.e., "Gap B" = $0.06 r_2$), $r_3 = 6.36$ in (161.5 mm); assuming $t_1 = 0.7 t_2 = 0.7 \times 0.24 = 0.168$ in. (4.3 mm); and using consistent units in (c); leads through the quadratic formula to

$$A_T = 32.60 \text{ in}^2 (21,032 \text{ mm}^2)$$

Thus $V_T = 24.60$ ft/sec (7.50 m/s)

B) Cross-sectional area

If volute cross-sectional area A_v is distributed linearly vs. θ_v , A_v and the radius r_v to the center of the cross section are as follows:

$$d) \quad A_v = A_T \frac{\theta_v (\text{deg})}{360} = \frac{32.60}{360} \times \theta_v \text{ in}^2 = \frac{21,032}{360} \times \theta_v \text{ mm}^2$$

$$e) \quad r_v = r_3 + \frac{1}{2} \sqrt{A_v} = 6.36 + \frac{1}{2} \sqrt{A_T (\text{in}^2)} \quad \text{in} \\ = 161.5 + \frac{1}{2} \sqrt{A_T (\text{mm}^2)} \quad \text{mm}$$

If A_v is found subject to constant angular momentum with $r_v V_v = 0.95 r_2 V_{\theta,2}$,

$$f) \quad A_v = \frac{Q}{V_v} \times \frac{\theta_v (\text{deg})}{360} = \frac{Q \left(r_3 + \frac{1}{2} \sqrt{A_v} \right)}{0.95 r_2 V_{\theta,2}} \times \frac{\theta_v (\text{deg})}{360}$$

TABLE 11 Volute casing (design example) (Continued)

This is a quadratic equation in $\sqrt{A_v}$, which, when solved, gives the results for "constant $r_v V_v$ " in the listing below, r_v being found from Eq. (e) for both this and the linear case.

Volute cross sections (design example)

θ_v deg	Linear A_v vs. θ_v				Constant ($r_v \times V_v$)			
	A_v		r_v		A_v		r_v	
	in ²	mm ²	in	mm	in ²	mm ²	in	mm
0	0.00	0	6.36*	161.5*	0.00	0	6.36*	161.5*
45	4.08	2,629	7.37	187.2	3.15	2,031	7.25	184.1
90	8.15	5,258	7.79	197.8	6.64	4,286	7.65	194.3
135	12.23	7,887	8.11	205.9	10.39	6,701	7.97	202.5
180	16.30	10,516	8.38	212.8	14.34	9,250	8.25	209.6
225	20.38	13,145	8.62	218.9	18.48	11,922	8.51	216.1
270	24.45	15,774	8.83	224.3	22.79	14,706	8.75	222.2
315	28.53	18,403	9.03	229.4	27.27	17,596	8.97	227.9
360	32.60	21,032	9.21**	234.1**	31.91	20,588**	9.18**	233.3**
* $r_v = r_3$ at $\theta_v = 0$								
** transition to $r_1 = 9.38$ in (238.3 mm) and $A_1 = 32.6$ in ² (21032 mm ²) due to tongue thickness								

This listing illustrates an earlier statement that the latter approach produces smaller areas upstream of the throat than does the former. Frictional effects in the smaller- A_v portion of the volute are less prominent using the linear approach. A CFD assessment at all flow rates can be made to guide the design choice here. Finally, the one-dimensional constant $r_v V_v$ method can be improved upon by integrating a constant $r V_v$ distribution over each cross section, and the proper design must satisfy also continuity²⁶.

Estimated Performance Characteristics Detailed computation of hydraulic losses, together with leakage, disk friction, and other mechanical drags was done as described previously, and the results are presented in Figure 30. The method predicts the pump efficiency to be 85.5% at the design point of 2500 gpm (0.1577 m³/s), peaking at 86 percent at a 5% lower flow rate. In view of the preceding discussion of the behavior of the mean relative velocity W at the leading edge of the impeller blades in Figure 29, this shift of the BEP to a slightly lower flow rate than the design value comes as no surprise. Slightly larger values of the inlet blade angles than those chosen for this example (see Figure 26a) would be needed for the BEP to have the same flow rate as the design point. Peaks at the design point at 77 hp (57 kW), indicating that this is a "non-overloading" design; that is, shaft power does not increase beyond this flow rate. The empirical methods reviewed earlier for shut-off head and power were applied here and blended into the one-dimensionally computed curves at half of the BEP flow rate.

The shut-off head coefficient value of 0.596 is close to the 0.585 value of Stepanoff⁵ and is admittedly open to alteration. The rise of head from design point to shut-off is from 104 up to 161 ft (31.7 up to 49.1 m) or 55%. This percentage could be smaller and still ensure stable operation in any typical system; however, the non-overloading feature could change, the power peaking at a higher flow rate. In retrospect, the design-point head coefficient ψ of 0.385 could be larger without the corresponding diffusion-factor results (Figure 29) becoming excessive. One would conclude from this exercise that the ψ -curve of Figure 12 is conservative and could be higher. Of more significance, however, is the demonstration in this design example of the utilization of easily applied fundamental fluid dynamical analyses such as the diffusion assessment illustrated in Figure 29 as the basic arbiters of design choices such as the head coefficient, number of blades, and so on. Furthermore, the shapes of the performance characteristics are revealed.

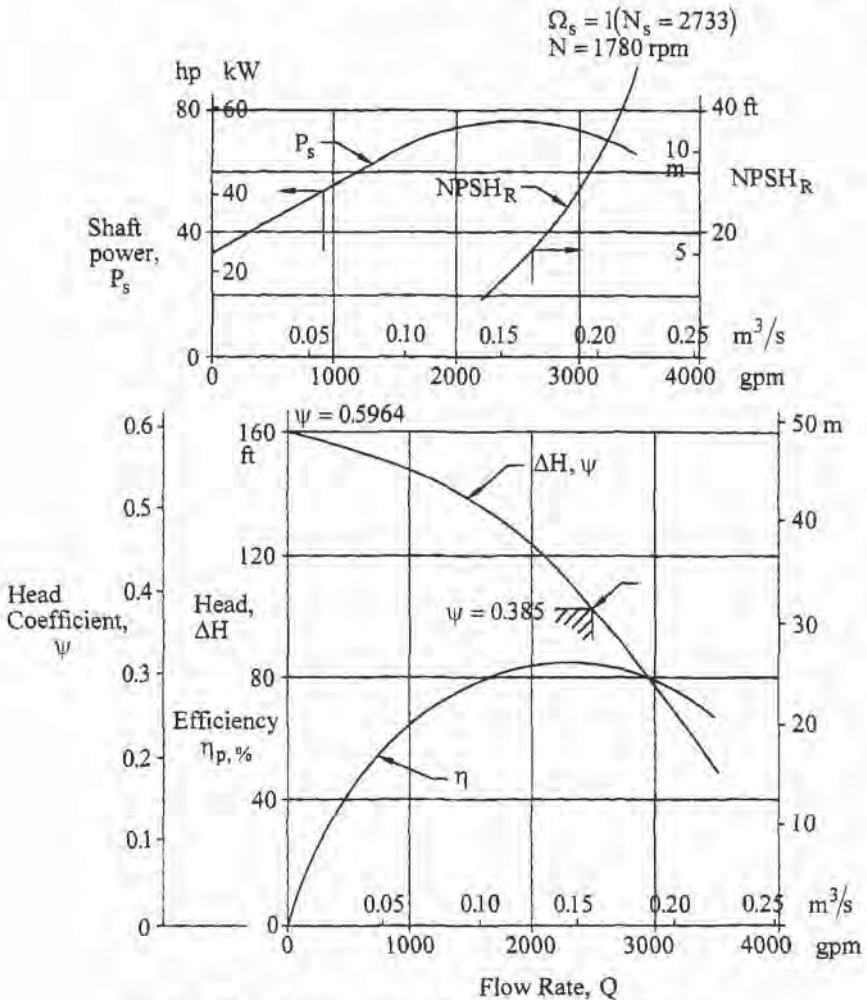


FIGURE 30 Estimated performance characteristics (design example)

The shut-off power is 34 hp (25 kW), arising from a power coefficient of 0.047 from Mockridge's correlation³⁶ for a b_2/D_2 of $1.59/12$ ($= 40.4/304.8$) $= 0.1325$ —as found from Figure 23. This is 44% of the design-point power, a typical result. As seen in Section 2.1.3, this percentage increases with specific speed, where, of course, b_2/D_2 is also larger³⁶.

The NPSHR ($= \text{NPSH}_{3\%}$ for this example) at off-design conditions was estimated empirically. At the higher, non-recirculating flow rates, NPSHR is related to the head required to increase the relative velocity within the blades to values higher than at the design point¹⁵. As stated in Section 2.1.3, operation at $\text{NPSH}_{3\%}$ involves performance in the presence of extensive internal two-phase behavior. This is complicated by recirculation at the lower flow rates. CFD codes exist that can deal with multiphase, recirculating flow and can successfully predict the entire NPSHR curve; which, prior to the advent of such capability, usually had to be established experimentally.

HIGH-ENERGY PUMPS

Over the past few decades, there has been a trend toward pumping machinery that concentrates more power within a given volume. This trend is driven by cost and technology improvements. The basic energy transfer relationships show that smaller size demands higher rotational speed. Thus, over the same time period, speeds of high-power pumps have been increasing. Moreover, the number of stages in multistage pumps has been decreasing. There have been spikes in these trends; the resulting pumps suffering from excessive vibration, rotor and hydraulic instabilities, component failure, and cavitation damage⁸⁷. The term “high energy” has been applied to these machines, and this label can be quantified in terms of the stresses arising in critical pump components and the likelihood of an adverse mechanical response that such stress levels imply. Research has led to technical solutions for effectively controlling rotodynamic behavior and reducing unsteady hydraulic thrust and surge as well as cavitation erosion^{53,58,59}. The resulting pump reliability improvements and life extension should enable the previous trends to continue. Being aware of the energy level enables the pump user to assess whether operation and maintenance difficulties are likely to occur after the pump is installed and running, and it enables the designer to take the appropriate measures to ensure the technical integrity of the product.

Pressure Pulsations Measured at the inlet or outlet port, the amplitude of the pressure pulsations can be a significant fraction of the pressure rise of the pump—especially at flow rates well below that of the BEP for the speed involved. Sources are a) the interaction of the pressure fields of the impeller and diffuser or volute, b) unsteady separated and reversed flows at impeller inlet and discharge and in the diffuser, c) cavitating flows, and d) combinations of these phenomena. Pressure pulsations presumed to exist at the impeller OD from the interaction of impeller blade-to-blade and diffuser vane-to-vane variations of pressure have been calculated by inviscid flow analysis to have a peak-to-peak amplitude that is of the same order as the static pressure rise of the impeller⁶⁰. Moreover, the viscous, thicker wakes existing at lower-than-BEP flow rates (here called “low flows”) and separated recirculating fluid from both impeller and diffuser that participate in these interactions can be expected to increase the pressure pulsation amplitude at such conditions.

Figure 31 confirms these ideas, showing a bronze impeller that operated extensively at low flow. Cavitation pitting can be observed near the OD of the impeller, which means that the rarefactions of the pressure waves were below the vapor pressure of the liquid—these pressure minima therefore being below the inlet pressure to the impeller. Moreover, the bulged-out shrouds can be assumed to be the result of the repeated occurrence of the associated pressure spikes (that is, the maxima of the pressure waves) within the radial gap (“Gap B”) between the impeller blades and the diffuser vanes, the sidewall pressures on the outsides of the shrouds remaining comparatively constant. At greater values of design pressure rise than was the case for this impeller, this phenomenon creates correspondingly greater forces that have led to actual breakage of the impeller shrouds and diffuser vanes⁶¹. The cavitation seen in Figure 31, can also be observed on the leading edges of diffuser vanes, as in Figure 21 of Section 12.5, and this raises the possibility of diffuser vane breakage.

Energy Level: Stage Pressure Rise Even in the absence of the weakening effect of cavitation erosion, the leading edge of a diffuser vane or volute tongue is a representative, highly stressed zone within a pump that is subject to failure if the magnitude of the pressure pulsations arising from the impeller-diffuser interactions just described is sufficiently large. Thus, the fluid dynamically induced stresses in these vanes can be the basis for quantifying the energy level of a pump stage. In Table 12, this concept is developed into an expression for the stress in terms of the fluctuating pressure magnitude δp that is assumed to act across the vane leading edge as illustrated in the table. The width b of the vane is close enough to b_2 of the impeller exit to utilize the relationships for ψ and ϕ_1 of Figure 12 to relate b/D to specific speed in Eq. (c) of the table. For similar velocity fields, the pressure pulsation magnitude δp is a constant multiplied by the stage pressure rise ΔP_{stage} . Thus, for a limiting value of stress, the concept of a limiting stage pressure rise

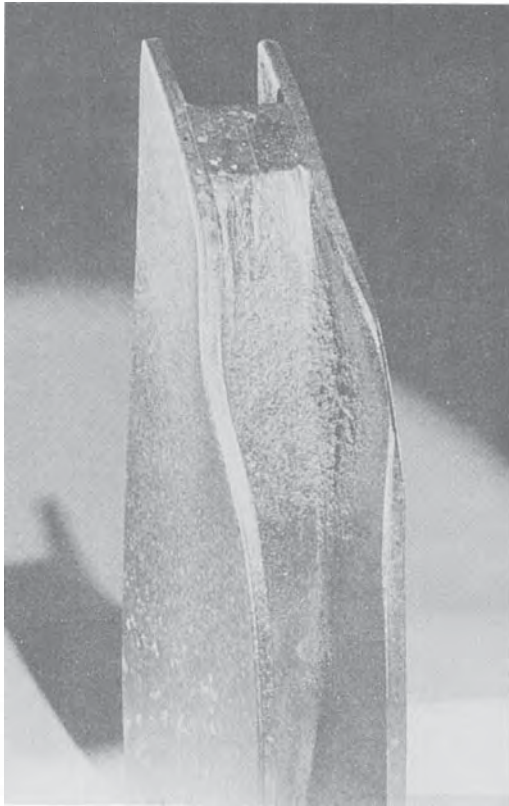


FIGURE 31 Damage to impeller from low-flow operation (Source: E. Makay in *Power*)

emerges in Eq. (e). The constant K is chosen from experience, which leads to the resulting Eq. (f). This relationship is plotted in Figure 32. [It will be observed that this choice for K corresponds to a limiting stress s from Eq. (d) of 6600 psi (45.5 MPa) that would exist if δp were equal to ΔP_{stg} —with $t/D = 0.01$ and $\varepsilon = 0.85$ as in the example.] The inverse variation with specific speed is a consequence of the greater b/D of higher- Ω_s pumps (as developed in Table 12), the wider vane introducing more stress at the juncture with the sidewalls for the same pressure loading and so imposing a lower stage pressure-rise limit. Conversely, lower- Ω_s pumps should have higher limits for ΔP_{stg} .

Figure 32, therefore, illustrates this concept of a *limiting stage pressure rise* as a measure of the energy level of centrifugal pumps, the basis being a limiting stress level in a critical component of the pump. Starting with stress at other locations in the pump leads to similar results. To provide perspective, specific examples of pumps that by this definition are in the *high-energy domain* are plotted on the figure. These data points are taken from Table 13, which contains information for several well-known liquid rocket engine turbo pumps^{62,63,64} and for some representative high-energy electric utility boiler feed pumps^{53,57}.

The last column in Table 13 is another, more general way of comparing the energy level of these machines; namely, the *torque per unit volume*, which also has the dimensions of stress. For fixed ratios of stage width, casing OD, and other dimensions to impeller radius r ($= r_2$), torque per unit volume differs from the listed values of torque/ r^3 by a factor. The actual torque per unit volume therefore ranges from one-half to one-sixth of the tabulated

TABLE 12 Hydraulically induced stress levels

- Stress at juncture of diffuser vane leading edge with sidewall:

$$s = \frac{\delta p b^2}{2t^2} = \frac{(\delta p/2)(b/D)^2}{(t/D)^2} \quad (a)$$

- From Fig. 12 and Eq. 36:

$$\frac{b}{D} = \frac{Q}{V_{m,2} 2\pi r_2 \varepsilon D} = \frac{Q}{\Omega r_2 \frac{V_{m,2}}{\Omega r_2} 4\pi r_2^2 \varepsilon} = \frac{Q_s}{\phi_1 4\pi \varepsilon} \quad (b)$$

- From Eq. 50:

$$\frac{b}{D} = \Omega_s^2 \psi^{3/2} / (4\pi \phi_1 \varepsilon) \quad (c)$$

- With $\psi \approx 0.383/\Omega_s^{1/3}$ and $\phi_1 \approx 0.1715\sqrt{\Omega_s}$

the stress Eq. (a) becomes, with Eq. (c)

$$s = \frac{\delta p \times \Omega_s^2}{165 \times (t/D)^2 \varepsilon^2} \quad (d)$$

- Example for $t/D=0.01$, $\varepsilon=0.85$, $\Omega_s=0.6$ and $\delta p=1000$ psi (6.895 MPa):

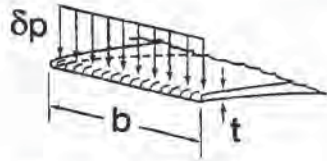
$$s = \frac{1000 \times (0.6)^2}{165 \times (0.01)^2 \times (0.85)^2} = 30,198 \text{ psi (208.2 MPa)}$$

- For δp being proportional to ΔP_{stg} , and for a fixed geometry and a given limiting value of stress, Eq. (d) becomes

$$\Delta P_{\text{stg}} = K / \Omega_s^2 \quad (e)$$

- Experience leads to a choice of K such that

$$\Delta P_{\text{stg}} = \frac{78.72}{\Omega_s^2} \text{ psi} = \frac{0.5427}{\Omega_s^2} \text{ MPa} \quad (f)$$



torque/ r^3 , depending on the casing or barrel thickness, and so on. However, as has been demonstrated, the critical stresses are more closely associated with the impeller OD, which makes comparison of the tabulated values more relevant. Thus, a pump with high torque/ r^3 can be expected to have correspondingly high local internal stresses. The maximum values listed, namely for the high-pressure propellant pumps on the RD-170 (Russian) and SSME (U.S. Space Shuttle) engines, tend to explain the high level of research and development that was necessary to successfully deploy these machines. Illustrations of some of these rocket engine pumps can be found in Section 12.18.2. Similarly, Section 12.5

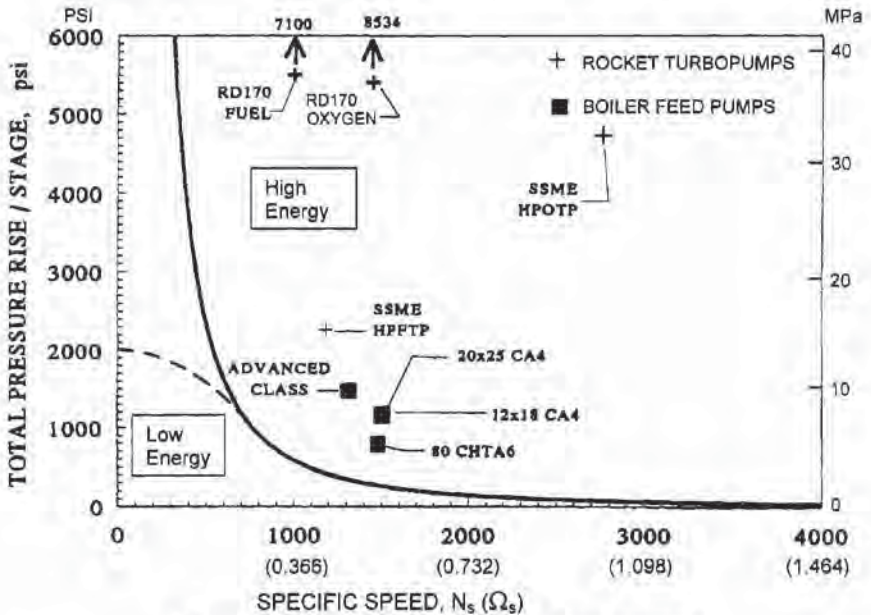


FIGURE 32 Pump energy level defined in terms of stage pressure rise

provides examples of high-energy boiler feed pumps: the massive, barrel-type construction of these machines is illustrated in Figure 18. Specifically, the first boiler feed pump listed in Table 13 is the sole feed pump supplying the steam generator of a super-critical 1300 MW electric generating unit and consumes nearly 50 MW of shaft power. It can be seen in Figures 7 and 20 of Section 12.5.

For pumps in the *low-energy domain* of Figure 32, normal design and manufacturing practices result in a more benign mechanical response to the abnormal fluid phenomena discussed here. However, for locations other than the diffuser entrance, which was the basis for the development of the figure, limiting stresses could be reached at considerably lower values of stage pressure rise. For this reason, the dashed line is offered as the upper limit of the low-energy domain; however, a thorough analysis of the stresses in any given application is the ultimate determinant of all the limits suggested in Figure 32 and of the acceptability of the design. It can now be seen that the design example treated earlier in this section is of the low-energy variety; therefore, the special design problems treated here and further on are of relatively little concern in such pumps. On the other hand, if the curve in Figure 32 were extended to much *higher specific speeds*, it would be found that many existing, large, high- Ω_s , low-head pumps are high-energy machines by this stress-related definition. It is therefore not surprising that such pumps generally require full stress and modal analyses to identify possible destructive resonances and stresses.

Fluid/Structure Interactions With the dimensions of pump energy level identified, the next step is to continue the examination of the problems mentioned previously and the methods that have become available for solving them. Attention is focused on fluid dynamical phenomena because, as the previous discussion of pressure pulsations implies, most of the adverse mechanical behavior exhibited by pumps originates from the behavior of the internal flow field. Excessive measured vibrations, material erosion, and component failures are often the external symptoms of fluid/structure interaction phenomena that are fundamentally explained from a fluid dynamics perspective. In addition to the fluid dynamically induced stresses in the pump structure that culminated in the domain

TABLE 13 Data on high-energy pumps*

Pump	No. of stages	Speed rpm	Flow rate gpm (m ³ /s)	Head/stg ft (m)	ΔP /stg psi (MPa)	Specific Spd N _s (Ω _s)	Torque/r ³ psi (MPa)
Liquid Rocket Propellant Turbopumps							
Saturn V Booster – F1 Engine: Oxygen	1	5,490	25,080 (1.582)	3,100 (945)	1,530 (10.5)	2,095 (0.767)	371 (2.6)
Saturn V Booster – F1 Eng: Fuel (RP1)	1	5,490	15,640 (0.986)	5,100 (1,555)	1,810 (12.5)	1,130 (0.413)	172 (1.2)
Space Shuttle Main Engine: Oxygen (= SSME HPOTP)	1	31,100	7,240 (0.457)	9,700 (2,957)	4,800 (33.1)	2,700 (0.988)	1,431 (9.9)
Space Shuttle Main Engine: Fuel (H ₂) (= SSME HPFTP)	3	37,400	16,300 (1.028)	66,700 (20,330)	2,280 (15.7)	1,150 (0.421)	200 (1.4)
RD-170 – Oxygen	1	13,850	25,008 (1.578)	17,300 (5,273)	8,534 (58.8)	1,448 (0.530)	1,521 (10.5)
RD-170 – Fuel (= RP1)	1	13,850	14,485 (0.914)	20,000 (6,096)	7,100 (49.0)	992 (0.363)	698 (4.8)
Boiler Feed Pumps							
20x25 CA-4 (IDP)	4	4,160	21,620 (1.364)	3,000 (914)	1,169 (8.1)	1,509 (0.552)	128 (0.9)
12x18 CA-4 (IDP)	4	5,800	11,000 (0.694)	3,000 (914)	1,169 (8.1)	1,500 (0.549)	130 (0.9)
80CHTA-6 (IDP)	6	5,800	6,250 (0.394)	2,100 (640)	820 (5.7)	1,479 (0.541)	87 (0.6)
Advanced Class (Sulzer)	2	6,500	9,510 (0.600)	3,800 (1,158)	1,480 (10.2)	1,312 (0.480)	136 (0.9)

*Notes to this table: Flow rates apply to the pump inlet. RD-170 pump data derived from Sutton⁶² and Advanced Class boiler feed pump data from Ref. 57. Remaining data derived from Table 1 of Section 12.18.2, NASA reports^{63,64} and from information supplied by Flowserve Corporation.⁵³

definition of Figure 32, these phenomena encompass a) blade-vane interactions, b) recirculation, c) anomalous axial thrust behavior, and d) cavitation.

In order to illustrate the methods for dealing with the problems created by these fluid/structure interactions, a sample suction stage of a high-energy multistage pump is utilized as the quantifying focus in each case. Table 14 contains the conditions and essential features of this machine, a meridional view of which is shown in Figure 33. As with the earlier design example, this stage has been designed in accordance with the procedures outlined for that example. This includes the velocity diagrams shown in the figure, as well as the design of the eye, the hub and shroud profiles, and the blading. A preliminary design

TABLE 14 Data for high-energy pump suction stage**A. Design Inputs**

Fluid: Boiler feed water at 350°F (177°C) $\Rightarrow$ sp. gr. = 0.89, ν = 0.17 cs.	
Flow rate, Q , gpm (m^3/s)	20,000 (1.262)
Head rise, ΔH , ft (m)	3,000 (914.4)
Rotative speed, N , rpm	4,700
Outlet flow coefficient, ϕ_2 (Figure 12)	0.1328
Head coefficient, ψ (Figure 12)	0.455
Specific speed, Ω_s (N_s / n_q)	0.6 (1640/32)
Inlet flow coefficient, ϕ_e	0.3 ($=\tan 16.7^\circ$ deg.)
Number of Impeller blades, n_b	7
Number of Diffuser vanes, n_v	9

B. Calculated Performance Data

1. Efficiency

a) BEP of 20,000 USgpm (1.2618 m^3/s)

$$\eta = \eta_p = 0.877 \quad (\text{vs. } \eta = 0.8682 - \text{from Fig 10, for } X = 1)$$

$$\eta_{hy} = 0.900$$

$$\eta_m = 0.985$$

$$\eta_v = 0.989 \quad [216 \text{ gpm } (0.0136 \text{ m}^3/\text{s}) \text{ front ring leakage; rear ring leakage included in } \eta_{hy}]$$

b) At 50% Flow : $\eta_p = 0.693$ ($= 0.79$ of η_{BEP} ; 0.8 is typical)

2. Head Rise

$$\Delta H_{\text{BEP}} = 3000 \text{ ft. (914 m)} \Rightarrow \psi = 0.455$$

$$\Delta H_{\text{shutoff}} = 3697 \text{ ft. (1127 m)} \Rightarrow \psi_{s/o} = 0.560$$

$$\Delta H\text{-vs-}Q \text{ curve stability indicator, } \Delta H_{s/o}/\Delta H_{\text{BEP}} = 1.23 \text{ (doubtful if less than 1.2)}$$

3. Power

$$P_{s,\text{BEP}} = 15,376 \text{ hp (11,466 kW)} \sim \hat{P} = 0.0571$$

$$P_{s,\text{SHUTOFF}} = 6,144 \text{ hp (4,583 kW)} \sim \hat{P}_{s/o} = 0.0228 \quad (= 0.01966 \times 1.16 \text{ from Fig. 23})$$

$$\uparrow$$

$$= K \text{ for } \frac{b_2}{D_2} = 0.0779$$

was also made of the vaned diffuser and return vanes, following the guidelines presented in the foregoing *Design Procedures* section. The result is a representative machine to which the following paragraphs continually refer.

Blade-Vane Combinations Certain numerical combinations of impeller blades and diffuser vanes (or inlet guide vanes, where they are employed) have been shown to have acoustic consequences that can exacerbate the pressure pulsations arising from the interaction of impeller and diffuser flow fields. Bolleter reviewed the types of interactions that can occur and the consequences with regard to pressure pulsations and resonance⁵³. These types are associated with the integer difference m between the multiples of the number of impeller blades and the number of diffuser vanes, as explained in Table 15. To check

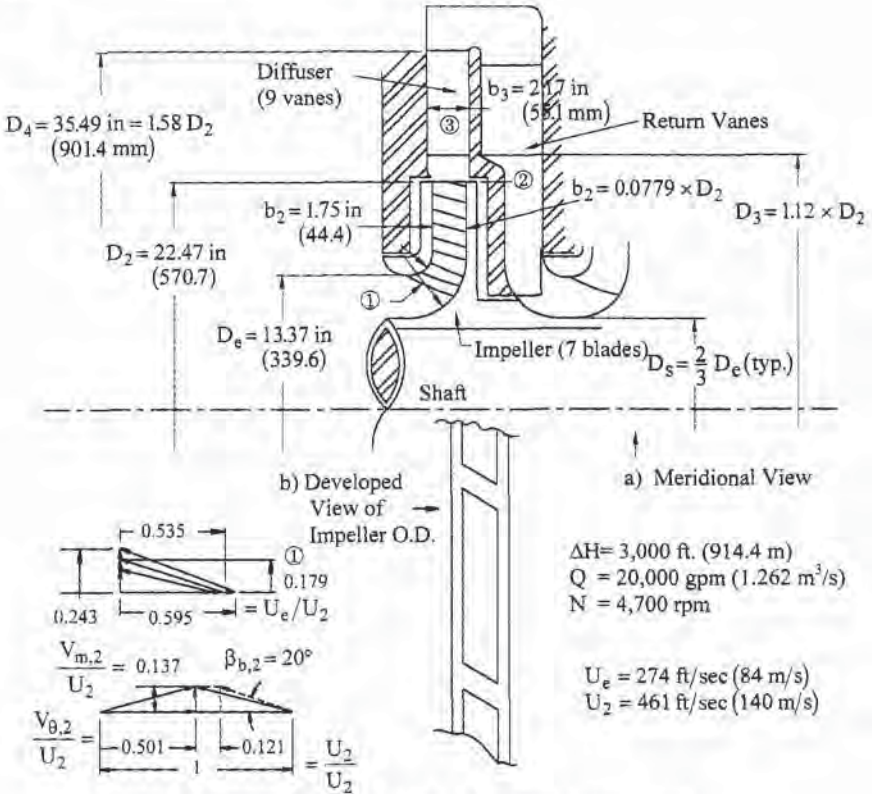


FIGURE 33 Sample suction stage of high-energy multistage pump

the blade-vane combination, one simply forms a matrix of the multiples as shown and lists the difference in each cell.

To illustrate the method, rather than use the pump of Table 14, a different example is chosen in order to dispel the notion that one can always use two different prime numbers for blade-vane combinations. This is an existing case of a 60,000 hp (45 MW) single-stage pump of $\Omega_s = 0.6$ ($N_s = 1640$) and $\Delta p = 357 \text{ psi}$ (2.46 MPa), which has 7 impeller blades and 13 diffuser vanes⁶⁵. Notice that $m = 1$ in the second order of impeller blade number, which means there should be pressure pulsations at $2 \times 7 \times \text{rpm}/60 \text{ Hz}$. In fact, this pump has exactly this vibration and pressure pulsation frequency in the field, not least because the pump is operating at just the right speed for the acoustic waves emanating from successive interaction points to reinforce each other in producing unacceptable pressure pulsations.

Even with this simple method, the final choice of the blade-vane combination is usually a compromise. For example, the method shows that all double-volute pumps have $m = 0$ or 1 in the first order of impeller blade number, depending on whether this number is even or odd. $m = 0$ means that all the blades or vanes are interacting at the same instant, a consequence of which is torque ripple. High-energy volute pumps exist for both even and odd cases, and quite large gaps between blades and vanes ($\approx$ "Gap B") are employed to mitigate these effects.

The reader who checks the pump of Table 14 by this method will find that $m = 2$ in the first order of both blades (7) and vanes (9) but that $m \neq 0$ everywhere else in the matrix. $m = 1$ occurs only in the highest orders of both and should therefore be of little consequence. Checking whether 10 or 11 vanes would be better yields $m = 1$ in the third order

TABLE 15 Choosing vane combinations to minimize pressure pulsations³³

1. Z_i is the number of impeller blades.
2. Z_d is the number of diffuser vanes.
3. Each vane combination produces interacting pressure fields between the blade rows.
4. These fields rotate about one or more "nodal diameters," "m" producing unsteady loads.
5. The value of "m" determines the nature and severity of these loads.
6. "m" is the difference between $(p \times Z_i)$ and $(q \times Z_d)$, where p and q are integers.
7. To find "m", one sets up a matrix of diffuser vanes and their multiples vs. impeller blades and their multiples or "orders." A minimum of the first three orders should be used. This is illustrated as follows for an impeller with 7 blades ($Z_i = 7$) and a diffuser with 13 vanes ($Z_d = 13$):

		Impeller blades and their multiples:			
		p = 1 7 (= Z_i)	p = 1 14	p = 1 21	p = 1 28
Diffuser vanes and their multiples:		Differences "m" between $(p \times Z_i)$ and $(q \times Z_d)$ are given below:			
q = 1	13 (= Z_d)	6	1	8	15
q = 2	26	19	12	5	2
q = 3	39	33	25	18	11

The significant values of "m" are 0, 1, and 2. The lower the order of either the impeller blades or the diffuser vanes, the stronger is the interaction. The responses that occur for these "m" values are as follows:

- $m = 0$ - strong pressure pulsations and fluctuating axial loads and torsional forces.
- $m = 1$ - pressure pulsations and fluctuating radial loads.
- $m = 2$ - small pressure pulsations and a chance of resonance of the impeller about the two nodal diameters—at a frequency equal to the value of $q \times Z_d$ times the rotational frequency (rpm or rps).

In this example, $m = 1$ in the second order of the impeller blades (i.e., $p = 2$); thus, the frequency of the resulting disturbances will be twice that of blade passing frequency $p \times Z_i \times \text{rpm}$ or $2 \times 7 \times \text{rpm}$ cycles per minute or $2 \times 7 \times \text{rpm}/60$ Hz. This is therefore a poor vane combination and should be avoided—especially for such low orders of vane number; i.e., for such small values of p and q. (Here, $p = 2$ and $q = 1$.) In general, $m = 0$ and 1 should be avoided altogether, and $m = 2$ should be avoided in the lower orders.

of impeller blades and second order of diffuser vanes, which is probably less desirable than $m = 2$ in the first orders. Therefore, Figure 33 shows a value of "Gap B" that is 12% of the impeller radius, which by thus distancing the interacting flow fields of the impeller blades and diffuser vanes from one another should provide adequate protection from pressure pulsations and excitations of resonance. (The gaps at the impeller OD will be discussed further on.)

Recirculation Separation and stall of the fluid flowing in the passages of impellers and diffusers occurs at low flow because of the incidence and large reduction in the one-dimensional velocity relative to the passages that happen at low flow. This and the consequent

recirculation patterns in the impeller were discussed and illustrated in Figure 6. Fischer and Thoma⁶⁶ visually observed and recorded the flow patterns, finding that as flow rate is reduced, wakes on the suction side of all blades thicken until they occupy half the passage width at half the BEP flow rate. At lesser flow rates, the wakes continue to thicken but become irregular, stalling in one passage and not the others—the stall pattern moving into and out of adjacent passages and so rotating relative to the impeller. As shut-off is approached, this rotating pattern is accompanied by reversed flow emerging from the inlet of the stalled passage. Fraser, working with typical impeller geometries, formulated rules for computing the flow Q_{SR} at which this reversal occurs as Q is reduced at constant speed⁶⁷. His expressions, found further on, include the effect of impeller eye size on Q_{SR} . As one might expect, a pump with an eye diameter approaching that of the impeller OD will have Q_{SR} approaching Q_{BEP} . At $Q < Q_{SR}$, the impeller flow patterns are highly unsteady—as is usually the case with massively separated flows—creating nonsynchronous, low-frequency or random pressure pulsations, the resulting shear layers between the reverse-flowing and in-flowing fluid having vortices with locally low pressures so that cavitation can also exist. Fraser also quantified the flow rate Q_{DR} below which impeller discharge recirculation exists. Forces from such motion can cause fatigue failure of the impeller blades, diffuser vanes or volute tongue, cavitation erosion also playing a part as in Figure 31. In Section 2.1.3, Fraser describes the identification and consequences of recirculation in detail, the more general designation Q_R referring to either Q_{SR} or Q_{DR} , depending on whether Q is between or below both.

The ability for pumps to operate with any form of separation; stall; or, worse, flow reversal (recirculation) depends on the energy level. This can be approximately quantified, as outlined under the subject of *Minimum Flow Limits* further on, which include consideration of accompanying cavitation activity.

Axial Thrust Response to Recirculation Discharge recirculation usually involves backflow from the diffuser, itself containing oscillating flow patterns and rotating stall. Fluid emerging from the diffuser will be spinning opposite to the direction of rotation, such fluid having a major effect on the sidewall gap flows as it joins the leakage flows described under *Predicting Axial Thrust*. As this fluid invades the sidewall gaps, it can slow or virtually cancel the usual positive swirling of the gap fluid. Iino, Sato, and Miyashiro experimentally observed and recorded this behavior, which was exaggerated by shifting the impeller axially and by changing the ring clearances⁶⁸.

An added, not unexpected effect is that as Q is reduced below Q_{DR} , the invading flow from the diffuser can favor the front or back side of the impeller and then *switch sides* upon further reduction of Q . This effect is clearly seen in the experimental thrust-versus- Q plots of Figure 34, the impellers having been shifted as just described. Depicted there is the resulting net load on the axial thrust bearing of an eight-stage, 3600-rpm diffuser pump (all impellers facing in the same direction) that had a cylindrical balancing drum (not a self-compensating balancing disk). The drum was sized so as not to completely eliminate the thrust—in order to avoid thrust reversals. The solid lines are the predicted net thrust according to the methods outlined in Table 4 for three axial positions of the impeller. The large excursions in net thrust were eliminated by restricting the entry of the invading diffuser backflow into the sidewall gaps—through a tightening of the gap between the shrouds of impeller and diffuser (Gap “A”) in Figure 35. Gap “A” is not effective unless the “overlap” of the two mating shrouds is from four to six times the gap dimension⁶¹. Moreover, if Gap “A” is minimized, this can exaggerate the blade-vane interactions, making it necessary to open up Gap “B” more than would be necessary were Gap “A” not minimized⁶⁹.

A further possibility that has been observed in a single-stage double-suction pump is the *unsteadiness* of impeller discharge recirculation and, most likely, of the diffuser or volute backflow. The side-to-side switching just mentioned appears in Figure 36 to be happening as a function of time as well as of flow rate Q , as evidenced by the axial motion, which is accompanied by discharge pressure pulsations. The “fix” mentioned in the figure was, again, mainly minimizing Gap “A.”

Closing Gap “A” and opening Gap “B” are procedures that have been widely and successfully applied in high-energy pumps, which usually work well at BEP but run into

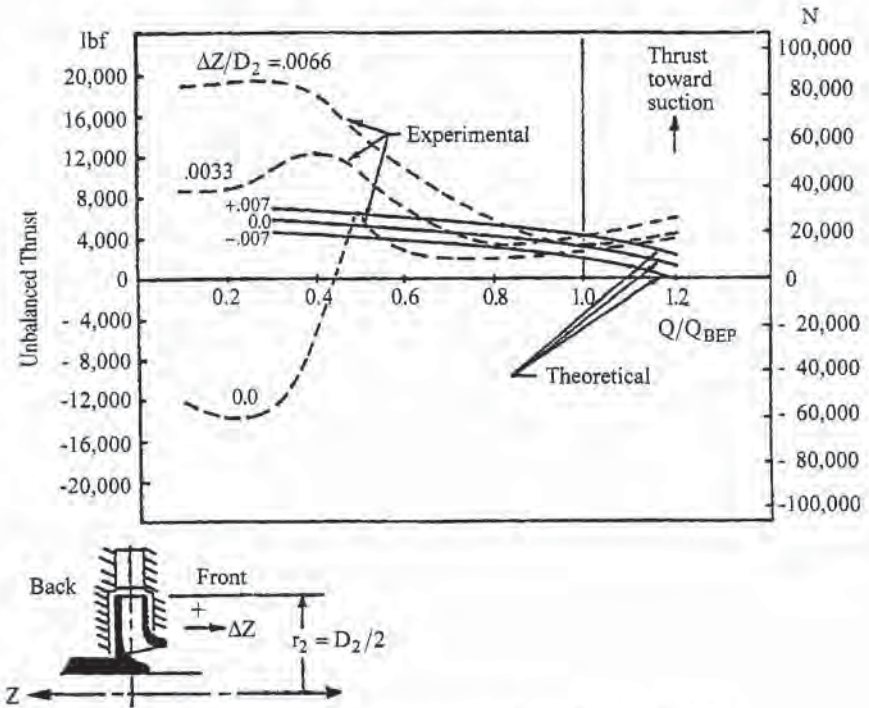


FIGURE 34 Axial thrust response to recirculation (Source: Flowserve Corporation)

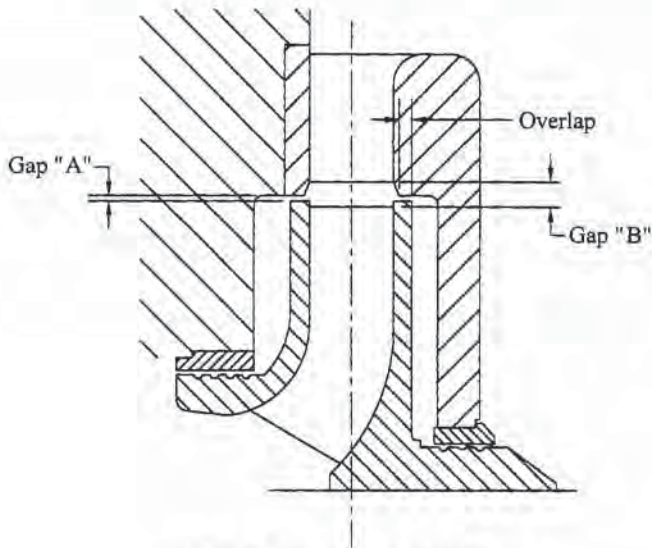


FIGURE 35 Gaps at impeller periphery²³

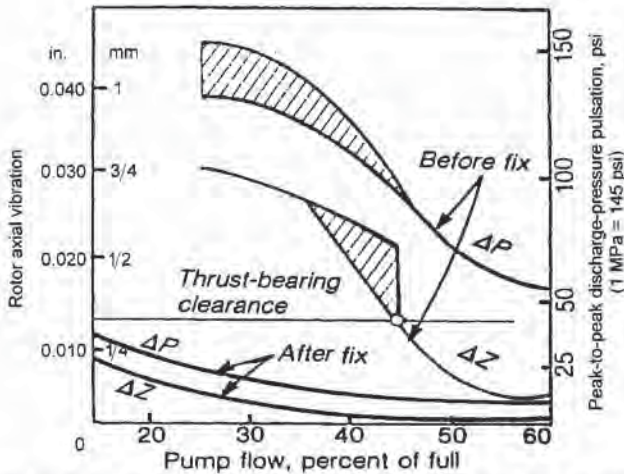


FIGURE 36 Eliminating unsteady thrust and pressure pulsations (Source: E. Makay in *Power*.)

difficulties at low flow⁶⁹. These procedures have proven to cure the thrust and pressure pulsation behavior just described and also improve the low-flow performance curve shape⁶⁵. The results are high-energy pumps that can operate smoothly over a wider range of flow rate than—in many cases—was originally expected or specified. Nevertheless, the higher the energy level, the more intolerable is any unsteadiness and pressure pulsation activity.

Minimum Flow Limits Because the intensity of pressure pulsations and the accompanying vibrations can increase beyond acceptable limits as flow rate is reduced at constant speed below the BEP flow, or more specifically, below Q_R , expressions for how low Q/Q_{BEP} can be without exceeding these limits have been developed. Manufacturer and user groups such as the Hydraulic Institute (HI), the International Organization for Standardization (ISO), and the American Petroleum Institute (API) have specified vibration limits that must be met at what is usually called the minimum continuous stable flow (MCSF) or simply Q_{min} . In an attempt to answer the question of how far into the recirculation zone (where $Q < Q_R$) an operator can take a pump before reaching the MCSF, Gopalakrishnan proposed a general rationale for computing Q_{min} that takes the energy level into account—through the flow rate and rotative speed of the machine⁷⁰. These two quantities imply the blade tip speed at the inlet of the impeller. (Through typical ratios of impeller OD to eye diameter, this also implies the OD tip speed and therefore the head, the energy level having been defined in Figure 32 in terms of stage pressure rise ΔP_{stage} .) The theory for this method is defined in Part A of Table 16, in which Q_{min} is computed as the product of a series of K-factors multiplying the value of Q_R , which is computed according to Fraser's earlier development⁶⁷. Conceptually, the product of these factors approaches unity in the maximum-energy case, where the instabilities accompanying any recirculation at all are significant. Conceivably, this product could exceed unity at the highest energy levels, as separation and stall and the attendant unsteadiness must occur—as Q is reduced—before reaching the backflows that characterize Q_R , which was observed by Fraser as the value of flow rate for the onset of recirculation (see Section 2.1.3).

K_1 and K_3 are given in Table 16, as equations that have been curve-fitted to speed/flow rate- and NPSH-effect charts that appeared in Gopalakrishnan's presentation⁷⁰. This includes the ability to enter any speed (rpm) into the computation for K_1 , only 3600 and 1800 rpm having explicitly appeared in the charts. NPSH plays a role in the determination of Q_{min} —through the factor K_3 —because of the exacerbation of the unsteadiness and pressure pulsations due to dilation of the cavities in the two-phase internal flows that

TABLE 16 General method for computing minimum flow: quantifying the energy-level effect on $Q_{\min}$ relative to Q_R^{70} **A) Theory**

$Q_{\min} = Q_R \cdot K_1 \cdot K_2 \cdot K_3 \cdot K_4 \cdot K_5$, or Q_R , whichever is smaller (i.e., product of K's ≤ 1)

$$Q_R = \begin{cases} Q_{SR} & \text{if } D_e \geq 0.5 D_2 \\ Q_{DR} & \text{if } D_e \leq 0.5 D_2 \end{cases}$$

$$Q_{DR} = \pi D_2^2 \cdot b_2 \cdot \frac{\Omega}{2} \cdot \overline{C_{DR}}$$

$$\overline{C_{DR}} = -0.01 + \beta_{f,2} / 218.75$$

$$Q_{SR} = \pi \Omega r_e^3 \left(1 - \frac{D_s^2}{D_e^2} \right) \cdot \overline{C_{SR}}$$

$$\overline{C_{SR}} = \tan \beta_1 \cdot \left\{ 1 - 0.2091 \cdot (\beta_1 - 9.5)^{0.4} \right\}$$

$$K_1 = \left[0.06862 \times \left(\frac{N-1800}{1700} \right) \right] + Y + 0.3843 \log_{10} Q_{BEP}$$

$$Y = \begin{cases} -0.4529 & \text{for } Q_{BEP} \text{ in gpm} \\ +1.1612 & \text{for } Q_{BEP} \text{ in m}^3/\text{s} \end{cases}$$

K_2 = Specific gravity

$$K_3 = \begin{cases} 0.6 + 0.4 \times \left[\frac{1}{3} \left(\frac{4}{R} - 1 \right) \right]^{1.1} & \text{Pump with shaft through eye} \\ 0.7 + 0.3 \times \left[\frac{1}{3} \left(\frac{4}{R} - 1 \right) \right]^{1.1} & \text{End-suction pump} \end{cases} \quad R = \frac{NPSH_A}{NPSH_{3\%}}$$

$$K_4 = \begin{cases} 1 & \text{Continuous operation at minimum flow} \\ 0.7 & \text{Operation at minimum flow less than 25\% of total operating time} \end{cases}$$

$K_5 = 1$ No specific mechanical design margins

B) Results for Sample High-Energy Pump Suction Stage (See Fig. 33)

$$Q_R = Q_{SR}, \text{ since } D_e \geq 0.5 D_2$$

$$Q_{SR} = \pi \Omega r_e^3 \left(1 - \frac{D_s^2}{D_e^2} \right) \cdot \overline{C_{SR}} = \pi \times 4700 \cdot \frac{\pi}{30} \times \left(\frac{6.6855}{12} \right)^3 \cdot \left(1 - \frac{2^2}{3^2} \right) \times \overline{C_{SR}} \cdot \frac{\text{ft}^3}{\text{sec.}}$$

$$\text{and } \overline{C_{SR}} = 0.3 \cdot \left\{ 1 - 0.2091 \cdot (16.7 - 9.5)^{0.4} \right\} = 0.1618$$

$$Q_{SR} = 24.03 \text{ ft}^3/\text{sec} = 10,785 \text{ gpm } (0.6804 \text{ m}^3/\text{s}) = 0.540 \times Q_{BEP}$$

$$K_1 = 1.317; K_2 = 0.89; K_3 = 0.768; K_4 = 1; K_5 = 1$$

$$Q_{\min} = 10,785 \times 0.9002 = \underline{9,709 \text{ gpm } (0.613 \text{ m}^3/\text{s})} = 0.485 Q_{BEP}$$

exist for any value of NPSH that is less than the inception $NPSH_i$. (See the ensuing cavitation discussion and the description in Section 2.1.3 that accompanies the definition of NPSH-limits.)

As an example, the minimum flow of the high-energy pump suction stage of Table 14 and Figure 33 is computed in Part B of Table 16. (The value of R for the calculation of K_s is taken from Table 18.) This machine has the relatively high energy level of some of the boiler feed pumps shown in Figure 32 and listed in Table 13. Thus it is not surprising that the resulting value of Q_{min} is 90% of Q_R (which equals Q_{SR} , as is typical) and 48.5% of the BEP flow rate. This is indicated on Figure 37, in which the computed performance curves of this pump are displayed. (See Part B of Table 14 for the main elements of these performance predictions.) However, some of the remedies that have been discussed in this section to counter the adverse effects of high-energy pump phenomena can reduce Q_{min} to lower values than this. In fact, the method contains the flexibility to take into account such improvements through the factor K_s .

In computing Q_{SR} , the angle β_1 is called for. This is approximated by the inlet flow coefficient $\phi_e = 0.3$ from Table 14, which corresponds to a nominal inlet tip flow angle of 16.7 degrees. The actual flow angle (and blade angle) at that location is slightly larger due to the hub-to-shroud variation of the incoming meridional velocity as assumed in the development of the inlet velocity diagrams of Figure 33. This will have a small effect on the results, depending on how one interprets " β_1 ". In this regard, as can also be appreciated from the choices that must be made for the K 's, the method of Table 16 is not precise; however, it is a useful indicator of what the user and designer can expect in determining the operating range of a pump.

The above general theory for computing minimum flow is inclusive of all types of centrifugal pumps and has found application especially for high-energy pumps, which can be difficult to evaluate precisely in the varying circumstances of installation and operation in which they are usually applied. The judgments that must be made in order to apply the

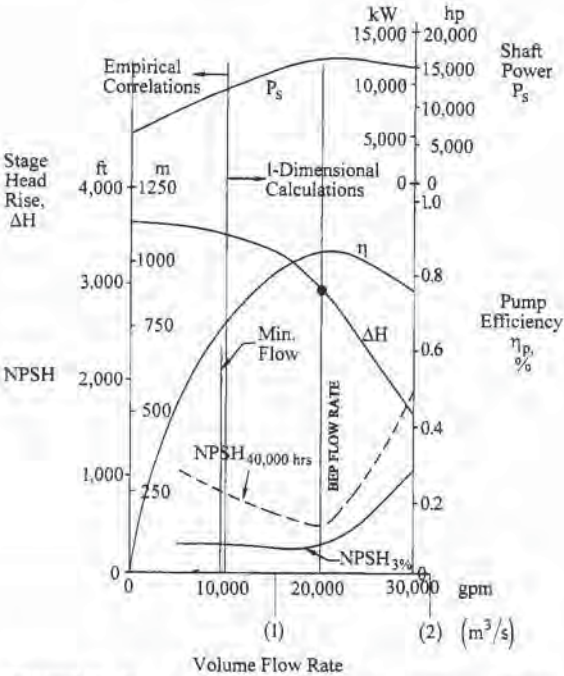


FIGURE 37 Estimated performance of high-energy pump suction stage

method of Table 16 can be largely avoided if actual data for $Q_{\min}$ are available. It would in fact be a monumental task to establish precise limits for all pump types and operational envelopes. Nevertheless, the effect of energy level on $MCSF$ has been found experimentally for several classes of API process pumps through extensive testing. From this, Heald and Palgrave have developed a method for computing the minimum flow of these pumps as follows⁷¹:

$$Q_{\min} = MCSF = \frac{K_7}{100} K_M \times Q_{BEP} \quad (68)$$

where

$$K_7 = K_7(\text{rpm, configuration, } N_{ss}) \quad (69)$$

and

$$K_M = K_M[(NPSH_A/NPSH_R)_{BEP}, \text{ fluid}] \quad (70)$$

Obtained from Figure 38, the factor K_7 accounts for the effects of speed and configuration. As used in that chart—and in Eq. 69, the term *suction-specific speed* N_{ss} (or Ω_{ss}) does not mean that NPSH-effects are also included; rather, this expresses the fact that the impeller design is affected by the designer's choice of N_{ss} as can be seen in the case of the earlier *Design Example*, beginning in Table 6. This leads immediately in Table 7 to the inlet flow coefficient ϕ_e (through the NPSH-correlations of Table 1) and the impeller eye diameter D_e or radius r_e . Following on from the discussion of *Recirculation*, the ratio Q_{SR}/Q_{BEP} can be expected to increase with eye size or D_e/D_S . Being strictly based on experiments, the Heald and Palgrave method does not deal explicitly with Q_{SR} or Q_{DR} , but one can see from Figure 38 that this principle is operative: Higher- N_{ss} pumps require greater $MCSF$. Before this fact was clearly understood, the trend was to design for greater N_{ss} —in order to increase suction capability of these pumps. When operators ran them back to the same low flows as they had done with earlier pumps designed for lower- N_{ss} capability, failures became epidemic. This led to a call for “lower-suction specific speed

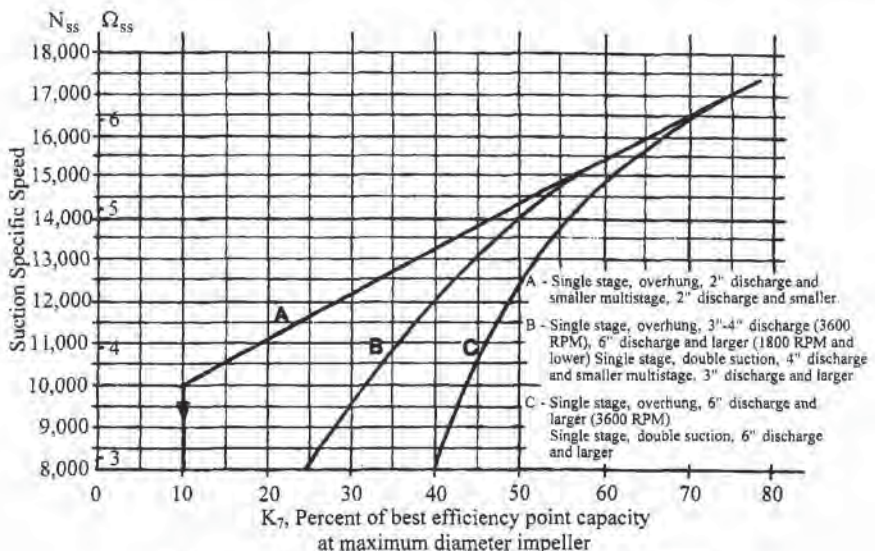


FIGURE 38 Minimum continuous stable flow (MCSF) for process pumps⁷¹

pumps," or, more accurately, pumps designed for lower- N_{ss} capability; specifically 11,000 ($\Omega_{ss} = 4$) or less⁷².

The NPSH-effect does come into play—as with K_s in the previous general method—through the factor K_M in Figure 39a. In this case, the term $NPSH_R$ means $NPSH_{3\%}$. Here again, it is the ratio $R = NPSH_A/NPSH_R$ that is the determining factor because it is a measure of the cavitation activity that is invariably present in the first stage of a pump, and therefore in all single-stage pumps. The lower the value of R , the more violent the pressure pulsations accompanying the recirculating fluid and the consequent vibration. The situation is mitigated considerably at greater R , and K_M operates through Eq. 68 to decrease the MCSF. Of course, if R were great enough—often 5 or more— $NPSH_A$ would exceed $NPSH_R$ (as discussed in Section 2.1.3) and there would then finally really be no cavitation in the pump. (That high a value of $NPSH_A$ is rare; for, if it were supplied, there might not be a need for a pump in the first place.)

Figure 39a also brings in the effect of the liquid being pumped. When room temperature water boils (cavitates), the mass boiled off by the local drop below the vapor pressure makes considerably more cavity or bubble volume than some other liquids, namely hydrocarbons and hot water¹⁷. (See Section 2.1.3.) Figure 39 refers to room temperature or cold water. Figure 39b shows the value of $NPSH_{40}$; so named, because, if this amount of NPSH, which is greater than $NPSH_{3\%}$, is supplied, the impeller thereby lasts for 40,000 hours in resistance to cavitation erosion, a phenomenon which is dealt with below.

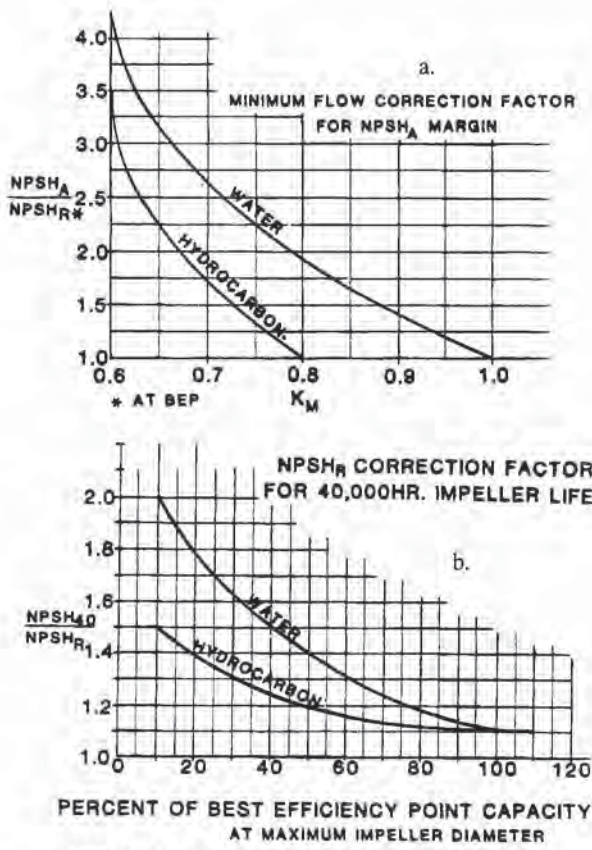


FIGURE 39 NPSH-effects on a) MCSF and b) impeller life¹⁷

The relationship of the ratio R to the design value of N_{ss} has been quantified more generally in terms of the "suction energy" $SE = D_e(in.) \times N \times N_{ss} \times sp.gr.$ by Budris, wherein he calls for a greater NPSH-margin ($= NPSH_A - NPSH_R$) for pumps with high values of SE , this margin increasing as the flow rate is reduced below BEP.⁷³ SE and the pump energy level are connected through the ratio D_1/D_2 (Eq. 51), because the limiting stage pressure rise of Figure 32 is proportional to $sp.gr. \times \Delta H_{sig}$, the stage head being found from D_2 and N , e.g., through the equation in Figure 12.

By way of illustration, computing the minimum flow of the end-suction volute pump of the *Design Example* begins with Curve B in Figure 38 (for pumps with 6-inch discharge and larger and for 1800 rpm and lower as stated on the figure). [This pump has a discharge port of about 9 inches (229 mm) as would be the case if the velocity in this port were half of the throat velocity V_T in Part A of Table 11, as suggested previously in the paragraphs on *Volutes* under *Designing the Collector*.] N_{ss} for this pump was chosen as 12,300 ($\Omega_{ss} = 4.5$), which yields 41.5% for K_T . If it were decided to provide this cold-water pump with 16.4 ft (5 m) of $NPSH_A$, R would be 1.17, and the figure yields 0.97 for K_M . [$NPSH_{3\%} = 14$ ft. (4.27 m) for this pump, as seen in Table 6.] Thus, from Eq. 68, this pump has an $MCSF$ of 0.415×0.97 or 40% of Q_{BEP} . Had it been designed for $N_{ss} = 11,000$ ($\Omega_{ss} = 4.025$), and if R still were 1.17, the $NPSH_A$ would have been 19 ft (5.8 m) and $MCSF$ would have been 0.36×0.97 or 35% of Q_{BEP} . Moreover, if it were pumping hydrocarbons at this same $NPSH_A$, $MCSF$ would have been even lower, namely 0.36×0.78 or 28% of Q_{BEP} . It would appear, though, that for many applications, the pump as designed (at $\Omega_{ss} = 4.5$) has a low enough energy level to allow for an adequate range of flow-rate capability, and that therefore, the value $N_{ss} = 12,300$ is in this case not excessive.

Cavitation Considerations Having alluded to and treated the subject of cavitation a number of times in this section, we should expand on the role that this ever-present phenomenon plays in the operation and durability of centrifugal pumps, particularly those having a high energy level. The manifestations of cavitation that are encountered and become issues for the operability and life of a pump are a) cavitation accompanying backflow from the impeller eye, b) cavitation-generated instabilities and pressure pulsations, and c) erosion, which involves the prediction of the $NPSH_R$ -versus-flow rate characteristic curve to maintain life and, conversely (d) the prediction of life for a given $NPSH_A$. The range of NPSH over which cavitation occurs within a pump extends from the point where pump head or pressure rise undergoes an identifiable drop—usually 3% for repeatable results for the NPSH-value involved—namely the "performance-NPSH" or $NPSH_{3\%}$ —upwards. As NPSH increases from this point, there is an extensive range over which no observable performance loss is detected, yet erosive damage is progressing at a sometimes excessive rate. Finally, at the upper end of the range, all two-phase activity ceases—all bubbles and cavities are suppressed—namely, at the inception-NPSH value or $NPSH_i$. As stated previously, $NPSH_i$ has been observed to be typically about five times $NPSH_{3\%}$. This is clearly described in Section 2.1.3, in which the different NPSH-limits are distinguished. The following cavitation considerations pertain to this range.

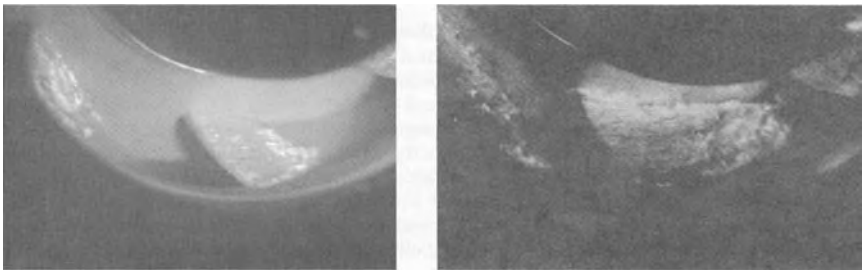
a) *Cavitation and backflow.* The minimum flow limits imposed by the R -value or NPSH-effect arise partly because a lower flow rate at a low NPSH—although it may be in excess of $NPSH_{3\%}$ —involves a strong interaction between suction recirculation and cavitation that can be intense, especially for inducers and large-eye (large D_e/D_2) impellers. At flow rates $Q \ll Q_{SR}$, there exists upstream of the impeller an annulus of back-flowing fluid that emerges from the impeller. Drawing the velocity diagram at the impeller leading edge at the shroud for the case of reversed flow reveals that the absolute velocity component V_i is mostly circumferential and ii) is greater than the impeller tip speed U_{t1} or U_e . Thus the fluid leaving the impeller hugs the outer wall of the approach passage. If this passage is an axial pipe supplying an end-suction impeller or inducer, the pressure along the centerline can be below the vapor pressure, thus creating a vapor core that extends many diameters upstream. Obviously the energetic backflow has to be balanced by an equivalent inflow that enters the impeller from the interior of the pipe and therefore along the hub streamline of the impeller. Vapor from the core is drawn in also and tends to fill the

impeller and vapor-lock it. At this point, the pressure to drive the highly spinning liquid upstream is nonexistent and the backflow ceases. The vapor core disappears and the impeller once more begins ingesting liquid, the process just described repeating itself at a very low frequency (as low as 1 to 6 Hz) and called cavitation surge⁷⁴. It has been found possible to passively divert the backflowing liquid outward from the inlet passage at the impeller or inducer eye into a series of vaned passages surrounding the inlet pipe, the vanes deswirling the backflowing liquid and returning it to a point or annular port upstream. Properly designed, this “backflow recirculator” has completely eliminated all cavitation instabilities in inducers—over the full range of flow rate from shut-off to run-out. It was shown to work for some high- N_{ss} impellers to which it was applied⁷⁴. Cavitation surge, however, is rarely seen in low- N_{ss} impellers and is usually completely avoided by running the pump at flow rates greater than minimum flow Q_{min} as established by one of the previous methods or by test.

b) Cavitation-related instabilities and pressure pulsations. At flow rates greater than Q_{min} , cavitation can cause or intensify pressure pulsations. Such instabilities are connected with the variety of cavity and bubble configurations that can exist over the range of flow rates and NPSH- or R -values. The nature and extent of cavitating flow within a pump has been studied extensively by visual observation. Figures 40 and 41 are laboratory photographs⁶¹ of a sector of a boiler feed pump impeller eye, in which the suction sides of some of the blades are visible. These were taken with the aid of a bright flash that lasted for 1 μs . They can also be found in Ref. 24 of Section 2.1.3. Figure 40a shows the sheet cavity that exists at BEP flow rate and at an R -value of 2. Also observed at BEP—but at $R = 1$ (that is, $NPSH = NPSH_{3g}$)—is the thick, extensive cavity of Figure 40b. This cavity extends from the blade leading edge to the throat formed by the leading edge of the following blade, and this same pattern exists on every blade. Most observers note that when this suction-side cavity reaches the next throat, the pressure rise or liquid head starts breaking down—which is what is recorded at this “3-percent-NPSH” point. Instabilities tend to be at a minimum at both of these relatively steady flow conditions.

Figure 41, on the other hand, displays highly unsteady cavity flows. Both photographs (A and B) were taken at the same half-flow condition at different instants—for $R = 2$. A similar sequence of three photographs at this condition appears in Section 12.5, and none of the three possesses a cavity pattern that resembles either of the others. The position of the somewhat smaller cloud in Figure 41b that has broken off the main cavity appears to be traveling toward the pressure-side (out of sight) of the next blade. Pressure-side cavitation erosion would be the result, and this accords with the findings in Section 2.1.3.

Pressure pulsations are associated with unsteady cavitation patterns, and plots of the amplitude and frequency of suction pressure pulsations show increasing frequency as R is increased—as would be expected as bubble size is reduced. The amplitude peaks at R



a.

b.

FIGURE 40 Cavities on impeller blades at BEP flow rate: a) $NPSH = 2 \times NPSH_{3g}$; b) $NPSH = NPSH_{3g}$ (Source: Flowserve Corporation)⁶¹

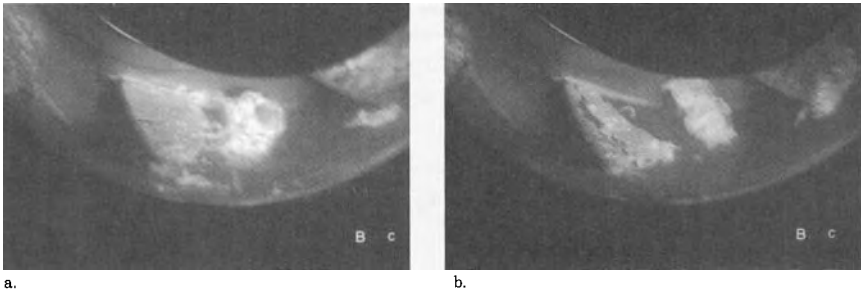


FIGURE 41A and B Cavities on impeller blades at half the BEP flow rate: $NPSH = 2 \times NPSH_{gr}$ (Source: Flowserve Corporation)⁶¹

greater than 1—usually closer to 2. Elimination of these instabilities is best done by avoiding cavitation. This can be done by increasing $NPSHA$ and optimizing the blade shape for minimum cavity activity⁷⁵. To avoid the excessive pump and system oscillations that can occur, some users specify that a cavitation-flow visualization test be conducted and require that minimal or no cavities shall be observed.

c) Erosion due to cavitation and the prediction of the Life-NPSH. In the majority of cases cavitation is unavoidable, and the issue becomes the life of the component (usually the impeller) in resistance to attack by collapsing bubbles and larger cavities as they are swept out of the low-pressure regions near the blade leading edges. The pressure created at the point and instant of collapse is immense. Photographs of the erosive damage resulting from this activity can be found in Sections 2.1.3 and 12.5. The surface failure mechanism is one of fatigue due to repeated collapse of bubbles adjacent to the blade. This happens at a large number of closely spaced sites, the resulting erosion having a strongly pitted texture. The rate of erosive depth penetration into the surface of a venturi subject to bubble collapse was found by Knapp⁷⁶ to increase as the sixth power of velocity V at a constant cavitation number $k = 2(p_1 - p_v)/\rho V^2$. In other words, the erosion rate increased as the third power of $(p_1 - p_v)$ or $NPSH$ at the inlet of the venturi, which means that the collapse pressure rises with $NPSH$, as is known from the Rayleigh bubble collapse theory.⁷⁷ The velocity V in Knapp's venturi corresponds in a pump to the maximum relative velocity W at inlet—which is conveniently represented by the impeller inlet tip speed $U_{t,1}$ as a criterion for damage rate—and τ corresponds to the cavitation number k . For a given impeller at constant Q/N and a constant value of τ , (or at a constant available suction specific speed,) the higher $U_{t,1}$ is, the greater the $NPSH$ and the greater the damage rate. Also, as mentioned earlier, a higher value of $U_{t,1}$ implies a correspondingly greater U_2 and, therefore, greater pump pressure rise or head. Therefore, high-head pumps are more likely to suffer from cavitation erosion, making cavitation a “high-energy” pump phenomenon. High pressure-rise has already been shown to be a feature of high energy pumps through the definition of energy level in terms of classical stress loading (Figure 32).

Facing the reality of high-energy pump destruction due to cavitation erosion, Vlamming redefined the term “ $NPSHR$ ” to mean $NPSH_{40,000 \text{ hrs.}}$; that is, the $NPSH$ needed to limit the damage sufficiently so as to ensure an impeller life of 40,000 hours. He developed an empirical method for predicting the curve of this “damage- $NPSH$ ” versus flow rate for conventionally-designed impellers⁷⁸. This method is defined in Table 17 and includes the inlet tip speed effect on the erosion rate. A value of the vaporization factor C_v that is less than unity applies to hot water and to other liquids such as hydrocarbons, which generate far less vapor volume when they cavitate than does cold water. (See the earlier discussion under *NPSH Effects* in the section on *Specific Speed and Optimum Geometry*.) (Rather than depend on theory for the API process pumps mentioned

TABLE 17 NPSH required for 40,000 hours life⁷⁷

$$NPSH_R = NPSH_{SE} + f \left(\frac{Q}{Q_{BEP}}, \frac{Q_{BEP}}{Q_{SE}}, NPSH_{SE} \right)$$

where

$$NPSH_{SE} = C_a \cdot C_b \cdot C_c \cdot \left[(k_1 + k_2) \phi_e^2 + k_2 \right] \cdot \frac{U_e^2}{2g}$$

and C_a = Specific Speed Effect (= 1 for $N_s \geq 1000$) ($\Omega_s \geq 0.366$)
 C_b = Vaporization Effect [= 1 for cold water; 0.74 for 350°F (177°C) water]
 C_c = Material Effect (= 1 for 12 Cr. Stainless Steels)
 k_1 = 1.2

$$k_2 = 0.2334 + \left\{ \frac{[U_e \text{ (ft/sec)/400}]^4}{[U_e \text{ (m/s)/128.3}]^4} \right\} = \begin{cases} \text{inlet tip} \\ \text{speed effect} \end{cases}$$

$$f = \begin{cases} \left[887 \left(\frac{Q_{SE}}{Q_{BEP}} - \frac{Q}{Q_{BEP}} \right) + 893 \left(\frac{Q_{SE}}{Q_{BEP}} - \frac{Q}{Q_{BEP}} \right)^2 \right] \cdot X \left\{ \frac{Q}{Q_{SE}} \leq 1 \right. \\ \left. \left[2820 \left(\frac{Q}{Q_{BEP}} - \frac{Q_{SE}}{Q_{BEP}} \right) + 6610 \left(\frac{Q}{Q_{BEP}} - \frac{Q_{SE}}{Q_{BEP}} \right)^2 \right] \cdot X \left\{ \frac{Q}{Q_{SE}} \geq 1 \right. \right. \end{cases}$$

$$\text{and } X = (S^{1.105} - S) / 1000 \quad \text{where } S = \begin{cases} NPSH_{SE} \text{ (ft)} \\ 3.2808 \times NPSH_{SE} \text{ (m)} \end{cases}$$

earlier in connection with their minimum flow data, Heald and Palgrave quantified $NPSH_{40,000 \text{ hrs}}$ in Figure 39b⁷¹.)

Applied to the high-energy multistage pump suction stage of Figure 33 and Table 14, Vlamings's method yields the $NPSH_R$ -values shown in Table 18 under the heading $\phi_e = 0.3$. These are plotted as the $NPSH_{40,000 \text{ hrs}}$ -curve on Figure 37. The blades of the pump are assumed to be set for zero incidence of the incoming flow to their camber lines at the BEP—called “shockless entry” and denoted by “SE.” (Many designers make Q_{SE} somewhat larger than Q_{BEP} to achieve lower $NPSHR$ at $Q > Q_{BEP}$. Also, Q_{SE} produces an even greater jump in the mean W at the leading edge than was shown in Figure 29 for the 2.2 degrees of mean positive incidence seen in Figure 26a. This is due to the blockage of the blade that is fully realized at this zero-incidence flow rate Q_{SE} . The impeller flow loss at this flow rate often causes Q_{BEP} to be about 10% less than Q_{SE} .) The column to the right in the table contains the results for the same method applied to the more common case for high-energy pump suction stages, namely $\phi_e = 0.25$ (and lower). For the same shaft diameter, flow rate and speed, $\phi_e = 0.25$ yields a larger eye diameter through Eq. 49; namely, 13.92 in. (353.6 mm) versus 13.37 in. (339.6 mm) for the $\phi_e = 0.3$ case as shown in Figure 33. Moreover, that figure shows the inlet tip speed $U_{t,1}$ (U_e) to be 274 ft/sec (84 m/s), whereas for the larger-eye case the value is computed to be 286 ft/sec (87 m/s). As Table 18 reveals, this larger-eye impeller has a lower value of $NPSH_{3\%}$ (as computed from the correlations of Table 1), which is the reason for sizing large eyes. But Vlamings's method indicates that the smaller eye requires less “damage-NPSH” ($R = 1.69$) than does the larger eye ($R = 2.06$), even though the latter requires less “performance-NPSH” or $NPSH_{3\%}$. So, from both a minimum-flow and a cavitation damage standpoint, the advantage of a smaller eye for high energy levels is evident.

TABLE 18 *NPSHR* of high-energy pump suction stages**A) *NPSH* Required to Maintain Head Rise: $NPSH_{3\%}$**

$NPSH_{3\%}$, ft(m) (3% reduction in ΔH)	$\frac{Q}{Q_{BEP}}$	$NPSH_{3\%}$, ft(m)	
		$\phi_c=0.3^*$	$\phi_c=0.25$
$k_1 = 1.69$ $k_2 = 0.102$ } [Table (1)]	[100]	0.2633	0.214
[Empirically Calculated]	100	307(94)	271(83)
	150	1150(350)	1150(350)

B) *NPSH* Required to Limit Cavitation Damage: $NPSH_{40,000 \text{ hrs.}}$

$NPSH_{40,000 \text{ hrs. Life}}$	$\frac{Q}{Q_{BEP}}$	$NPSH_R$, ft(m)	
		$\phi_c=0.3^*$	$\phi_c=0.25$
Vlamings' } Empirical } Method } (for $Q_{SE} = Q_{BEP}$)	0.25	1081(329)	1174(358)
	0.50	840(256)	910(277)
	0.75	653(199)	705(215)
	0.90	566(173)	610(186)
<div style="border: 1px solid black; padding: 5px; display: inline-block;"> $R = \frac{NPSH_{40,000 \text{ hrs.}}}{NPSH_{3\%}}$ </div>	1.00	519(158)	559(170)
	[1.00]	R=1.69	R=2.06
	1.10	687(209)	743(226)
	1.25	1058(322)	1148(350)
	1.50	1994(608)	2173(662)

*Values predicted for sample high-energy pump suction stage: $\phi_c = 0.3$; $\phi_c \leq 0.25$ widely typical of suction (1st) stages.

d) Prediction of life for a given $NPSH_A$. The inverse of the foregoing problem is how to determine the life under cavitating conditions at a given value of available $NPSH$. Gülich found a connection between an observed length L_{cav} of the cavity trailing off the leading edge of the blade at BEP (Figure 40a) and the life of an impeller operating at this condition⁷⁹. The resulting procedure is outlined in Table 19, beginning with the definition that the life is the time it takes for the erosion to penetrate through 75% of the blade (or wall) thickness. In the absence of a cavitation-visualizing test, L_{cav} can be estimated as suggested by the last two formulas in Table 19⁷⁵. Critical to this estimate is the assumed value of the "inception- $NPSH$ " or τ_i . For conventionally designed impeller blades, $\tau_i \approx 1$ at the BEP, whereas aerodynamically shaped blades that minimize the local reduction of static pressure have been produced with $\tau_i < 0.5$ at the BEP^{75,90}. The life computations of Table 20 follow from application of the method of Table 19 to the sample suction stage of Figure 33 for $\tau_i = 1$ and 0.5. Higher values of τ_i apply for $Q \neq Q_{BEP}$ —as might be expected from the shape of Vlamings' $NPSH_{40,000 \text{ hrs.}}$ -curve in Figure 37 and the fact that the local pressure reduction in the leading-edge region increases with incidence.

TABLE 19 Prediction of life under cavitating conditions

Life = the minimum of (Life_{s.s.}, Life_{p.s.}),

where s.s.= suction side of blade;

and p.s. = pressure side of blade.

And where

$$\text{Life} = 0.75 \times t / \text{MDPR},$$

where t = blade thickness, mm;

MDPR=mean depth of penetration rate, mm/hour.

$$\text{MDPR} = \left[C \times (L_{\text{cav}}/10)^n \times (\tau_A - \phi_e^2)^3 \times U_e^6 \times \rho_L^3 \times A \right] / \left[8 \times F_{\text{mat}} \times (\text{TS})^2 \right] \text{ mm/h}$$

Where C = 8.28 E-06 (s.s.), 396 E-06 (p.s.); and n = 2.83 (s.s.), 2.6 (p.s.);

$\tau = \text{NPSH} / (U_e^2 / 2g)$; dimension less

U_e is blade inlet tip speed in m/s

$\phi_e = V_{m,1} / U_e$, where $V_{m,1}$ = average axial inlet velocity, (typically $\phi_e = 0.25$);

$\rho_L = 1000 \times \text{sp. gravity of the liquid}$; A = 1 for cold water, 0.705 for 350°F (177°C) water;

$F_{\text{mat}} = 1$ for martensitic: stainless steel, 1.7 for austenitic stainless steel;

TS = tensile strength in Pa [= 860,000,000 (= 125,000 psi) for 13 Cr and
= 552,000,000 (= 80,000 psi) for 18 Cr, 8 Ni.]

$$L_{\text{cav}} \approx (\pi \times D_e / n_b) \times \left\{ 1 - [(\tau_A - \tau_{3\%}) / (\tau_i - \tau_{3\%})]^{1/3} \right\} \text{ if } \tau_A < \tau_i \text{ (mm)}$$

$$L_{\text{cav}} = 0 \text{ if } \tau_A > \tau_i$$

From this and the results of Table 20, it is evident that improvements to conventional blade-design practice are essential if life is to exceed half a year. Further, this life calculation method is based on the existence of a sheet cavity like that of Figure 40a; and the disordered cavity structures of Figure 41, which happen in the presence of recirculation, are not addressed. Low-flow cavitation damage is generally more severe, and is described in Section 2.1.3. Moreover, corrosion can play a role in cavitation-related erosive activity, an effect that was also addressed in Gülich's work.⁷⁹ Nevertheless, the ability to compute erosive behavior—even at the BEP—allows one to evaluate design improvements and provides a good idea of the life that can be expected under normal operating conditions.

Obviously, if the available NPSH is greater than the inception-NPSH (that is, $\tau_A > \tau_i$), there is no bubble activity of any kind, and at Q_{BEP} the cavity length $L_{\text{cav}} = 0$. An illustration of what can be achieved in this regard is presented in Figures 24 and 25 of Section 12.5, which are the “before” and “after” photographs of the model impeller blades for the first stage of a 24,000 hp (18 MW) pipeline pump. Quasi-three dimensional analysis of the blade pressure loading led to changes in shape that eliminated the cavities^{83,75}. This complete absence of cavitation is becoming the desired objective for the design and application of new and upgraded multistage high-energy pump suction stages. This approach eliminates both the erratic mechanical behavior that occurs in response to unsteady cavity patterns and the erosion due to bubble collapse; thereby substantially increasing the life and reliability of such machines⁸¹.

TABLE 20 Life of high-energy pump suction stage

A) Pump data for computing life (at the B.E.P. of 20,000 gpm)

- $\text{NPSH}_{3\%} = 307 \text{ ft (94 m)}$; thus, $\tau_{3\%} = 0.263$
- $\text{NPSHA} = 519 \text{ ft (158 m)}$; thus, $\tau_A = 0.444$
- $t_{\text{blade}} = 0.42 \text{ in (10.7 m)}$; thus, $F_{\text{mat}} = 1$ (13 Cr martensitic stainless steel)
- Suction-side cavitation in 350°F boiler feedwater, for which $A = 0.705$
- Inlet flow coefficient $\phi_e = 0.3$; $U_e = 274.2 \text{ ft/sec (83.57 m/s)}$; $U_e^2/2g = 1168 \text{ ft (356 m)}$

B) Results for inception occurring at $\text{NPSH}_i = \text{blade inlet tip speed head}$ and half that NPSH_i .

$$\begin{aligned} \text{a) } \tau_i &= 1.0 = 3.8 \times \tau_{3\%} \\ \text{NPSH}_i &= 1168 \text{ ft (356 m)} \\ L_{\text{cav}} &= 57.0 \text{ mm} \\ \text{MDPR} &= 0.0014482 \text{ mm/h} \\ \text{Life} &= 5541 \text{ h} = 7.6 \text{ months} \end{aligned}$$

$$\begin{aligned} \text{b) } \tau_i &= 0.5 = 1.9 \times \tau_{3\%} \\ \text{NPSH}_i &= 584 \text{ ft (178 m)} \\ L_{\text{cav}} &= 13.1 \text{ mm} \\ \text{MDPR} &= 0.0000226 \text{ mm/h} \\ 355,000 \text{ h} &= 40 \text{ years} \end{aligned}$$

C) Sample Life Calculation * $\left\{ \begin{array}{l} \text{High Energy} \\ \text{Pump Suction Stage} \end{array} \right.$

$$L_{\text{cav}} = 25.4 \times \frac{\pi \times 13.37}{7} \times \left[1 - \left(\frac{0.444 - 0.263}{1 - 0.263} \right)^{1/3} \right] = \underline{57.0 \text{ mm}}$$

$$\text{MDPR} = \frac{8.28 \times 10^{-6} \times \left(\frac{57}{10} \right)^{2.83} \times (0.444 - 0.3^2)^3 \times (83.57)^6 \times (890)^3 \times 0.705}{8 \times 1 \times (860 \times 10^6)^2}$$

$$= 0.0014482 \text{ mm/h} \left(\times 8760 \frac{\text{h}}{\text{year}} = 12.7 \text{ mm/y} \right)$$

$$\begin{aligned} \text{Life} &= 0.75 \times 10.7/0.0014482 &= \underline{5541 \text{ h}} \\ & &= \underline{7.6 \text{ months}} \end{aligned}$$

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2.1.2 CFD ANALYSIS OF FLOW AND PERFORMANCE

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INTRODUCTION

Until relatively recently, prediction of flow behavior in pumps and the systems in which they operate has generally been limited to a practical combination of (a) obtaining, applying, and scaling experimental data on components and models, (b) modifying components according to rules that were developed for specific geometries but were often extrapolated to others, and (c) one- and two-dimensional computations by which designs of components were made and which could be used to predict flows and performance in a reasonable period of time. With respect to the pumps themselves, in Section 2.1.1, “Centrifugal Pump Theory,” fluid dynamical analysis of the flow through the internal passages is demonstrated as the means for predicting the performance of pumps. Formulas are given for energy transfer, static pressure development, and flow losses, which can be applied one dimensionally to evaluate the effectiveness of the design of the impeller and the accompanying stationary inlet, volute, diffuser—and the crossover passages of multistage pumps. In several instances, one-dimensional (1D) analysis is found to be inadequate—as for example, in finding the distribution of velocity in the approaching fluid along the leading edge of an impeller. Here, the need for a two- or three-dimensional solution of the basic fluid dynamical equations of motion is called for, which now is more and more accurately and rapidly achieved on a digital computer or cluster(s) of computers as ever-increasing computing power has become available. Hence the term “Computational Fluid Dynamics” or “CFD” has become commonplace in the quest for understanding the flow behavior and patterns as needed to achieve optimum design and matching of flow elements. An accurate solution of the flow fields in the components from the inlet to the discharge port of the machine also yields the overall “hydraulic” performance; namely, head, power, efficiency—for both single- and multiphase flows. Moreover, as described in Chap. 13, flows in system

elements, especially intake sumps and suction piping, play a crucial role in the behavior of both pump and system. The design and deployment of such elements and especially the combinations of them that can include the pump as well are among the increasing list of CFD applications.

Yet, all this computational capability is not a substitute for the experienced and creative designer, who must provide the hydraulic geometries and fluid systems to be evaluated and interpret the results. For example, the traditional approach to creating and optimizing the gamut of highly developed impeller shapes versus specific speed—which often produce overall pump efficiencies in excess of 90% as seen in the chart efficiencies of this book—had been accomplished by the middle of the last century and well before digital computers came to be employed in the design and optimization process. Intuitive fluid dynamical reasoning had been exercised by these designers as they assessed both theoretical and experimental results. The former were the output of flow analyses of varying degrees of complexity from 1D to quasi-three dimensional (Q3D—combinations of two-dimensional [2D] solutions) that were needed in order to apply the diffusion limits per NACA research as in Figure 17 and Eqs. 57 to 59 in Section 2.1.1. This traditional approach enabled especially the development of highly efficient impellers, as well as imaginative solutions to new design and performance challenges.^{1,2,3}

Early CFD work on pumps focused on the steady flow field within a single hydraulic component, usually the impeller; the objective being to provide recognizable flow data for judging it. Graf demonstrated that several of the known and observed flow patterns within impellers could be reproduced via CFD.⁴ Used to judging impellers in terms of the Q3D-computed distributions of relative velocity W just outside the boundary layers on the blades, he presented in this same early work the equivalent static pressure loadings on the blade surfaces. Pressures were readily available to him on the actual blade surfaces; whereas $W = 0$ on these surfaces, because CFD solutions include the boundary layers or approximations for them.⁴ Substantial improvements to inlets, diffusers, and multistage pump crossovers are often more readily identified than for the more highly developed impellers.^{5,6} Indeed, many shortcomings of established geometries remain, despite the impressive achievements of generations of past designers. Even the best of these geometries can be and are being further improved. Interactive optimization of the design is thus a major function of CFD in the hydraulic development process.

Accurate prediction of overall performance has been less forthcoming, mainly because of limitations on treatment of the losses accompanying diffusing and separated flows that are imposed by inadequate computational power. As demonstrated by Gülich et al, the ideal or “theoretical” head H_i can be quite accurately calculated at flow rates near the best efficiency point (BEP) with minimum CFD power by present standards, because it is connected primarily with flow directions and velocities in the impeller and less with the details of the loss calculations therein.⁷ The most accurate CFD prediction of overall pump developed head requires analyzing all hydraulic components and in turn correctly modeling the interfaces between rotating and stationary components. For example, the flow distortions produced by an asymmetric inlet passage produce losses in the impeller it feeds,⁵ and an unsteady CFD solution of these coupled flow elements should correctly model these losses.

The computing power required for an unsteady solution of the entire pump has been the limiting factor for many users, as it generally requires multiple processors operating in parallel. Approximations to get around this limitation have been made by restricting the analysis to steady-flow computations—often in a single passage—and, at the interfaces between rotating and fixed components, by (a) using circumferential averages—but not spanwise hub-to-shroud averages—of the quantities being calculated or (b) freezing the impeller at a selected position relative to the fixed adjacent components and doing no such averaging. Neither (a) nor (b) prevents simulation of flows across the interfaces in both the forward and reverse directions as would happen at flow rates below BEP; however (b) preserves variations in all directions. In one typical case, the frozen rotor approach (b) predicted the head at shutoff and at flow rates from 80 to 120% of BEP within 5% of the measured head on test.⁸ In another example, similarly accurate results for complete stages via the circumferential-averaging approach (a) were reported for flow rates in the neighborhood of the BEP only.⁹

With sufficient computing power, improvements in CFD-predicted performance are possible at all flow rates, together with the influence on this performance of rotating stall and cavitation and other multiphase flows. This includes the thermodynamic effects on vaporization and other gases that may be present in the liquid. Finally, the inherently unsteady flow in the meshing rotors of screw pumps and other rotary positive displacement pumps (Sections 3.7 and 3.8) has been investigated by CFD;¹⁰ however, this section concentrates on the use of CFD for flows in rotodynamic pumps and inducers and particularly for understanding the off-design behavior of these machines. From the attendant description of the concept and methods of implementing it for pump internals, readers will readily appreciate the fact that CFD is equally applicable to all flow-related system elements, including intake sumps and suction piping configurations.

NOMENCLATURE

NOTE: See the Nomenclature for Section 2.1.1 for most of the symbols used in this section. Additional definitions are presented here for further clarity and convenience.

k = turbulent kinetic energy $= \frac{1}{2}(\overline{w_r^2} + \overline{w_\theta^2} + \overline{w_z^2})$ where each overbar represents a time average

t = time

u = generic velocity component

W = local mean value of fluid velocity relative to the rotating impeller, ft/s (m/s). For stationary components ($\Omega = 0$), substitute the absolute velocity V for W in the equations presented here.

w' = fluctuating component of the fluid relative velocity due to turbulence, ft/s (m/s). For stationary components ($\Omega = 0$), substitute the absolute velocity v' for w' in the equations presented here.

τ = cavitation coefficient for impeller or inducer $(= 2g \times NPSH/U_e^2 \text{ or } 2g \times NPSH/U_{1,t}^2)$

τ_{xy} = shear stress in the x-direction due to velocity variation in the y-direction

$\mu_i = \mu$ = the absolute viscosity as defined in Section 2.1.1. This quantity is more precisely known as the molecular or "laminar" viscosity.

$\mu_T = \mu_t$ = the turbulent or "eddy" viscosity, which has the same units as μ_i or μ

ε = turbulent energy dissipation rate

ϕ = flow coefficient $(= \overline{V_{m,1}}/U_e)$

ω = fluid vorticity

CFD BASICS

The Reynolds-Averaged Navier-Stokes Equations The basis for all CFD analyses are the Navier-Stokes equations. These are the equations of general, unsteady motion of a compressible viscous fluid, which relate pressure to the instantaneous velocity field. They are derived in fluid mechanics texts by applying Newton's second law, which relates forces and accelerations to those occurring in the three spatial dimensions within the fluid. For example, Schlichting does this by developing expressions for the normal and shear stresses on a fluid element and then invoking Stokes' hypothesis that these stresses are proportional to the rate of strain or shear, which is the case for a Newtonian fluid.¹¹ The simplest form of this hypothesis is the expression for shear stress τ_{xy} in the fluid in

the gap of width δ between two plates, one of these plates moving steadily at velocity U in the x -direction parallel to the other (fixed) plate:

$$\tau_{xy} = \mu \frac{U}{\delta} \quad \text{or, locally in the fluid, } \tau_{xy} = \mu \frac{du}{dy} \quad (1)$$

where μ is the absolute viscosity or constant of proportionality between the stress and the rate of strain du/dy , u is the local fluid velocity, and y is distance in the direction normal to the fixed plate. In this case du/dy is equal to U/δ throughout the gap. The fluid in this illustration is assumed to be laminar, and for this case the Navier-Stokes equations reduce to the following single expression:

$$\mu \frac{d^2 u}{dy^2} = 0 \quad (2)$$

The solution of Eq. 2 is the expected straight-line velocity distribution:

$$u = \frac{U}{\delta} y \quad (3)$$

There are many other solutions in the literature for laminar flows. However, when the flow is turbulent, the quantities involved are fluctuating, and the flow field is best understood in terms of the averages of the velocity components, pressure, and density; where the instantaneous value of any velocity component u is represented as the sum of an time-averaged portion $\bar{u}$ and a fluctuating portion u' ; namely,

$$u = \bar{u} + u' \quad (4)$$

When this expression for each of the velocity components (and similarly for the pressure and density) is substituted into the Navier-Stokes equations, one obtains what are called the *Reynolds-Averaged Navier-Stokes* equations, and so CFD specialists speak of *RANS* codes. For illustrative purposes and the general interest of pump engineers, it is convenient to specialize these *RANS* equations to the incompressible flow case and to render them in terms of the relative velocity components, W_r , W_θ , and W_z in the system of cylindrical coordinates r , θ , and z that are commonly used for analysis within the rotating frame of reference of an impeller. Four equations are involved here, the assumption being that the absolute viscosity μ is constant. The first three are Newton's second law or the momentum equation in each of the three coordinate directions, and the fourth is the continuity equation.

Momentum in the r -direction:

$$\begin{aligned} \rho \left(\frac{\partial W_r}{\partial t} + W_r \frac{\partial W_r}{\partial r} + \frac{W_\theta}{r} \frac{\partial W_r}{\partial \theta} - \frac{W_\theta^2}{r} + W_z \frac{\partial W_r}{\partial z} \right) &= \rho \Omega^2 r + 2\rho \Omega W_\theta - \frac{\partial p}{\partial r} + \frac{1}{r} \frac{\partial}{\partial r} \left(\mu r \frac{\partial W_r}{\partial r} - \rho r \overline{w_r'^2} \right) \\ &- \frac{1}{r} \left(\mu \frac{W_r}{r} - \overline{\rho w_r'^2} \right) + \frac{1}{r} \frac{\partial}{\partial \theta} \left(\mu \frac{\partial W_r}{\partial \theta} - \rho \overline{w_r' w_\theta'} \right) + \frac{\partial}{\partial z} \left(\mu \frac{\partial W_r}{\partial z} - \rho \overline{w_r' w_z'} \right) - \frac{2\mu}{r^2} \frac{\partial W_\theta}{\partial \theta} \end{aligned} \quad (5)$$

Momentum in the θ -direction:

$$\begin{aligned} \rho \left(\frac{\partial W_\theta}{\partial t} + W_r \frac{\partial W_\theta}{\partial r} + \frac{W_\theta}{r} \frac{\partial W_\theta}{\partial \theta} - \frac{W_r W_\theta}{r} + W_z \frac{\partial W_\theta}{\partial z} \right) &= -2\rho \Omega W_r - \frac{1}{r} \frac{\partial p}{\partial \theta} + \frac{1}{r} \frac{\partial}{\partial r} \left(\mu r \frac{\partial W_\theta}{\partial r} - \rho r \overline{w_\theta' w_r'} \right) \\ &- \frac{1}{r} \left(\mu \frac{W_\theta}{r} + \overline{\rho w_\theta' w_r'} \right) + \frac{1}{r} \frac{\partial}{\partial \theta} \left(\mu \frac{\partial W_\theta}{\partial \theta} - \rho \overline{w_\theta'^2} \right) + \frac{\partial}{\partial z} \left(\mu \frac{\partial W_\theta}{\partial z} - \rho \overline{w_\theta' w_z'} \right) + \frac{2\mu}{r^2} \frac{\partial W_r}{\partial \theta} \end{aligned} \quad (6)$$

Momentum in the z -direction:

$$\begin{aligned} \rho \left(\frac{\partial W_z}{\partial t} + W_r \frac{\partial W_z}{\partial r} + \frac{W_\theta}{r} \frac{\partial W_z}{\partial \theta} + W_z \frac{\partial W_z}{\partial z} \right) = & -\frac{\partial p}{\partial z} + \frac{1}{r} \frac{\partial}{\partial r} \left(\mu r \frac{\partial W_z}{\partial r} - \rho r \overline{w'_z w'_r} \right) \\ & + \frac{1}{r} \frac{\partial}{\partial \theta} \left(\mu \frac{\partial W_z}{\partial \theta} - \rho \overline{w'_z w'_\theta} \right) + \frac{\partial}{\partial z} \left(\mu \frac{\partial W_z}{\partial z} - \rho \overline{w'^2_z} \right) \end{aligned} \quad (7)$$

Continuity:

$$\frac{1}{r} \frac{\partial(rW_r)}{\partial r} + \frac{1}{r} \frac{\partial W_\theta}{\partial \theta} + \frac{\partial W_z}{\partial z} = 0 \quad (8)$$

On the left-hand side of Eqs. 5, 6, and 7 appear the fluid acceleration terms—including the unsteady terms (time derivatives) that are evaluated in unsteady CFD solutions of these equations. Because these equations apply to a reference frame rotating at angular speed Ω , the centrifugal and Coriolis terms $\rho\Omega^2 r$ and $2\rho\Omega W_\theta$ appear on the right-hand side of Eq. 5, and the Coriolis term $2\rho\Omega W_r$ on the right-hand side of Eq. 6. By setting Ω equal to zero and substituting the absolute velocity V for the relative velocity W , these equations can be applied to stators, such as vaned or vaneless diffusers.

For simplicity, the bar over each of the time-averaged velocity components has been omitted; however, the bar does appear over each of the six unique products of the fluctuating velocity components, indicating a time average of these quantities, which characterize turbulent flows and are called *Reynolds stresses*. The task for the developers of CFD codes is to properly represent or “model” these six terms.¹²

Turbulence Modeling Schemes Because the RANS equations (5 to 8) deal with the time-mean values of the velocity components, they are more practical to deal with in computing the flow field than are the Navier-Stokes equations, which deal with the instantaneous values. However, relationships for the Reynolds stresses must be supplied in order to obtain a solution to the RANS equations, and generating these relationships is called “turbulence modeling.” These include the earlier algebraic eddy viscosity models and the later, now widely applied two-equation models. With an increasing array of complexity, advanced Reynolds stress models are possible, but they require many additional equations to be solved in order to define all the Reynolds stresses.

- a. *Algebraic eddy viscosity models.* Thinking in terms of the mean velocities, it is logical to try and relate the Reynolds stresses of turbulent flow to the mean-velocity gradients, just as was done in Eq. 1 for the shear stress in a 2D laminar flow.

$$\overline{\rho u'v'} = \mu_t \frac{du}{dy} \quad (9)$$

where μ_t is called the eddy viscosity, and $\overline{\rho u'v'}$ is the relevant Reynolds stress from the RANS equations (as written in cartesian coordinates x, y , and z and therefore in terms of velocity components u, v , and w , respectively). Eq. 9 models the only Reynolds stress involved in the analysis of a 2D turbulent boundary layer of a flow in the x -direction parallel to a fixed wall. Boundary layers were the first area of endeavor in the numerical solution of the RANS equations that eventually came to be called CFD. The eddy viscosity μ_t in a turbulent boundary layer increases with distance from the wall in accordance with Prandtl’s mixing length theory to very large multiples of the molecular viscosity μ . Incidentally, the flow becomes laminar very near the wall, which is the domain of the “laminar sublayer” in which μ alone is operative per Eq. 1. Cebeci and Smith, using algebraic equations for μ_t , were successful in computing all the parameters of 2D turbulent boundary layers.¹³ Similar success was achieved by applying this method to compute the pump disk friction, (see Table 3 of Section 2.1.1,) wherein

the skewed boundary layers on the impeller shrouds and adjacent casing walls required two Reynolds stresses to be modeled at each point in the flow—both in terms of the same value of μ_t at that point.¹⁴ Later this μ_t model was modified by Baldwin and Lomax in order to give it wider application.¹²

- b. Two-equation models for eddy viscosity.** To apply CFD throughout the flow field involves all of the Reynolds stresses appearing in Eqs. 5 to 8, not just the one or two of boundary layer flows. The algebraic eddy viscosity model is inadequate to this task because the eddy viscosity really depends on the characteristics of the turbulence, primarily the length scale or size of the turbulent eddies. Near the wall, in the boundary layer, this length scale is known to increase with distance from the wall per the Prandtl mixing length theory, and this is included in the algebraic eddy viscosity model. Outside the boundary layer, where the flow behavior can range from frictionless to separated and recirculating, two equations are solved to determine the eddy viscosity. The quickest and most useful method of this kind is called the k - ε model. The Jones-Launder version of this model is described by Lakshminarayana and is briefly summarized here.¹² The eddy viscosity is modeled as

$$\mu_t = \mu C_{\mu} e^{\left(\frac{-2.4}{1+5.0}\right)} R_T \quad (10)$$

where R_T is the turbulence Reynolds number

$$R_T = \frac{\rho k^2}{\mu \varepsilon} \quad (11)$$

and where k is the kinetic energy of the turbulence

$$k = \frac{1}{2} \left(\overline{w_r^2} + \overline{w_\theta^2} + \overline{w_z^2} \right) \quad (12)$$

and ε is the turbulent energy dissipation rate or dissipation power due to the action of viscous stresses.

k is found from a kinetic energy (KE) equation derived from the Navier-Stokes equation for each instantaneous velocity component. This KE transport equation equates the convection of the turbulent kinetic energy k to the diffusion of the turbulence due both to the fluctuations of pressure and velocity and to the dissipation.

ε is found from another equation derived from the Navier-Stokes equations. This resulting transport equation equates the convection of ε to the sum of (a) turbulence production, (b) diffusive transport of the turbulence, and (c) viscous destruction of the turbulence.

For incompressible flow, these latter two transport equations for k and ε , together with Eq. 10 for the eddy viscosity are solved simultaneously for the mean velocities, k , ε , and the Reynolds stresses. Finally, with this information, the solution of the *RANS* equations for the turbulent flow field is accomplished.

In summary, it can be seen that the two-equation model is far more involved in both concept and execution than is the algebraic eddy viscosity model. However, this method and some variants of it such as the k - ω model have been remarkably successful in predicting both the velocity fields and the performance characteristic curves of pumps, including head, efficiency and NPSH effects. Moreover its popularity has been the focus of much testing, correlation with experiment and improvement by CFD researchers.

While even more complex turbulence modeling and solution procedures have been and continue to be developed, these require more transport equations to be solved and assumptions to be made about constants and other parameters appearing in them. The “feel” that the simpler and practical models described here give the user—as illustrated by the original eddy viscosity concept—is lost to the practicing pump engineer who wishes to use CFD as a practical tool. In short, the k - ε model remains the quickest and most useful choice.

Solving the Equations The ways of dealing with the Reynolds stress terms in the RANS equations having been described, the next step is to obtain the solution of the systems of equations involved. This cannot be accomplished in closed mathematical form, except in some special cases as was illustrated in Eqs. 1 to 3 for 2D laminar gap flow. Rather, numerical methods are employed to obtain the values of the velocity components and pressure at all desired points in the flow field. The steps in obtaining such numerical solutions are as follows:

1. *Discretization of the equations.* The differential equations; namely, Eq. 5 to 8, as well as the eddy viscosity and transport equations if any, are changed into algebraic equations by expressing each derivative in finite-difference form. For example, the derivative du/dy in Eq. 1 is represented by the ratio of the difference in the velocity u between two points in the flow field to the difference in the position y between those two points, namely one at $y = \delta$ on the moving plate for which $u = U$ and the other at $y = 0$ on the fixed plate where $u = 0$

$$\frac{du}{dy} = \frac{U - 0}{\delta - 0} \quad (13)$$

where the right-hand side of this expression is the finite-difference form of the derivative du/dy in this particular case. However, in general, finite differences are approximations for actual values of the derivatives they represent. Moreover, discretization can have different levels or orders of accuracy depending on the numerical scheme being utilized. Lakshminarayana gives a description of these orders of accuracy.¹²

2. *Meshing the computational domain.* Accurate representation of a derivative by its equivalent finite-difference expression requires a closely spaced mesh or "grid" of points throughout the domain of interest, especially in the zones where the derivatives are expected to be large or rapidly varying. The impeller in Figure 1 is shown meshed with a grid of 62,000 points or nodes per passage,⁵ a total that would result from about 25 points from hub to shroud and from blade to blade at each of 100 stations in the streamwise direction, including suitable extensions of the passage upstream and downstream to points where known boundary conditions can be imposed on the problem. CFD codes are available with grid generators that work with geometric design information in the form of 2D digitized drawings or three dimensional (3D) solid models to create the complete array of computational elements (bounded by the nodes or points) which are suitably shaped for optimum numerical stability during iterative computations of the solution.

Today it is not uncommon for a CFD mesh to contain an order of magnitude more nodes than were used to analyze one passage of the impeller of Figure 1. However, many more nodes than this is still a challenge in terms of computer time and power. Yet the earlier very successful turbulent boundary layer work dealt with a mesh of 500 or more points in the y -direction (normal to each wall), although with far fewer in the streamwise x -direction. Moreover, the spacing of these y -points was decreased as the wall was approached, which is essential in order to properly model the very thin sublayer near the wall and so to deduce the correct shear stress at the wall.^{13,14} Such a high density of grid points near wall boundaries together with the meshing requirements of a whole passage or the combined passages of the inlet, impeller, volute or diffuser, and crossover is still not feasible; so, CFD codes usually include known "wall functions" such as the turbulent "log law of the wall" related to Prandtl's mixing length theory and suitable representations of the laminar sublayer in the near-wall regions instead.

3. *Iterative computation of the solution.* Next, the appropriate boundary conditions for velocity components and pressure are imposed at the solid boundaries and the upstream and downstream throughflow boundaries. These are specified to be consistent with the number of unknowns and what is known at boundaries; for example, the velocity of the fluid at a solid boundary is generally zero relative to that boundary. Commercial CFD codes provide directions in this and many other aspects of their use. Initial values of all the variables are assigned to the nodes, and the algebraic system

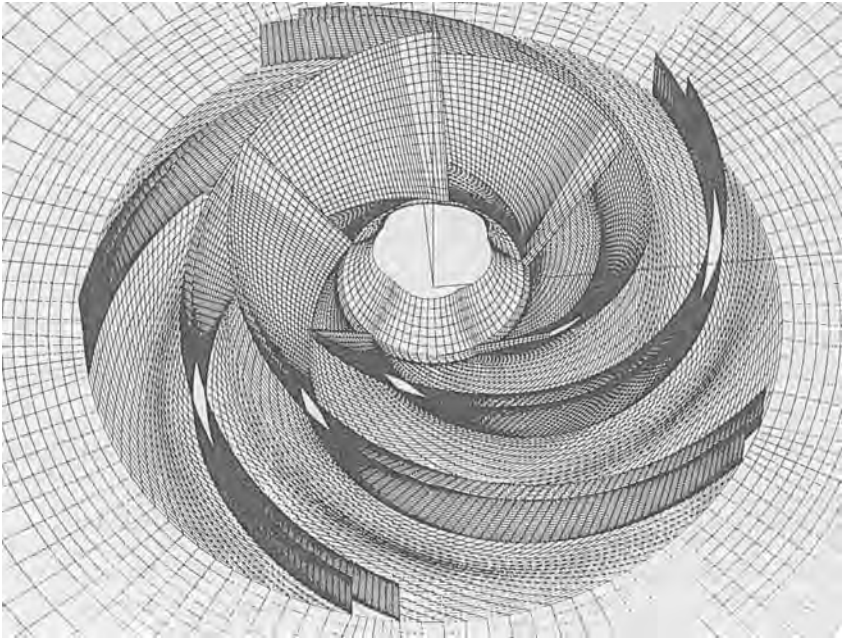


FIGURE 1 Centrifugal pump impeller of single-stage end-suction pump with a design specific speed N_s of 1500 ($\Omega_s = 0.55$), showing mesh for CFD analysis

Source: Reference 5 (International Pump Users Symposium, sponsored by Texas A&M University).

of equations is solved by iterating the values of the variables at each node until convergence on a set of values that satisfies the equations at all points is achieved within an acceptable margin of error. A variety of schemes for performing these iterations is employed such as (a) “marching” through the nodes in a prescribed order and assigning improved values at every point, such values themselves being obtained through algorithms that bring them closer to satisfying the equations; or (b) “relaxation” or making improvements first at the nodes where the errors are largest, or some combination of these and other approaches known to be suitable for the geometries and character of the expected solutions.

4. *Approximations employed in the computations.* Ideally, the CFD analysis of a pump should include all the terms of Eqs. 5 to 9 together with a level of turbulence modeling sophistication for the Reynolds stresses that is known to give results to the desired degree of accuracy. Required here is a sufficient number of nodes in both the stationary and rotating components of the entire machine from upstream of the inlet to downstream of the discharge port. This includes the unsteady terms, namely the derivatives with respect to time.

- a. *Steady flow in single components.* All of the above has indeed been done on multi-stage machines in research institutions. However, as mentioned in the introduction, early CFD work on pumps concentrated on steady-flow solutions (time-derivatives set to zero) for one component, especially the impeller. Moreover, such solutions were done for only one of the passages. This saved computer time and avoided the complexities of properly describing the flow across the interfaces between stationary and moving grids, and the results were surprisingly good.^{5,7} For example, the total impeller head for the impeller of Figure 1 is shown in Figure 2 to be reasonably accurate over a wide range of flow rate from 25 to 100% of that at BEP.

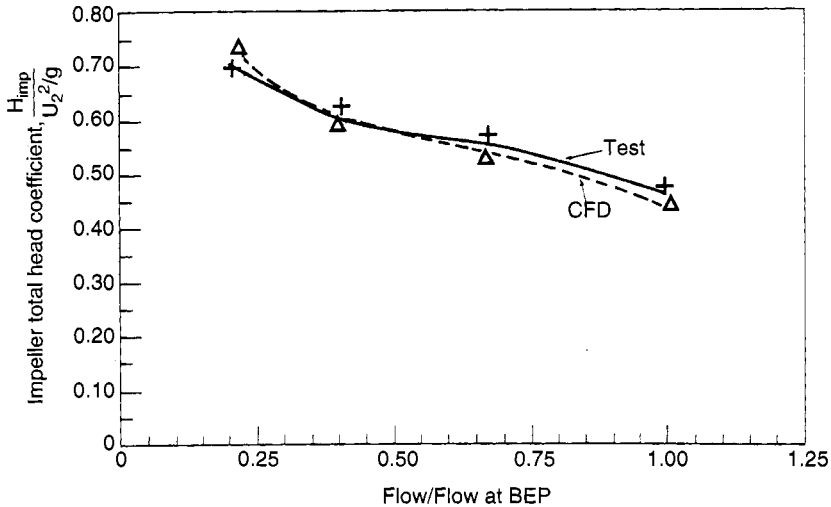


FIGURE 2 CFD prediction of total head compared to test results for the impeller of Figure 1
Source: Reference 5.

b. Steady flow in multiple components. The next step in complexity, while still avoiding unsteady computations, was to circumferentially average the flow variables at the interfaces between the rotating and fixed components. For example, this approach was utilized for the rotor-stator interface in the analysis of a mixed-flow diffuser pump⁹ and an axial-flow water-jet pump¹⁵ with mixed results on the accuracy of stage head and power consumption. However in another case, such averaging was avoided by freezing the impeller of a boiler feed pump stage relative to the volute and the inlet-plus-crossover passage. Moreover, all three components, including all of the impeller passages were included in the computational domain, which had a total of 276,352 nodes. This method, although for steady flow, preserved the recirculating flow patterns at the interfaces and resulted in the prediction of overall stage head at shutoff and at flow rates from 80 to 120% of BEP within 5% of the measured head on test.⁸

5. Enhancements for greater accuracy. Unsteady *RANS* solutions provide more accurate predictions of the flow field than the above steady-flow approximations. Yet, extensive experience with two-equation eddy viscosity models has led to advanced computational methods for further improvements of accuracy in the prediction of unsteady flow effects and losses and therefore of the overall pump performance curves. These are known as *Large Eddy Simulation (LES)*; *Hybrid RANS-LES*, which can include the so-called *Detached Eddy Simulation (DES)* method; and, ultimately, *Direct Numerical Simulation (DNS)*. Unsteady-flow computations are a feature of these enhancements, all of which require the computing power of parallel processing. Briefly these methods are described as follows.

a. LES. The basis of *LES* models is that fluctuations in the inertial regime, where turbulence production occurs and are characterized by a relatively large length scale, need to be resolved inherently by the numerical procedure. In contrast, turbulent eddies that have very small length scales and are primarily dissipative in nature may be modeled. Thus the grid size is chosen to be fine enough that the cell dimensions are smaller than the “large” inertial scales, while turbulent scales smaller than the cell size are modeled with so-called “sub-grid” turbulence models. For instance in wakes and other areas of turbulent mixing, especially at high Reynolds numbers, the large unsteady vortical structures as illustrated in the famous image by Brown and Roshko¹⁶ for a mixing layer are computed as part of the unsteady

solution procedure while the fine-grain turbulent dissipation is modeled. The “sub-grid” turbulent viscosity μ_t is low in these regions, sometimes as small as 10 to 20 times the molecular viscosity μ (also called the laminar viscosity μ_l) since it only represents the fraction of turbulent energy that is not being resolved by the calculation. In contrast, the turbulent viscosity in a *RANS* model represents the entire turbulent spectrum and may be very large (order of 3000 to 4000 times μ_l). One drawback of the *LES* framework is that the meshes become very large (small grid spacings), and the calculations can become very expensive particularly for very high Reynolds number flows where the dimension of the inertial length scale that needs to be resolved becomes smaller with increasing Reynolds number.

- b. *Hybrid RANS-LES or DES methods.* The sheer size and computational expense of using *LES* methods for large scale, complex 3D applications has necessitated the development of this more practical procedure. The philosophy here is to use a fine *LES* quality grid in specific regions of the flow where unsteady effects may be expected to dominate while using a coarser *RANS* grid in other more benign regions. For example, in a mixing layer an *LES* quality grid would be used in the vicinity of the shear layer encompassing the vortical structures while a coarse *RANS* grid would be used elsewhere. Many different procedures to smoothly blend the variation of μ_t as we cross the interface from a fine cell in the *LES* region to a coarse cell in the *RANS* region have been developed. One of the more popular blending methods is the *DES* method that has been extensively used in the modeling of separated flows.
- c. *DNS.* To avoid uncertainties about the scale of the turbulence as well as the exact locations of the corresponding regions of the flow field where the various scales exist, the Navier-Stokes equations can be solved directly for the unsteady velocity field. The grid for this *DNS* approach is extremely fine, and every turbulent length scale is involved. This is the ultimate way of solving a turbulent flow field; however, it is largely a theoretical concept and would command too much computer power to be practical for pump engineers. Moreover, for Reynolds numbers—based on the turbulent length scale—that are less than 1000, *DNS* is not practical.
6. *Parallel computing.* Ideally, increasing the number of processors—really individual computers networked together and sharing the computational tasks—reduces the costs, especially those of these enhanced computational methods. For example, it may take a day to generate the grid of an impeller passage and another day to run a CFD code and obtain a solution for it. In a fully unsteady computation, all the impeller passages are involved, and the number of processors should increase directly with the number of passages. Further, for multistage machines, the number of processors is found by multiplying their quantity for one stage by the number of rotor-stator combinations. More than 200 processors are required to perform unsteady flow calculations for a complete pump stage in the *Hybrid RANS-LES* mode. The computations are accomplished by transferring information for a component of the flow field, such as a rotor or rotor passage, to a waiting processor. Any disparity in the time required to compute the domain of a single component and that of another is termed “latency” because of the inaction of the waiting processors. Latency produces inefficiency in balancing the computational load across the various components. And latency, combined with inefficiencies in transferring information among the processors can lead to “saturation” or no improvement in computational speed, which in turn limits the maximum number of processors for the same grid and results in greater cost or “overhead.” From this discussion it can be seen that, while solving steady problems using the coarse meshes associated with *RANS* solvers requires basic computer resources, unsteady problems of large size using the above enhanced methods and solved with parallel processing require significant investment in computer resources and system administration.

OPTIMIZING THE HYDRAULIC DESIGN

Widespread acceptance of commercial CFD codes by the fluids engineering community in the product design process is the result of the extensive development and experience

with this valuable tool as outlined above. CFD codes and the computers on which they run are taking the place of the traditional upfront laboratory testing phase of the pump hydraulic development process. Here, the assumptions of steady flow and circumferential averaging across interfaces between rotating and stationary members are adequate to the purpose. The resulting rapid approximate CFD solutions enable engineers to more quickly sort out ideas and to identify and focus on areas of improvement of these pump hydraulic components.

Inlet Passageways Inlet passageways or “suction approach bays” that are integral to pump casings—for example, between-bearings pumps as illustrated in Section 2.2.1—receive the incoming fluid in a plane normal to the shaft and then redirect the fluid axially into the eye of the impeller, simultaneously distributing it circumferentially. An incoming radial conduit on many multistage pumps, this passageway is essentially an elbow discharging into the first-stage impeller eye. Such asymmetric inlet passageways can create major circumferential and radial distortions of the fluid entering the impeller, producing losses, cavitation, and pump vibration. Similarly, the crossover passageways on multistage pumps can deliver poor flow to the subsequent stages of multistage pumps.^{5,8} Because of its complex geometry, the suction approach bay can be one of the most challenging components for the hydraulic designer; and 3D flow analysis is essential in order to get any idea at all about the flow field approaching the impeller. This was clearly demonstrated with an early CFD-like finite-element code for the case of a low-loss double-suction pump bay. The viscous stresses were not computed; and this omission fortunately had little influence on that particular flow field, which agreed well with experimental measurements.¹⁷

Impellers Impellers were the first major focus of CFD-based optimization, because, as the energy transfer agent, an impeller (or inducer) has a major impact on overall pump performance. Q3D had been an effective analytical tool for judging the flow field.^{2,3} The objective was to design blading that could carry the load without separating the flow and suffering the consequent mixing losses, while minimizing skin friction losses. Q3D was basically an inviscid analysis method, to which empirically based loss expressions could be added. Among the primary outputs were the distributions of the local free-stream velocity W relative to the blade surfaces, which are related to the static pressure distributions p on the blades through the Bernoulli equation in terms of W and p at successive meridional positions through the impeller from inlet to exit, as illustrated in Figures 17 and 29 and approximated by Eq. 54 of Section 2.1.1. The diffusion or “D” factors (Eqs. 57 to 59 and computed on Figure 29 of Section 2.1.1) were not supposed to exceed experiential values that were guided by the limits established by NACA for maintaining lift on airfoils by avoiding separation. However, CFD solutions compute all the losses, thereby removing the uncertainties in applying the D-factor empiricism. Nevertheless, some impeller analysts using CFD still require outputs for the equivalent static pressure distributions on the blades—as an aid to interpretation of the fluid behavior and the optimization of the impeller hydraulic design.^{4,15} Moreover, in this CFD-assisted optimization process, the complexity and interrelation of the losses can reveal the need for subtle modifications in the blade shapes. In this regard, the spanwise distribution of the inlet angle and leading edge shape of the impeller blades must be optimized with respect to possible distortion of the flow entering from the inlet passageway. If needed, this is possible by coupling at least these two components in an unsteady analysis that preserves the circumferential flow distortions on the interface between the stationary inlet and rotating impeller grids. Such distortions, as stated earlier, can have a profound effect on the incidence losses of the impeller and unsteadiness arising from suction recirculation.⁵

Diffusers Substantially diffusing flows are well known to be distorted and unsteady. 1D analysis is impossible due to the unsteady blockages of boundary layers and wakes in various stages of incipient separation and stall even in the neighborhood of the BEP of the pump. Thus, in the design description of vaned diffusers in Section 2.1.1, all that can be offered by way of performance prediction are references to experimental diffuser data. As the flow rate is reduced below the design-point value, the stall patterns in vaned diffusers

intensify until rotating stall appears. CFD analysis of vaned diffusers, both isolated and coupled with an impeller, has reproduced these complex flow phenomena; indicated the effect of the proximity of the impeller blades to the diffuser vanes (called “Gap B” in Section 2.1.1) on the rotating stall; and provided diffuser performance predictions. These computations also illustrated the influence that recirculating flow from the diffuser into the impeller has on impeller flows at the low flow rates associated with rotating stall patterns and frequencies in both impeller and vaned diffuser.⁸ Similarly, vaneless diffusers can develop stall patterns that influence impeller flow stability and so are prime candidates for unsteady CFD analysis, particularly when coupled to the impeller. An unsteady-flow study of such a pump, using $k-\omega$ turbulence modeling, revealed that this geometry produced disturbances of the jet-wake impeller flow patterns. These in turn caused periodic fluctuations of velocity and pressure that gave insight into the relation of such fluctuations to experimental acoustic measurements of the flow.¹⁸

Volutes The influence of the volute sizing and design on volute pump performance and the circumferential pressure distribution around the impeller is discussed in Section 2.1.1. Experimental data for the resulting radial thrust magnitude and direction on the impeller is given for a typical pump in Section 2.1.3 as a function of the flow rate relative to BEP. Thus, it is expected that the volute affects the impeller flow field and performance and vice versa, and that a meaningful CFD solution of a volute must be done with it properly coupled to the impeller. This was done in the frozen-flow computation of the previously mentioned multistage boiler feed pump, in which the crossover-and-inlet passage were also included.⁸ While not truly unsteady, this solution produced the effect on the impeller and total pump performance of the circumferential pressure variation in the volute which developed at the lower flow rates that were analyzed. However, in the mid-flow range, there was some disparity between the predicted and experimental values of overall pump total head which might not occur for the predictions of a truly unsteady CFD solution.

PREDICTING PERFORMANCE

There are two main uses of CFD with regard to predicting overall pump performance, namely, (a) as part of the hydraulic design optimization process and (b) for presentation of pump characteristics of high accuracy for purchasers and users of pumps.

Predictions During the Design Process Steady-flow CFD is used to rapidly obtain reasonable approximations to the complete set of performance curves of a new pump hydraulic design. This is a step that is revisited frequently during an interactive design process. Such use is especially appreciated when designing variable geometry machines, as for example the axial flow pump of Reference 19, for which a unique set of performance curves exists for each setting of a row of inlet guide vanes. In this case the authors computed the desired range of performances by utilizing a commercial CFD code in the steady-flow mode and using circumferential averages of the velocities and pressure at the interface between the guide vanes and the impeller.¹⁹

Head and Efficiency versus Flow Rate As the vision for CFD usage moves beyond the optimization of hydraulic components, the degree to which such predictions approximate the actual test curves has become an increasingly visible metric for judging the effectiveness of this computational tool. Instances of steady-flow CFD solutions producing accurate results for total developed head and power—and, therefore, efficiency—of pumps in the neighborhood of the BEP have been cited in the foregoing discussions.^{8,9} As will be seen below, unsteady LES and Hybrid RANS-LES solutions consistently achieve near-perfect duplication of test results, not only at BEP but at other flow rates as well. This includes NPSHR versus flow rate, a topic which, for clarity, has so far been omitted from this CFD presentation; however, the additions to the codes that are required for CFD to deal with cavitation will now be described.

CAVITATION AND TWO-PHASE FLOW

Modeling Cavitation and Two-Phase Flow In the computation of the performance of the variable-geometry pump mentioned above, the authors also computed the curve of NPSHR versus flow rate for each guide-vane setting.¹⁹ Indeed, commercial CFD codes now possess this capability, which entails computation of the vapor cavity volumes that exist within the impeller when operating at conditions where the NPSH available is less than that of cavitation inception, $NPSH_i$ (see definition and descriptions in Sections 2.1.1 and 2.1.3.) As reviewed by Sloteman et al in Reference 20, the methods of accounting for cavitating flows in CFD codes began by establishing the domains of the vapor or gas cavities known to exist under such conditions; (see Figure 40 in Section 2.1.1 and Figure 24 in Section 12.5.) The cavities were treated as solid blockages whose extent and shape were “tracked” iteratively until their interfaces with the neighboring liquid had a uniform static pressure p that was equal to the vapor pressure p_v . The additional codes required to do this were eventually eliminated in favor of a constant-enthalpy vaporization model embedded in the RANS code, and that then evolved into a volume-of-fluid (VOF) model which combined the Rayleigh-Plesset bubble-growth equation with an accounting for the presence of noncondensable gases, such as air, in the fluid. Thermal equilibrium among the phases is assumed. This VOF model involves greater complexity in the equations but has worked well.

Cavitation Performance at the Design Point Commercial CFD codes embodying this cavitation model are now being applied in the pump industry to determine the NPSHR capability and two-phase behavior of centrifugal pumps and inducers. This had been readily accomplished at the design point, which usually does not involve backflow from the inlet.

1. *Centrifugal pump cavitation.* Applied to a boiler feed pump impeller with vapor cavities trailing off the leading edges of the blades, a commercial CFD code using a two-phase VOF model under the steady-flow approximation produced the cavity shown in Figure 3, which is seen to compare closely with the corresponding experimentally observed cavity from a laboratory flow visualization test.²⁰ As illustrated in Figure 4, the code was applied at each of four levels of NPSH as expressed in terms of the dimensionless

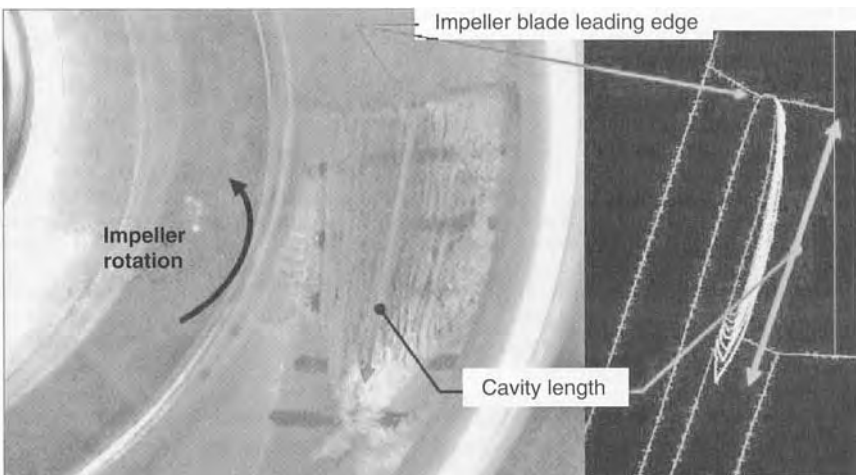


FIGURE 3 Comparison of suction surface cavity prediction with observations of the flow in a high-energy boiler-feed-pump impeller. The impeller blade on the left is operating near its design flow rate with a substantial sheet cavity. The plot at the right shows the vapor volume fraction computed with a two-phase CFD code.

Source: Reference 20 (International Pump Users Symposium, sponsored by Texas A&M University).

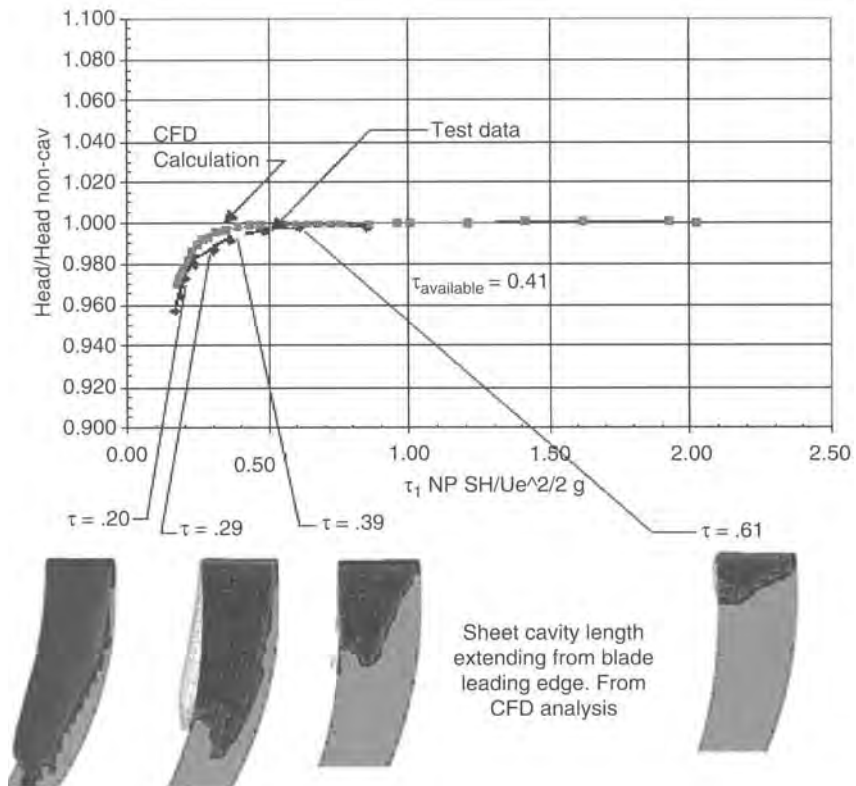


FIGURE 4 Suction performance of boiler feed pump at design flow rate compared to prediction via two-phase CFD. Illustrated at points along the curve of head versus cavitation coefficient τ (dimensionless $NPSH$) are the boundaries defined by a cavity region containing more than 10% vapor by volume. As is commonly observed, at $NPSH$ 3% (i.e., for head dropping off by 3% of its noncavitating value) the cavity extends back to the throat formed by the adjacent blade.

Source: Reference 20.

quantity $\tau = 2gNPSH/U_c^2$, τ ranged from 0.65 (somewhat below visual inception at τ_i) to 0.20 at the 3% break-off point, where the pump stage liquid head had deteriorated to 97% of its noncavitating value. Comparison of computed and observed bubble lengths at each τ -point are shown. More impressive is the virtual duplication of the Head-versus- $NPSH$ test curve of Figure 4 by the CFD solution. These results afford a quantitative validation of the two-phase CFD code that was used.²⁰

2. Inducer cavitation. The axial-flow helical inducer that is used for high suction specific speed (N_{ss} or Ω_{ss}) applications such as rocket propulsion pumps (as described in Section 12.18.2) achieves its large design value of N_{ss} with extensive cavitation activity present within the internal flow field. As such, the truly meaningful hydraulic analysis of an inducer includes both single- and two-phase flows. Figure 5 shows a typical inducer meshed for CFD analysis. The head-versus- N_{ss} curve at the design point for a three-bladed inducer similar to this one was generated in the steady-flow application of a commercial CFD code with advanced cavitation modeling. The computations produced the simulated increase in blockage of the passages by vapor and the attendant progressive unloading of the inducer blades downstream of their leading

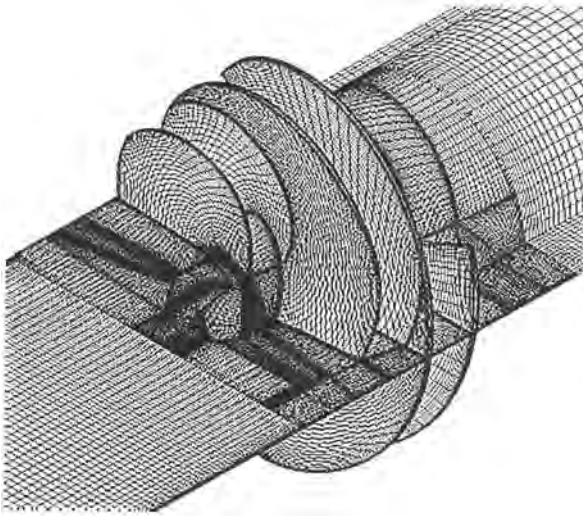


FIGURE 5 Inducer meshed for *LES*-type CFD analysis. This test inducer is similar to one used in a rocket engine liquid oxygen turbopump. It has three blades and an inlet blade angle at the tip of 7.5 degrees. Design flow inlet flow coefficient $\phi_d = 0.078$.

Source: Reference 25 (ASME).

edges as N_{ss} is increased (τ decreased), which is associated with head breakdown in these machines. Moreover, the CFD results for head versus N_{ss} agreed closely with experimental data for the whole range of NPSH to breakdown, which was a significant accomplishment in view of the large volumes of vapor involved. The cavitation modeling that was contained in this CFD code was thus validated for inducers, at least at the design point.²¹ Other inducer investigations have been carried out, one of which was unsteady and revealed rotating alternate-blade cavitation in a four-bladed inducer.²² An extensive experimental and analytical study of three inducers using the same CFD code as for the pump of Reference 20 (and in the steady-flow mode as well), produced three head-versus-NPSH curves for each inducer. Although computations were made at about 70% and 114% in addition to 100% of design flow coefficient, there was less agreement with experimental data than was realized in the pump case.²³

Off-Design Flows in Inducers Recirculation and backflow, as discussed in Sections 2.1.1 and 2.1.3, develop at flow rates reduced below the design point or BEP of a pump. For an inducer, which typically has an inlet eye diameter equal to its exit diameter, this off-design phenomenon sets in at flow rates not far below the BEP. As cavitation is usually present in inducers, they are known to develop extensive, highly complex, unsteady, cavitating flows under such conditions.²⁴ (Figure 41 in Section 2.1.1 and Figure 22 in Section 12.5 visualize such off-design two-phase, recirculating flows within impellers. Note the breakup of the cavities.) Therefore inducers are a logical focus of researchers trying to achieve agreement of two-phase CFD predictions with experimental data over the whole flow range from shutoff to runout.

1. *Single-phase CFD solution for backflow.* One approach in tackling this problem was the single-phase, unsteady CFD analysis of the inducer of Figure 5, which shows the mesh used for the *LES* solution that was conducted.²⁵ The solution was developed over a period corresponding to 40 to 55 revolutions of the inducer. Near-perfect agreement with experiment of the predicted developed-head performance was obtained as shown in Figure 6. The design flow coefficient ϕ_d was 0.078; so, ϕ ranged from 64 to 136% of ϕ_d .

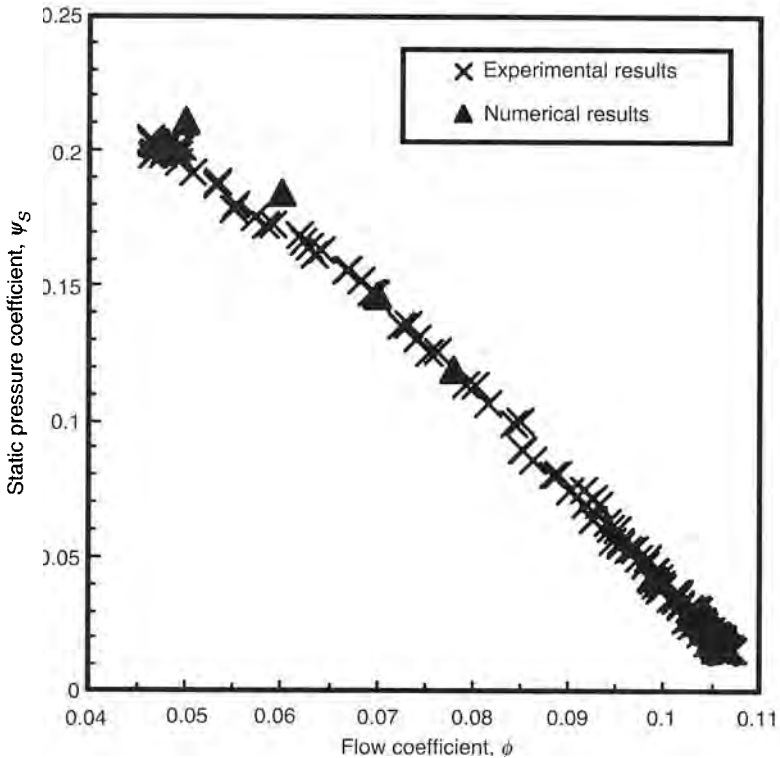


FIGURE 6 Comparison of CFD predictions of static pressure with experimental results for the test inducer of Figure 5 on water. Excellent agreement was achieved.
Source: Reference 25 (ASME).

The authors reported that backflow out of the inlet eye (left side of Figure 5) existed at 90% or $\phi = 0.070$, and they computed the pattern illustrated in Figure 7 at $\phi = 0.060$ ($= 77\%$ of ϕ_d). Corresponding to a high-NPSH condition, the intensity and extent of this backflow would be less at the more typical values of NPSH available to inducers, because the leading portions of the blades of an inducer tend to unload in the presence of the large volumes of vapor existing at low NPSH.²¹

2. *Single- and Two-phase CFD solutions with backflow.* As a further step in validating the unsteady *Hybrid RANS-LES* two-phase CFD code that was employed for the design-point study of Reference 21, a solution of the noncavitating backflow field was also computed for the low-pressure fuel turbopump or LPFTP on the space shuttle main engine (see Figure 3 and Table 1 of Section 12.18.2).²⁶ Computed at 70% of ϕ_d , the resulting upstream velocity field is presented in Figures 8 and 9. Figure 8 shows the concentration of the swirling backflow near the OD of the inlet duct, whereas the nonswirling incoming fluid is mainly at the axis of rotation. Figure 9 shows that in the region of intense interaction of these flows the turbulent viscosity μ_t is thousands of times greater than the laminar or molecular viscosity μ_l (or simply μ), whereas μ_t vanishes outside the interaction zone. Comparison of the velocity results of Figure 8 with experimental data was made at three upstream planes, as shown in Figures 10 and 11 for the swirl and axial components, respectively. The next step was to utilize the advanced cavitation modeling feature, and this was done at 90% of ϕ_d , at which condition rotating modes of cavitation were computed. This enabled quantification of the

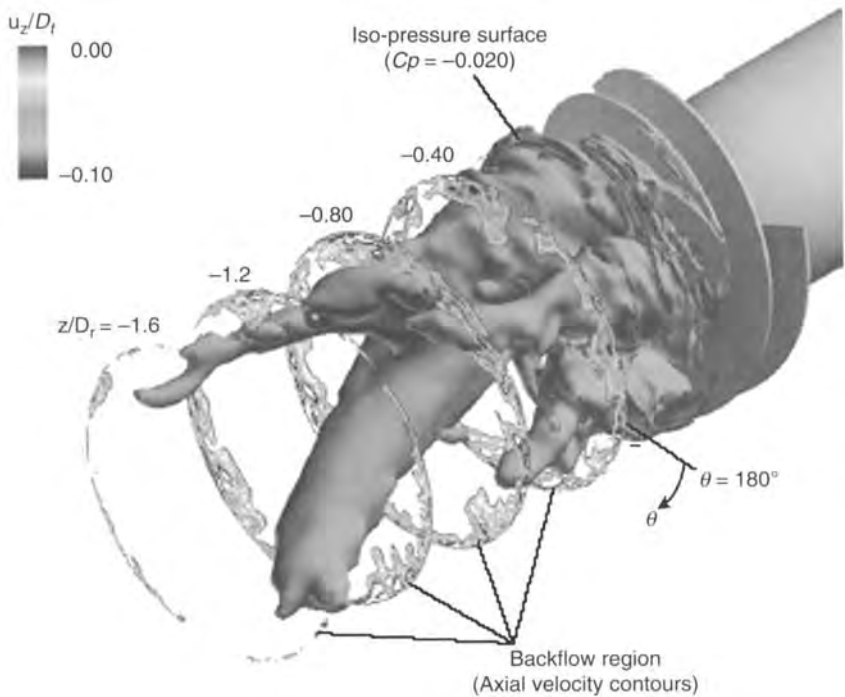


FIGURE 7 Backflow from the test inducer of Figure 5, computed at an inlet flow coefficient $\phi_i = 0.06$ after 55 revolutions. Illustrated is the backflow vortex at the shear layer between the incoming main flow and the swirling backflow.

Source: Reference 25 (ASME).

consequent radial loads that could have adverse mechanical consequences during operation at this condition. Both the computed frequency and amplitude of this instability agreed with experimental observations.

Thermodynamic Effects on Cavitation Performance Necessarily built in to the cavitation models of two-phase CFD codes is the effect of the fluid thermodynamic properties

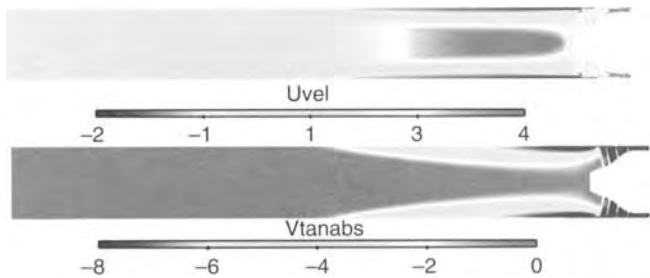


FIGURE 8 Backflow from the Space Shuttle Main Engine Low Pressure Fuel Pump (SSME LPPP) inducer, showing meridional (uvel) and swirl (vtanabs) velocity contours for 70% of design flow rate. The affected region extends well upstream of the pump, which is on the right.

Source: Reference 26 (AIAA).

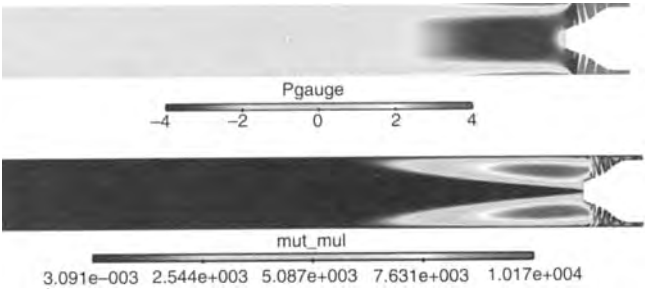


FIGURE 9 Backflow from the SSME LPPP inducer, showing static pressure p in psi (bar = psi/14.504) and turbulent viscosity μ_T contours for 70% of design flow rate, μ_T vanishes upstream of the region of interaction of the swirling backflow near the OD of the axial inlet pipe with the incoming (nonswirling) flow along the axis of rotation. μ_T is shown here as the ratio μ_T/μ_i , which is in the thousands in the highly turbulent interaction region.
Source: Reference 26 (AIAA).

involved in the vaporization process for computing the effects of cavitation. At points in the flow field where the static pressure drops below the vapor pressure, the liquid cools locally as the vapor forms, essentially by a constant-enthalpy or nearly constant-entropy process, as was alluded to in the above explanation of cavitation modeling.²⁰ Thus the heat of vaporization and the densities of the vapor and liquid all influence the volume of gas that appears in the flow for a given volume of liquid that is vaporized. Explained in Section 2.1.1 under *NPSH Effects*, this becomes important if the fluid is near its critical point, as are the cryogenic liquids encountered in liquid rocket propulsion systems, namely hydrogen and oxygen; (Section 12.18.2.) Inducers are generally deployed in such systems. Liquid hydrogen produces orders of magnitude less vapor volume for a given volume of liquid vaporized than does cold water. An early attempt to quantify the thermodynamic effect on the NPSH-capability of pumps was a Q3D-type inducer analysis code which contained a constant-entropy vaporization model as a point function of the pressure in the flow field.²⁷ That code successfully approximated this effect on breakdown NPSHR but not

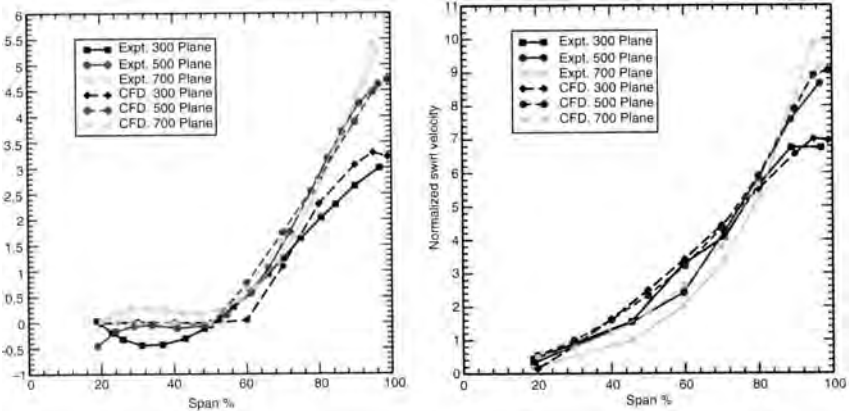


FIGURE 10 Comparisons of swirl velocity profiles with experimental data for the SSME LPPP inducer at three upstream planes, namely 0.29D (the 700 plane), 0.49D (the 500 plane), and 0.99D (the 300 plane). On the left (a) are the results for 90% of design flow rate; and, on the right (b), for 70% of design flow rate.
Source: Reference 26 (AIAA).

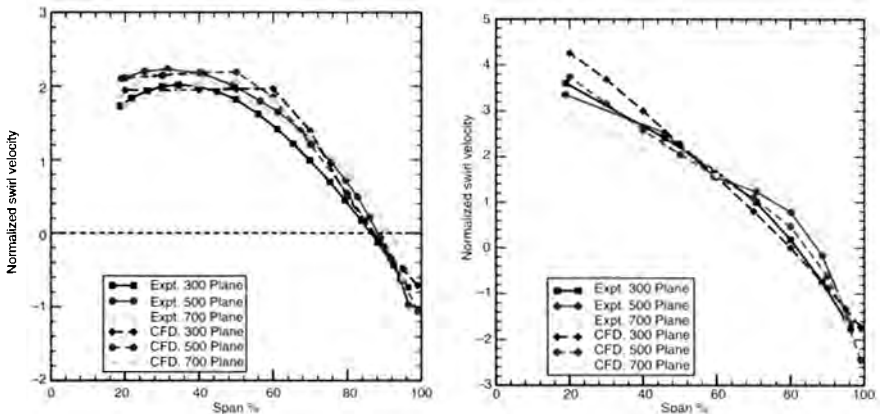


FIGURE 11 Comparisons of axial velocity profiles with experimental data for the SSME LPPP inducer at the same three upstream planes as in Figure 10. On the left (a) are the results for 90% of design flow rate; and, on the right (b), for 70% of design flow rate.

Source: Reference 26 (AIAA).

the shape of the fall-off curve of head versus NPSH. Initial efforts at validation of a current two-phase CFD code with respect to this effect consisted in successfully replicating the vapor cavity configurations and pressures observed in experiments with flows of cavitating liquid nitrogen and oxygen past bluff bodies.^{28,29} Simulation of the effect on inducer head versus N_{ss} was achieved for liquid oxygen in a comparison to that of water.²⁹ Extensive experiments in liquid hydrogen and water by NASA on the 6 degree (inlet tip blade angle) of Figure 12 were then simulated with the same *Hybrid RANS-LES* code mentioned above in both fluids.³⁰ In Figure 13, results for the head breakdown in liquid hydrogen computed at the design flow coefficient $\phi_d = 0.072$ are compared with test data taken at $\phi = 0.072, 0.074$, and 0.075 . Results displayed extreme sensitivity to even slight changes in liquid temperature and flow coefficient, and it is believed that uncertainties in the measurement of test temperature and flow coefficient, including variations during



FIGURE 12 NASA 6-degree flat-plate inducer, tested in water and in liquid hydrogen at various temperatures at the NASA Glenn facility between 1964 and 1970.

Source: Reference 30 (JPP).

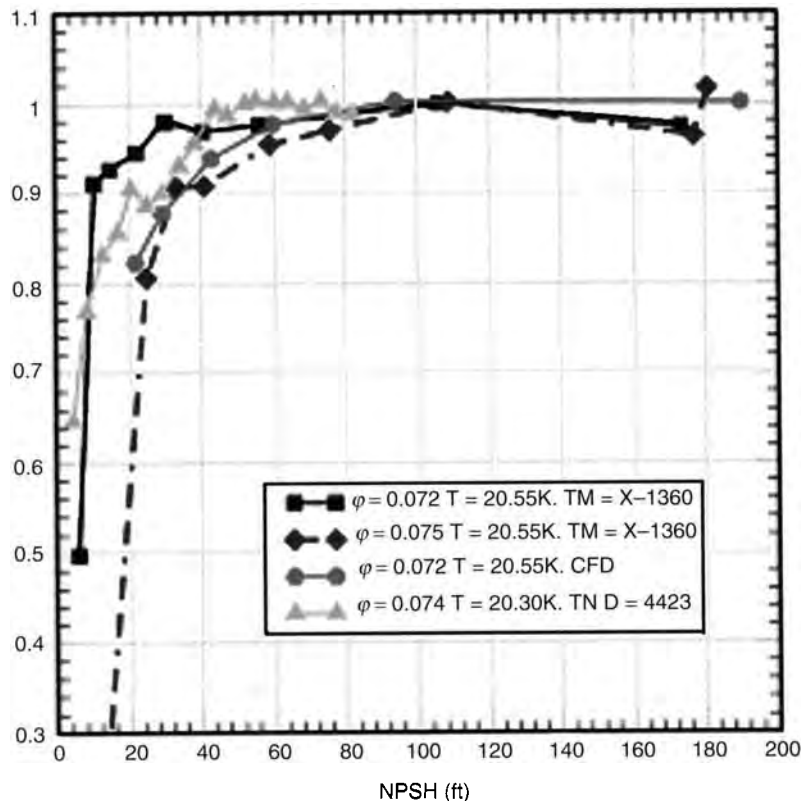


FIGURE 13 Suction performance of the NASA 6-degree inducer of Figure 12 in liquid hydrogen at 20,000 rpm and a temperature T of 20.55K (-423°F), comparing three sets of test results with computed CFD data. Flow rate at the design flow coefficient $\phi_d = 0.072$ was 1436 gpm (326 m^3/h). The test flow coefficient varied slightly, the greater value of ϕ producing the expected greater value of NPSH at head breakdown. ($\text{m} = \text{ft} \times 0.3048$). Source: Reference 30 (JPP).

each test run, are of the order of the difference between CFD and experiment at $\phi = 0.072$. And the CFD curve is conservative with respect to the test curve at $\phi = 0.072$. Moreover, the slightly lower temperature for the experimental data for $\phi = 0.074$ appears to bring its results for breakdown NPSH closer to the predicted values at $\phi = 0.072$. Finally, in Figure 14, the comparison of the CFD-predicted suction specific speed (N_{ss} or Ω_{ss}) or NPSHR capability of this inducer in liquid hydrogen to that of water is dramatic but expected from the NASA tests to which the CFD results were compared.³⁰

CONCLUSIONS

Computational Fluid Dynamics, or CFD, is the computerized numerical solution of the fundamental equations of motion of fluids throughout the domain of the passageways, both rotating and stationary, of a pump—either taken together or as separate components. Usually CFD involves solutions of the *Reynolds-averaged Navier-Stokes* equations (RANS equations) which contain the products of the fluctuating velocity components called Reynolds stresses. The success of CFD has depended on the relationships that represent

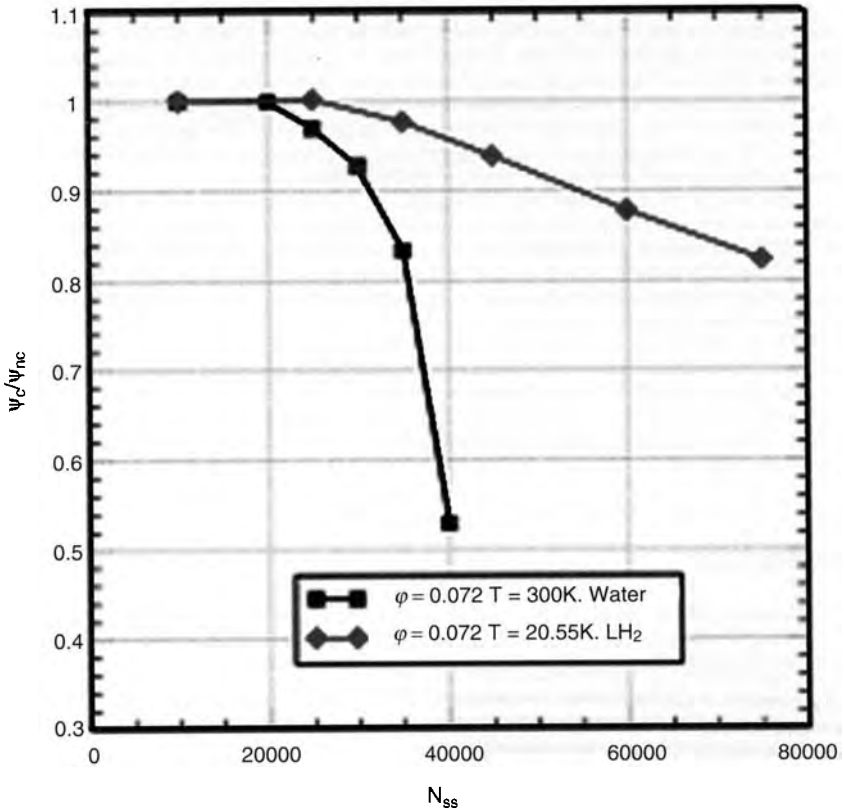


FIGURE 14 Comparison of curves of head-versus-suction specific speed N_{ss} of the NASA 6-degree inducer of Figure 12 pumping water at 300K (80°F) and liquid hydrogen at 20.55K (−423°F) as computed via CFD ($\Omega_{ss} = N_{ss}/2733$).

Source: Reference 30 (JPP).

these stresses and is called turbulence modeling. The most popular turbulence model currently in use is called the k - ϵ model, which expresses the eddy viscosity of the turbulence in terms of the turbulence kinetic energy k and the turbulent dissipation ϵ . Improvements and variations on this model to provide greater accuracy in simulation of actual flows exist and will continue to appear. To this model has been added the treatment of cavitation and two-phase flow that is needed to cover the gamut of pump flows. Methods of solution to increase accuracy in wakes and other areas of separated flows include large eddy simulation (LES) and Hybrid RANS-LES. Both steady flow and fully unsteady flow solutions can be obtained, the latter taking the most time, especially if LES is involved. Steady-flow CFD solutions take less time and can give useful information; however, precision and accuracy of time-varying velocities and pressures require fully unsteady computations. These are done with commercially available CFD codes running on multiple computers in a parallel processing mode.

The use of CFD in both the development and prediction of performance and the related operational features of the flow field velocities and pressures, both steady and fluctuating, has proven to be reliable and even essential for pump engineers, designers and users. Included are all flow rates from shutoff to runout, NPSH levels from inception to breakdown in liquids both close to and far from their thermodynamic critical points.

Head has been predicted to within experimental uncertainties at shutoff and from flow rates at least as low as 64% of BEP and at least as great as 136%. At all flow rates, predictions of both performance and flow patterns and instabilities, including domains of backflow, show the character of these phenomena with remarkable accuracy.

Inducers have been the vehicle for improving and validating both single- and two-phase CFD codes, as they have extensive zones of backflow and vapor due to cavitation. Such work has been successful and has created confidence that CFD is reliable for the more traditional and ubiquitous pumping configurations.

Applicability of CFD to any flow problem indicates that its use to define the operational behavior of systems in which pumps are installed should be as reliable as it has proven to be in turbomachinery and pumps and inducers in particular. Moreover, CFD solutions of the flow fields in system elements such as intake sumps, manifolds and the variety of suction piping configurations can function as the same aid to the system design process as has been described for pump internals.

CFD is creating a vision for the future. More computing power will lead to wider demand for and use of unsteady solutions, including *LES* and, perhaps, ultimately, direct numerical solution of the Navier-Stokes equations in terms of the instantaneous velocities and pressures. Already, development of pumps in some locations is proceeding on the basis of CFD alone—without benefit of development testing. Acceptance testing, however, will always be required.

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2.1.3 CENTRIFUGAL PUMPS: HYDRAULIC PERFORMANCE AND BEHAVIOR

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DEFINITIONS

Nomenclature Many of the quantities involved in this section are also dealt with in Section 2.1.1. Therefore, a single nomenclature that applies to both sections appears at the beginning of Section 2.1.1. Differences in notation exist for some of these quantities as a result of the coexistence of different traditions and pump cultures, so the nomenclature shows the equivalence in each case. An example is the use in this section of “c” and “w” to denote absolute and relative velocity respectively, whereas the NASA system of capital letters V and W is employed in Section 2.1.1.

Units The units used in this section are as defined in the nomenclature unless specifically noted in the text. In particular, the primary units for this section are those of the U.S. Customary System (USCS). A distinction in USCS usage in this section is that the pound force (lbf) is represented simply as “lb.” Conversions to SI units are given throughout this section, or the actual equivalent SI values are given in parentheses.

However, the number appearing in parentheses after the USCS value of specific speed n_s is the equivalent value of the universal specific speed Ω_s . Note that the value of specific speed corresponding to the best efficiency point (BEP) operating conditions of the pump is the value of interest and is often used to identify the impeller geometry involved.

Volume Flow Rate Abbreviated to “flow rate” and known traditionally as “pump capacity” Q , this is the volume of liquid per unit time delivered by the pump. In USCS units, Q is expressed in U.S. gallons per minute or USgpm, for which the abbreviation “gpm” is used. (1 US gallon = 231 in³)

Datum for Pump Head As defined in Eq. 3 and Figure 1 of Section 2.1.1, the total head has components of pressure, velocity, and elevation Z (or Z_e). Because pump head H (more

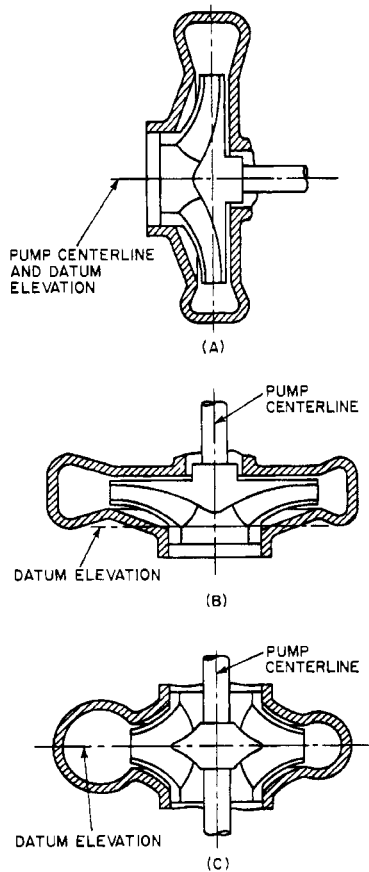


FIGURE 1A through C Elevation datum for defining pump head (Hydraulic Institute ANSI/HI 2000 Edition Pump Standards, Reference 27)

precisely ΔH) is the difference of the total heads evaluated at the discharge flange d and the suction flange s respectively, the elevation of the datum from which Z is measured cancels out. However, for purposes of identification, computing NPSH, and so on, the standard datum as shown in Figure 1 is used.

The standard datum for horizontal-shaft pumps is a horizontal plane through the centerline of the shaft (Figure 1a). For vertical-shaft pumps, the datum is a horizontal plane through the entrance eye of the first-stage impeller (Figure 1b) if single suction or through the centerline of the first stage impeller (Figure 1c) if double suction. Because pump head is the difference between the discharge and suction heads, it is not necessary that the standard datum be used, and any convenient datum may be selected for computing the pump head.

Power In USCS, the pump output is customarily given as liquid horsepower (lhp) or as water horsepower if water is the liquid pumped. It is given by

$$lhp = \frac{QH(\text{sp. gr.})}{3960} \quad (1)$$

where Q is in gallons per minute, H is in feet, and sp. gr. is specific gravity. If Q is in cubic feet per second, the equation becomes

$$lhp = \frac{QH(\text{sp. gr.})}{8.82} \quad (2)$$

In SI, the power P in watts (W) is given by

$$P = 9797QH (\text{sp. gr.}) \quad (3)$$

where Q is in cubic meters per second and H is in meters.

When Q is in liters per second and H is in meters

$$P = 9.797QH (\text{sp. gr.}) \quad (4)$$

Efficiency The pump efficiency η is the liquid horsepower divided by the power input to the pump shaft. The latter usually is called the brake horsepower (bhp). The efficiency may be expressed as a decimal or multiplied by 100 and expressed as percent. In this section, the efficiency will always be the decimal value unless otherwise noted. Some pump driver-units are so constructed that the actual power input to the pump is difficult or impossible to obtain. Typical of these is the "canned" pump for volatile or dangerous liquids. In such case, only an *overall efficiency* can be obtained. If the driver is an electric motor, this is called the *wire-to-liquid efficiency* or, when water is the liquid pumped, the *wire-to-water efficiency*.

CHARACTERISTIC CURVES

Pump with Nonviscous Flow and Zero Slip The basic shapes of centrifugal pump performance characteristics arising from various geometries can be ascertained and compared without the necessity of evaluating the slip. (Illustrated in Figure 15 of Section 2.1.1, the slip phenomenon is explained in the related discussion¹.) For this purpose, one employs the artifice of nonviscous flow through an impeller with an infinite number of blades having infinitesimal thickness and that therefore produce neither structural (geometric) nor flow (boundary layer) blockage. The result is the ideal head for no slip or blockage H_e as given by Eq. 5 (cf. Eq. 15b of Section 2.1.1):

$$H_e = \frac{u_2 c_{u2}}{g} - \frac{u_1 c_{u1}}{g} \quad (5)$$

The velocity components in Eq. 5 can be seen in the velocity diagrams of Figure 2, slip being neglected. Generally the inlet swirl term is small, and Eq. 5 can be approximated by

$$H_e \cong \frac{u_2 c_{u2}}{g} \quad (6)$$

As shown in Figure 2b, the absolute velocity vector c may be resolved into the meridional, or radial, velocity c_m and the peripheral velocity c_u . From the geometry of the figure

$$c_{u2} = u_2 - \frac{c_{m2}}{\tan \beta_2} \quad (7)$$

which, substituted into Eq. 6, gives

$$H_e = \frac{u_2^2}{g} - \frac{u_2 c_{m2}}{g \tan \beta_2} \quad (8)$$

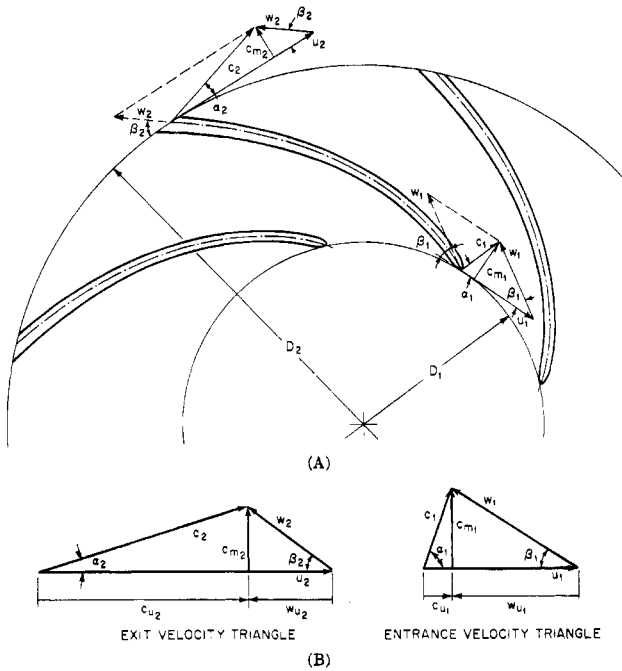


FIGURE 2A and B Velocity diagrams for radial-flow impellers, neglecting slip and blockage

Neglecting leakage flow, the meridional velocity c_m must be proportional to the flow rate Q . With the additional assumption of constant impeller speed, Eq. 8 becomes

$$H_e = k_1 - k_2 Q \quad (9)$$

in which k_1 and k_2 are constants, with the value of k_2 dependent on the value of the vane angle β_2 . Figure 3 shows the H_e -vs- Q characteristics for the three possible conditions on the vane angle at exit β_2 . The second right-hand term in Eq. 5 may be treated in like manner to the foregoing and included in Eqs. 8 and 9. The effect on Figure 3 would be to change the value of $H_e = u_2^2/g$ at $Q = 0$ and the slopes of the lines, but all head-flow rate characteristics would remain straight lines.

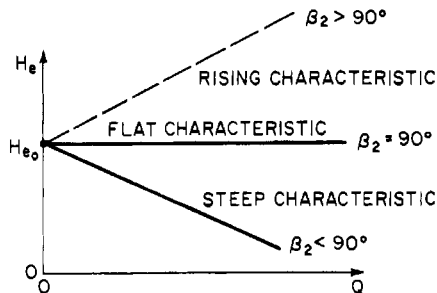


FIGURE 3 Head-versus-flow rate characteristics for nonviscous, zero-slip impeller flow

Viscous Flow with Slip The real flow situation involves friction losses in an impeller with a finite number of relatively widely spaced blades. Thus, slip occurs, reducing the exit flow angle $\beta_{f,2}$ below that of the blade β_2 (or, more precisely, $\beta_{b,2}$), which in turn reduces $c_{u,2}$ (cf. Figure 15 of Section 2.1.1). Therefore, the ideal head H_i drops below H_e . Moreover, losses and recirculation occur to cause additional deviation of pump head H from H_e . While CFD flow analysis can be employed to predict H with fair accuracy², lesser means, such as one-dimensional analysis, require experienced correlation and calibration skills to make such predictions. Therefore, many engineers commonly depend on testing and empirical modification of test results on the exact or similar geometry to make the final determination of the performance characteristics of a pump. This effort involves constant-speed plots of data as shown in the example of Figure 4.

Pumps are designed to operate at the point of best efficiency. The head, power, and flow rate at best efficiency, often called the *normal* values, are indicated in this section by H_n , P_n , and Q_n , respectively. Sometimes a pump may be operated continuously at a flow rate slightly above or below Q_n . In such case, the actual operating point is called the *rated* or *guarantee* point if the manufacturer specified this flow rate in the guarantee. It is unusual to operate a pump continuously at a flow rate at which the efficiency is much below the maximum value. Apart from the unfavorable economics, the pump may be severely damaged by continued off-design operation, as described later.

Backward-Curved Blades, $\beta_2 < 90^\circ$ Figure 4 shows the characteristics of a double-suction pump with backward-curved blades, $\beta_2 = 23^\circ$. The impeller discharged into a single volute casing, and the specific speed was $n_s \approx 2200$ ($\Omega_s \approx 0.8$) at best efficiency. At shutoff ($Q = 0$), Eq. 8 predicts H_i to be 372 ft (113 m), whereas the pump actually developed about 210 ft (64 m), and this head remained nearly constant for the range $0 < Q < 1000$ gpm (63 l/s). For $Q > 1000$ gpm (63 l/s), the head decreased with increasing flow rate but not in the linear fashion predicted by Eq. 9. At best efficiency, where $Q_n = 3200$ gpm (202 l/s), Eq. 8 predicts $H_e = 281$ ft (86 m), whereas the pump actually developed $H_n = 164$ ft (50 m).

Radial Blades, $\beta_2 = 90^\circ$ Large numbers of radial-blade pumps are used in many applications, from cellar drainers, cooling-water pumps for internal combustion engines, and other applications where low first cost is more important than high efficiency to highly engineered pumps designed for very high heads. The impellers are rarely more than 6 in

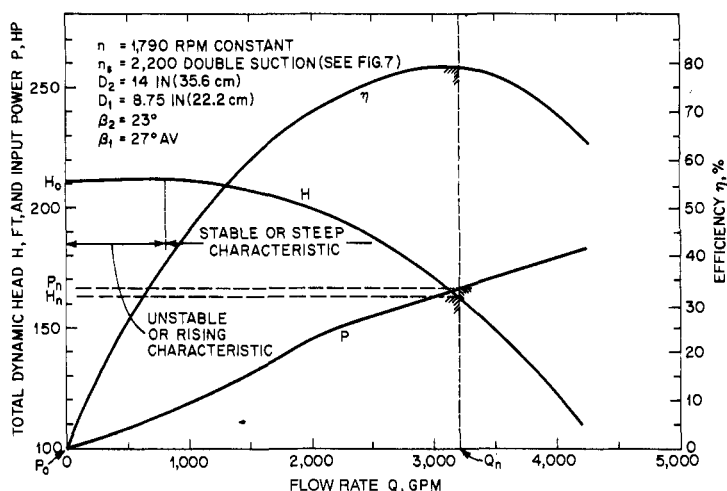


FIGURE 4 Typical pump characteristics, backward-curved blades (ft $\times 0.3048$ = m; hp $\times 745.7$ = W; gpm $\times 0.06309$ = l/s)

(15 cm) in diameter, but the speed range may be from a few hundred to 30,000 rpm or more. The casings usually are concentric with the impellers and have one or more discharge nozzles that act as diffusers. The impellers usually are open, with three or more flat blades. The clearance between blades and casing is relatively large for easy assembly. Such pumps exhibit a flat head-capacity curve from shutoff to approximately 75% of best efficiency flow rate, and beyond this flow the head-flow curve is steep. The pumps develop a higher head, up to 8000 ft (2400 m) per stage, than pumps with backward-curved blades, but the efficiency of the former usually is lower. (See also Section 2.2.1.)

Figure 5 shows the characteristics of a pump as reported by Rupp.³ The impeller was fully shrouded, $D = 5.25$ in (13.3 cm), and fitted with 30 blades of varying length. The best efficiency, $\eta = 55\%$, was unusually high for the specific speed $n_s = 475$ (0.174), as may be seen from Figure 6. The head-flow curve showed a rising (unstable) characteristic for $0 < Q < 25$ gpm (1.6 l/s) and a steep characteristic for $Q > 25$ gpm (1.6 l/s).

Figure 7 shows the characteristics of a pump as reported by Barske.⁴ The impeller was open, $D \approx 3$ in (7.6 cm), and fitted with six radial tapered blades. The effective β_2 may have been slightly greater than 90° due to the taper. At 30,000 rpm, the best efficiency was more than 35% at $n_s = 355$ (0.130), which is much higher than for a conventional pump of this specific speed and flow rate (Figure 6). The head-flow curve showed a nearly flat characteristic over most of the usable range, as predicted by Eq. 8, but the head was always lower than H_e . The smooth concentric casing was fitted with a single diffusing discharge nozzle. When two or more nozzles were used, the head-flow curves showed irregularities at low flow rates and became steep at high flow rates.

Manson⁵ has reported performance characteristics for jet engine fuel pumps having straight radial blades in enclosed impellers. The head curves showed unstable characteristics at low flow rates and steep characteristics at higher flow rates. The best efficiency reported was 54.7% for an impeller diameter of 3.300 in (8.382 cm) and speed $n = 28,650$ rpm.

Forward-Curved Blades, $\beta_2 > 90^\circ$ Pumps with forward-curved blades have been proposed,⁶ but the research necessary to achieve an efficient design appears never to have been carried out. However, radial-flow fans with forward-curved blades are common. Tests have been made of conventional, backward-curved-blade, double-suction pumps with the impellers mounted in the reversed position but with rotation correct for the volute casing. As tested, these pumps therefore had forward-curved blades.

Table 1 shows the pertinent results for six different pumps. Both flow rate and efficiency were drastically reduced, and there was only a modest increase in head for five of the six pumps. The sixth pump showed a 38% increase in head over that obtained with the impeller correctly mounted. Published estimates^{7,8} of the head-flow curves to be expected

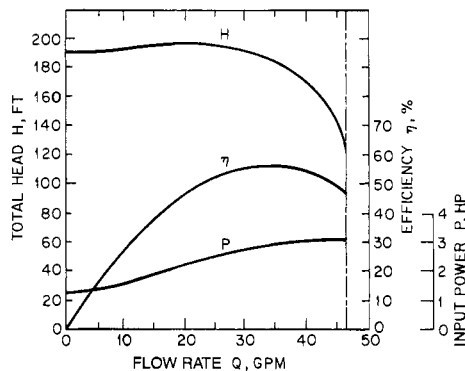


FIGURE 5 Pump characteristics, radial blades (ft $\times 0.3048 =$ m; hp $\times 745.7 =$ W; gpm $\times 0.06309 =$ l/s) (Reference 3)

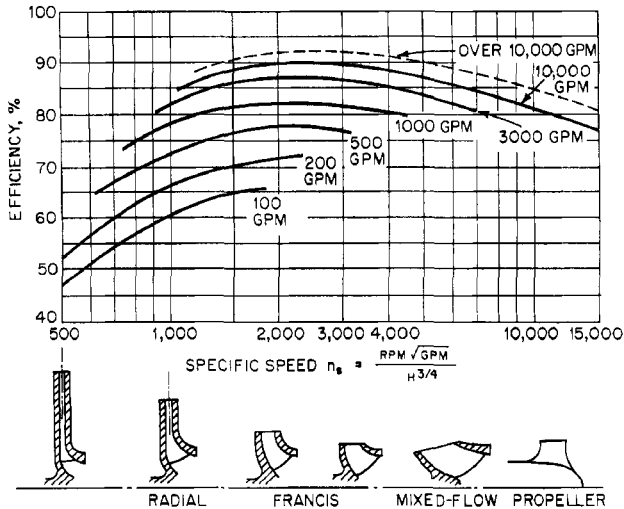


FIGURE 6 Pump efficiency versus specific speed and size. By way of comparison, Figure 10 of Section 2.1.1 (generated by Eq. 44 of that section) shows higher efficiency at low n_s and low flow rate at a constant value of surface roughness ϵ . However, if (in Eq. 44 of Section 2.1.1) substantially greater surface roughness is used for the small sizes, the results are similar to what is shown here. ($\text{gpm} \times 0.06309 = \text{l/s}$, and $\Omega_s = n_s/2733$). (Flowserve Corporation)

from reversed impellers predict an unstable characteristic at the low end of the flow rate range and a steep characteristic at the high end of the range.

PERFORMANCE EFFECTS

Classification of Curve Shapes A useful method for comparing characteristics of pumps of different specific speeds is to normalize on a selected operating condition, usually best efficiency. Thus,

$$q = \frac{Q}{Q_n} \quad h = \frac{H}{H_n} \quad p = \frac{P}{P_n} \quad (10)$$

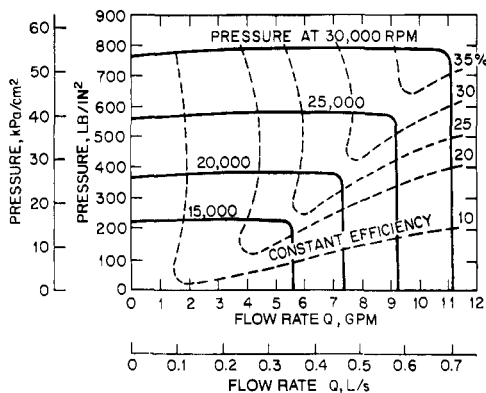


FIGURE 7 Pump characteristics, radial blades ($\text{lb/in}^2 \times 6.894 = \text{kPa}$; $\text{gpm} \times 0.06309 = \text{l/s}$) (Reference 4)

TABLE 1 Effects of reversed mounting of impeller

Number of stages	Specific speed per stage n_s (Ω_s)	Percent of normal shutoff head	Percent of normal values at best efficiency			
			Head	Flow Rate	Power	Efficiency
2	828 (0.303)	86	111	65	104	71
2	1024 (0.375)	82	112	88	145	68
1	1240 (0.454)	75	105	38.5	68.5	59
1	1430 (0.523)	82	106	69.7	138	53.5
1	2570 (0.940)	74.5	117	62	138	52.5
1	2740 (1.003)	77.5	138	61.5	180	47

Source: Flowsolve Corporation

where the subscript n designates values for the best efficiency point. Figures 8, 9, and 10 show approximate performance curves normalized on the conditions of best efficiency and for a wide range of specific speeds as defined in Table 2. These curves are applicable to pumps of any size because absolute magnitudes have been eliminated. In Figure 8, curves 1 and 2 exhibit a *rising head* or *unstable* characteristic where the head increases with increasing flow rate over the lower part of the flow rate range. This may cause instability at heads greater than the shutoff value, particularly if two or more pumps are operated in parallel. Curve 3 exhibits an almost constant head at low flow rates and is often called a *flat* characteristic. Curves 4 to 7 are typical of a *steep* or *stable head* characteristic, in which the head always decreases with increasing flow rate. Although the shape of the head-flow curve is primarily a function of the specific speed, the designer has some control through selection of the vane angle β_2 number of impeller vanes n_b , and capacity coefficient $\phi = c_{m2}/u_2$, as described in Section 2.1.1 (see also Figure 2). For pumps having a single-suction specific speed of approximately 5000 (1.83) and higher, the power is at its maximum at shutoff and decreases with increasing flow rate. This may require an increase in the power rating of the driving motor over that required for operation at normal flow rate.

Efficiency The efficiency η is the product of three component efficiencies (defined in Section 2.1.1):

$$\eta = \eta_m \eta_v \eta_h \tag{11}$$

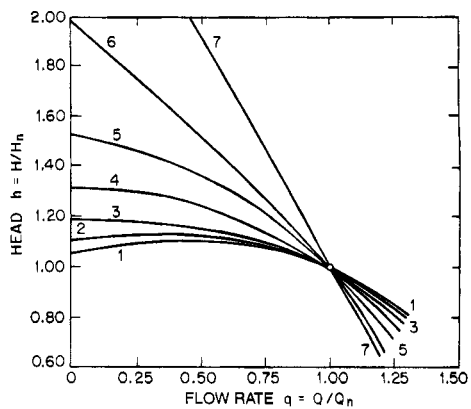


FIGURE 8 Head curves for several specific speeds, as defined in Table 2 (Reference 12)

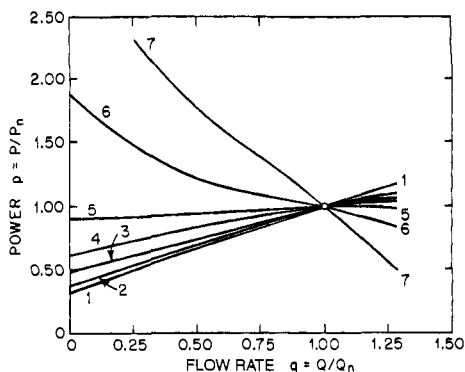


FIGURE 9 Power curves for several specific speeds, as defined in Table 2 (Reference 12)

The mechanical efficiency η_m accounts for the bearing, stuffing box, and all disk-friction losses including those in the wearing rings and balancing disks or drums if present. The volumetric efficiency η_v accounts for leakage through the wearing rings, internal labyrinths, balancing devices, and glands. The hydraulic efficiency η_h accounts for liquid friction losses in all through-flow passages, including the suction elbow or nozzle, impeller, diffusion vanes, volute casing, and the crossover passages of multistage pumps. Figure 11 shows an estimate of the losses from various sources in double-suction single-stage pumps having at least 12-in (30-cm) discharge pipe diameter. Minimum losses and hence maximum efficiencies are seen to be in the vicinity of $n_s \approx 2500$ (0.91), which agrees with Figure 6.

Effects of Pump Speed Increasing the impeller speed increases the efficiency of centrifugal pumps. Figure 7 shows a gain of about 15% for an increase in speed from 15,000 to 30,000 rpm. The increases are less dramatic at lower speeds. For example, Ippen⁹ reported about 1% increase in the efficiency of a small pump, $D = 8$ in (20.3 cm) and $\eta_s = 1992$ (0.73), at best efficiency, for an increase in speed from 1240 to 1880 rpm. Within limits, the cost of the pump and driver usually decreases with increasing speed. Abrasion and wear increase with increasing speed, particularly if the liquid contains solid particles in suspension. The danger of cavitation damage usually increases with increasing speed unless certain suction requirements can be met, as described later.

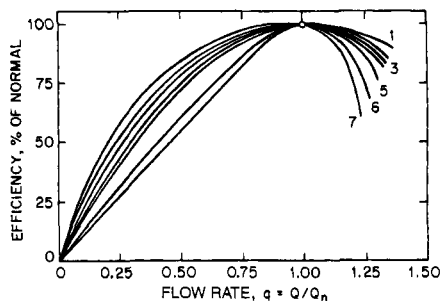


FIGURE 10 Efficiency curves for several specific speeds, as defined in Table 2 (Reference 12)

TABLE 2 Characteristic curves as a function of specific speed (Figures 8, 9, and 10)

Curve Number on Figures 8, 9, and 10	USCS Specific Speed n_s	Metric Specific Speed n_q (Eq's 38, Sect. 2.1.1)	Universal Specific Speed Ω_s	Impeller Suction Configuration
1	900	17	0.33	Double
2	1500	29	0.55	Double
3	2200	43	0.80	Double
4	3000	58	1.10	Double
5	4000	77	1.46	Double
6	5700	110	2.09	Single
7	9200	178	3.37	Single

Effects of Specific Speed Figures 6 and 11 show that maximum efficiency is obtained in the range 2000 (0.73) < n_s < 3000 (1.10), but this is not the only criterion. Pumps for high heads and small flow rates occupy the range 500 (0.18) < n_s < 1000 (0.37). At the other extreme, pumps for very low heads and large flow rates may have $n_s = 15,000$ (5.49) or higher. For given head and flow rate, the pump having the highest specific speed that will meet the requirements probably will be the smallest and least expensive. However, Figure 9 of Section 2.1 shows that it will run at the highest speed and be subject to maximum wear and cavitation damage, as previously mentioned.

Effects of Clearance

WEARING-RING CLEARANCE Details of wearing-ring construction are given in Section 2.2.1. Schematic outlines of two designs of rings are shown in Figure 12. The L-shaped construction shown in Figure 12a is very widely used with the close clearance between the cylindrical portions of the rings. Leakage losses increase and pump performance falls off as the rings wear. Table 3 shows some of the effects of increasing the clearance of rings similar to Figure 12a.

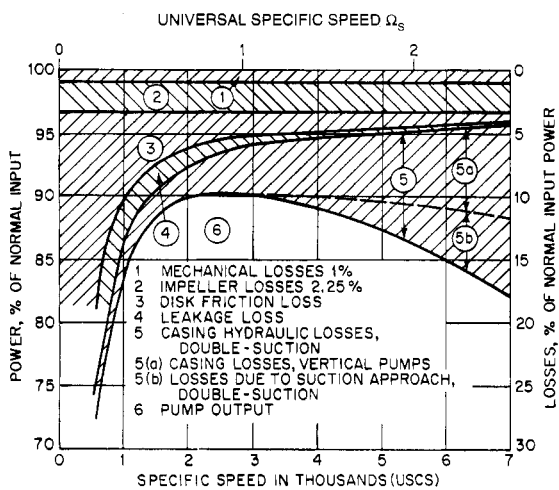


FIGURE 11 Power balance for double-suction pumps at best efficiency (Reference 12)

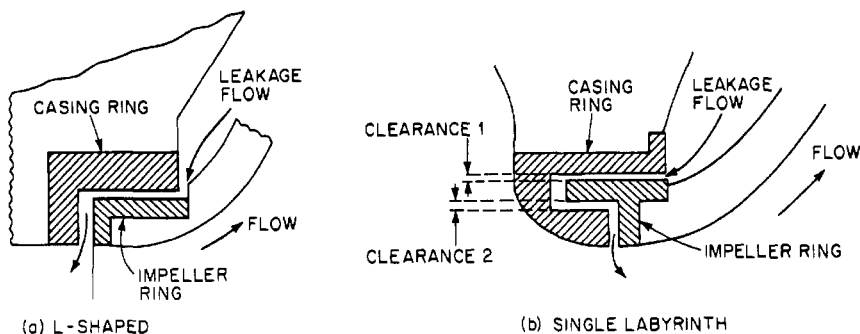


FIGURE 12A and B Typical wearing rings

The labyrinth construction shown in Figure 12b has been used to increase the leakage path without increasing the axial length of the rings. If the pressure differential across these rings is high enough, the pump shaft may take on lateral vibrations with relatively large amplitude and long period, which can cause serious damage. One remedy for these vibrations is to increase clearance 2 in Figure 12b relative to clearance 1, at the expense of an increased leakage flow. High-pressure breakdown through plain rings may cause vibration,¹⁰ but this is not usually a serious problem. It is considered good practice to replace or repair wearing rings when the nominal clearance has doubled. The presence of abrasive solids in the liquid pumped may be expected to increase wearing-ring clearances rapidly.

BLADE-TIP CLEARANCE Many impellers are made without an outer shroud and rely on close running clearances between the blade tips and the casing to hold leakage across the blade tips to a minimum. Although this construction usually is not used with pumps having specific speeds less than about 6000 (2.20), Wood et al.¹¹ have reported good results with semi-open impellers at $1800 (0.66) \leq n_s \leq 4100 (1.50)$. It appears that both head and efficiency increase with decreasing tip clearance and are quite sensitive to rather small changes in clearance. Reducing the tip clearance from about 0.060 in (1.5 mm) to about 0.010 in (0.25 mm) may increase the efficiency by as much as 10%. Abrasive solids in the liquid pumped probably will increase tip clearances rapidly.

TABLE 3 Effects of increased wearing-ring clearance on centrifugal pump performance

Specific speed n_s (Ω_s)	Design head, ft (m)	Ring clearance, % of normal value	Percent of values at shutoff with normal ring clearance			Percent of values at best efficiency with normal ring clearance		
			Q	H	P	η	H_0	P_0
2100 (0.77)	63 (19.2)	178	100	98.3	98.9	99.4	97.0	100
...	...	356	100	97.5	99.0	98.5	93.6	98.2
...	...	688	100	96.0	98.9	97.1	91.2	94.8
...	...	1375	100	94.3	97.4	96.8	88.8	92.5
3500 (1.28)	65 (19.8)	354	100	90.0	99.1	90.8	85.0	96.2
4300 (1.57)	41 (12.5)	7270	62	65.5	81.7	49.8	44.3	106
4800 (1.76)	26 (7.9)	5220	96	78.8	89.2	84.8	78.2	83.3

Source: Flowserve Corporation

MODIFICATIONS TO IMPELLER AND CASING

Diameter Reduction To reduce cost, pump casings usually are designed to accommodate several different impellers. Also, a variety of operating requirements can be met by changing the outside diameter of a given radial impeller. Eq. 6 shows that the head should be proportional to $(nD)^2$ provided that the exit velocity triangles (Figure 2b) remain similar before and after cutting, with w_2 always parallel to itself as u_2 is reduced. This can be achieved if the impeller meridional exit area $A_{m,2}$ is the same before and after cutting—and if the flow angle $\beta_{f,2}$ (Figure 15, Section 2.1) also stays the same. [$\beta_{f,2}$ would stay the same if the blade angle $\beta_2 (= \beta_{b,2})$ does, the difference being due to slip velocity V_s that should also scale down with D .] For radial discharge impellers, area $A_{m,2}$ equals $\pi D b_2$ (minus blade and boundary layer blockage) and requires that b_2 increase as D decreases. This is typical of many impellers—as is constancy of β_2 over the cutting range—and, together with Eqs. 27–33 of Section 2.1.1, leads to the following rules for predicting performance:

$$\frac{Q_1}{Q_2} = \frac{n_1 D_1}{n_2 D_2} \quad (12a)$$

$$\frac{H_1}{H_2} = \frac{n_1^2 D_1^2}{n_2^2 D_2^2} \quad (12b)$$

$$\frac{P_1}{P_2} = \frac{n_1^3 D_1^3}{n_2^3 D_2^3} \quad (12c)$$

which apply only to a given impeller with altered D and constant efficiency but *not* to a geometrically similar series of impellers. (Eqs. 12 have been termed “affinity laws”; however, true affinity requires geometric similarity, which in turn yields the *Scaling and Similitude* relationships of Section 2.1.1.) The assumptions on which Eqs. 12 were based are rarely if ever fulfilled in practice, so exact predictions by the equations should not be expected. A common example is the low- n_s radial discharge impeller with parallel radial hub and shroud profiles over most of the path from inlet to exit (Figure 6). Here, $A_{m,2}$ decreases with cutting, and H falls more than would be predicted by Eq. 12b. [This type of impeller is often found in multistage pumps, particularly those in which the designer, driven by cost reduction goals, has minimized a) the axial length occupied by each stage and b) the number of stages, thereby pushing down the n_s of the individual stage to the point that a tolerable sacrifice in efficiency results.]

RADIAL DISCHARGE IMPELLERS Impellers of low specific speed may be cut successfully provided the following items are kept in mind:

1. The angle β_2 may change as D is reduced, but this usually can be corrected by filing the blade tips. (See the discussion on blade-tip filing that follows.)
2. Tapered blade tips will be thickened by cutting and should be filed to restore the original shape. (See the discussion below on blade-tip filing.)
3. Bearing and stuffing box friction remain constant, but disk friction should decrease with decreasing D .
4. The length of flow path in the pump casing is increased by decreasing D .
5. Because $c_{m,1}$ is smaller at the reduced flow rate, the inlet triangles no longer remain similar before and after cutting, and local flow separation may take place near the blade entrance tips.
6. The second right-hand term in Eq. 5 was neglected in arriving at Eqs. 12, but it may represent a significant decrease in head as D is reduced.
7. Some blade overlap should be maintained after cutting. Usually the initial blade overlap decreases with increasing specific speed, so the higher the specific speed, the less the allowable diameter reduction.

8. Diameter reductions greater than from 10 to 20% of the original full diameter of the impeller are rarely made.

Most of the losses are approximately proportional to Q^2 , and hence to D^2 by Eq. 12. Because the power output decreases approximately as D^3 , it is reasonable to expect the maximum efficiency to decrease as the wheel is cut, and this often is the case. By Eq. 12 and the n_s -definition (Eq. 38a of Section 2.1), the product $n_s D$ should remain constant so the specific speed at best efficiency increases as the wheel diameter is reduced (Table 4).

The characteristics of the pump shown in Figure 13 may be used to illustrate reduction of diameter at constant speed. Starting with the best efficiency point and $D = 16\frac{5}{16}$ in (41.4 cm), let it be required to reduce the head from $H = 224.4$ to $H' = 192.9$ ft (68.4 m to 58.8 m) and to determine the wheel diameter, capacity, and power for the new conditions.

Because the speed is constant, Eqs. 12 may be written

$$H = k_H Q^2 \quad \text{and} \quad P = k_p Q^3 \quad (13)$$

where k_H and k_p may be obtained from the known operating conditions at $D = 16\frac{5}{16}$ in (41.4 cm). Plot a few points for assumed flow rates and draw the curve segments as shown by the solid lines in Figures 13b and 13c. Then, from Eqs. 12

$$D' = D\sqrt{H'/H} \quad Q' = Q\sqrt{H'/H} \quad P' = P(H'/H)^{3/2} \quad (14a)$$

$$D' = D(Q'/Q) \quad H' = H(Q'/Q)^2 \quad P' = P(Q'/Q)^3 \quad (14b)$$

from which $D = 15\frac{1}{8}$ in (38.4 cm), $Q = 3709$ gpm (234 l/s), and $P' = 215.5$ hp (160.7 kW). In Figure 13, the initial conditions were at points A and the computed conditions after cutting at points B. The test curve for $D = 15\frac{1}{8}$ in (38.4 cm) shows the best efficiency point a at a lower flow rate than predicted by Eqs. 14, but the head curve satisfies the predicted values very closely. The power prediction was not quite as good. Table 4 and Figure 13 give actual and predicted performance for three impeller diameters. The error in predicting the best efficiency point was computed by (predicted value minus test value) (100)/(test value). As the wheel diameter was reduced, the best efficiency point moved to a lower flow rate than predicted by Eq. 14 and the specific speed increased, showing that the conditions for Eqs. 12 to hold were not maintained.

Wheel cutting should be done in two or more steps with a test after each cut to avoid too large a reduction in diameter. Figure 14 shows an approximate correction, given by Stepanoff,¹² that may be applied to the ratio D'/D as computed by Eqs. 12 or 14. The accuracy of the correction decreases with increasing specific speed. Figure 15 shows a correction proposed by Rüttschi¹³ on the basis of extensive tests on low-specific-speed pumps. The corrected diameter reduction ΔD is the diameter reduction $D - D'$ given by Eqs. 14 and multiplied by k from Figure 15. The shaded area in Figure 15 indicates the range of scatter of the test points operating at or near maximum efficiency. Near shutoff the values of k were smaller and at maximum flow rate the values of k were larger than shown in Figure 15. Table 5 shows the results of applying Figures 14 and 15 to the pump of the preceding example.

There is no independent control of Q and H in impeller cutting, although Q may be increased somewhat by underfiling the blade tips as described later. The flow rate and power will automatically adjust to the values at which the pump head satisfies the system head-flow curve.

MIXED-FLOW IMPELLERS Diameter reduction of mixed-flow impellers is usually done by cutting a maximum at the outside diameter D_o and little or nothing at the inside diameter D_i , as shown in Figure 16. Stepanoff¹⁴ recommends that the calculations be based on the average diameter $D_m = (D_o + D_i)/2$ or estimated from the blade-length ratio FK/EK or GK/EK in Figure 16d. Figure 16 shows a portion of the characteristics of a mixed-flow impeller on which two cuts were made as in Figure 16b. The calculations were made by Eqs. 14 using the mean diameter

$$D_m = \sqrt{(D_o^2 + D_i^2)/2}$$

TABLE 4 Predicted characteristics at different impeller diameters on a radial-flow pump

	Test values					Predicted from D = 16 $\frac{5}{16}$ in (41.4 cm)			Predicted from D = 15 $\frac{1}{8}$ in (88.4 cm)		
D, in (cm)	Q, gpm (l/s)	H, ft (m)	P, hp (kW)	n_s (Ω_s)	$n_s D$ ($\Omega_s D$)	Q', gpm (l/s)	H', ft (m)	P', hp (kW)	Q', gpm (l/s)	H' ft (m)	P', hp (kW)
16 $\frac{5}{16}$ (41.4)	4000 (252)	224.4 (68.4)	270.4 (201.6)	1953 (0.7146)	31,860 (29.54)	...	...	...	3888 (245)	227.3 (69.3)	272.2 (203.0)
									-2.93% error ^a	1.29% error ^a	0.68% error ^a
15 $\frac{1}{8}$ (38.4)	3600 (227.1)	195.4 (59.6)	217.0 (161.8)	2055 (0.7519)	31,080 (28.87)	3709 (234)	192.9 (58.8)	215.5 (160.7)	...	...	...
						3.02% error ^a	-1.28% error ^a	-0.69% error ^a			
14 (35.6)	3200 (201.9)	163.6 (49.9)	167.2 (124.7)	2214 (0.8101)	31,000 (28.84)	3433 (217)	165.3 (50.4)	170.9 (127.4)	3332 (210.2)	167.4 (51.0)	172.1 (128.3)
						7.28% error ^a	1.03% error ^a	2.33% error ^a	4.13% error ^a	2.33% error ^a	2.93% error ^a

^aError in predicting best efficiency point

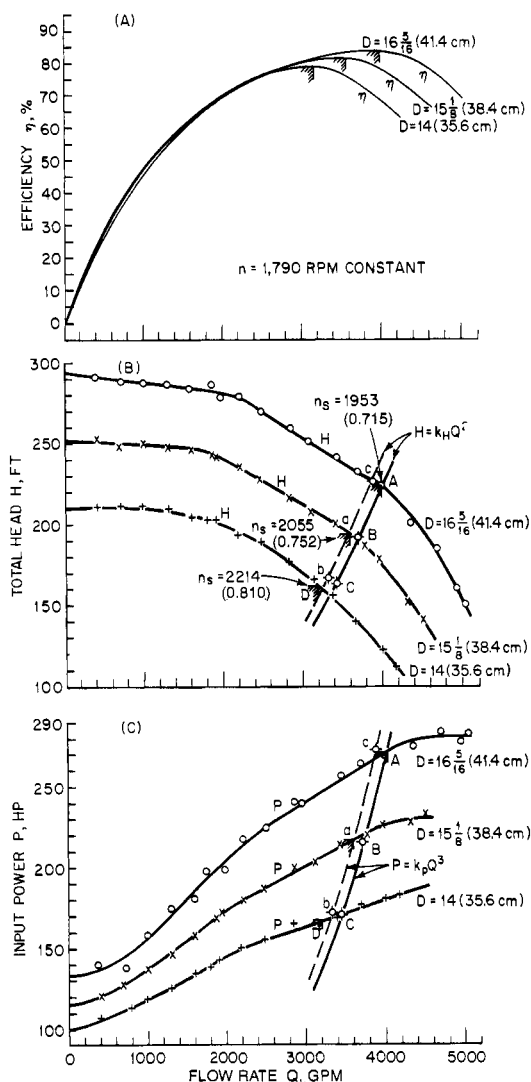


FIGURE 13A through C Diameter reduction of radial-flow impeller ($\text{gpm} \times 0.06309 = \text{l/s}$). D is measured in inches (centimeters)

instead of the outside diameter in each case. The predictions and test results are shown in Figure 16 and Table 6. It is clear that the actual change in the characteristics far exceeded the predicted values. Except for the use of the mean diameters, the procedure was essentially the same as that described for Figure 13, and all points and curves are similarly labeled. The corrections given in Figure 14 would have made very little difference in the computed diameter reductions, and those of Figure 15 were not applicable to impellers having specific speeds greater than $n_s = 2000$ (0.73). In this case, the product $n_s D_m$ did not remain constant and the maximum efficiency increased as D_m was reduced. Although the

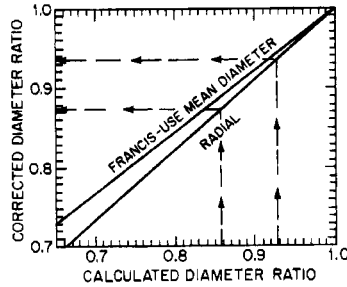


FIGURE 14 Corrections for calculated impeller diameter reductions (Reference 12)

changes in diameter were small, the area of blade removed was rather large for each cut. The second cut eliminated most of the blade overlap.

The characteristics of mixed-flow impellers can be changed by cutting, but very small cuts may produce a significant effect. The impellers of propeller pumps are not usually subject to diameter reduction.

Shaping Blade Tips If the discharge tips of the impeller blades are thick, performance usually can be improved by filing over a sufficient length of blade to produce a long, gradual taper. Chamfering, or rounding, the discharge tips may increase the losses and should never be done. Reducing the impeller diameter frequently increases the tip thickness.

OVERFILING This is shown at *B* in Figure 17, and the unfiled blade is shown at *A*. Usually there is little or no increase in the blade spacing d before and d_f after filing, so the discharge area is practically unchanged. Experience indicates that any change in the angle β_2 due to overfiling usually produces a negligible change in performance.

UNDERFILING This is shown at *C* in Figure 17. If properly done, underfiling will increase the blade spacing from d to d_f and hence the discharge area, which lowers the average meridional velocity c_{m2} at any given flow rate Q . The angle β_2 usually is increased slightly. Figure 18A and Eq. 6 show that the head and consequently the power increase at the same flow rate. The maximum efficiency usually is improved and may be moved to a higher flow rate. At the same head, Figure 18B shows that both c_{m2} and the flow rate will increase. The change both in the area and in C_{m2} may increase the flow rate by as much as 10%. Table 7 shows the results of tests before and after underfiling the impeller blades of nine different pumps. In general, they confirm the foregoing predictions based on changes in area and in the velocity triangles.

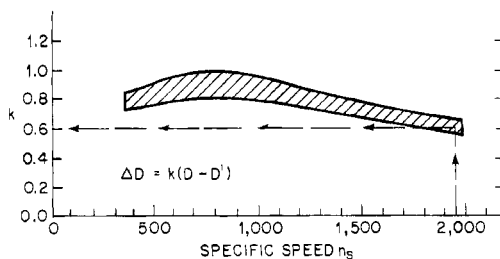


FIGURE 15 Corrections for calculated impeller diameter reductions (Reference 13). ($\Omega_s = n_s/2733$)

TABLE 5 Impeller diameter corrections

D before cutting	in	16.3125	16.3125
D' predicted by Eqs. (14)	in	15.125	14.000
D' corrected by Figure 14	in	15.25 ^a	14.26 ^b
D' corrected by Figure 15	in	15.60 ^c	14.93 ^d

^a $D'/D = 15.125/16.3125 = 0.927$; by Figure 14, corrected $D'/D = 0.935$. Corrected $D' = (0.935)(16.3125) = 15.25$ in.

^b $D'/D = 14.000/16.3125 = 0.858$; by Figure 14, corrected $D'/D = 0.874$. Corrected $D' = (0.874)(16.3125) = 14.26$ in.

^c $D - D' = 16.3125 - 15.125 = 1.1875$; by Figure 15, $K \cong 0.6$ at $n_s = 1,953$. Corrected $D - D' = (0.6)(1.1875) = 0.7125$ and corrected $D' = 16.3125 - 0.7125 = 15.60$ in.

^d $D - D' = 16.3125 - 14.000 = 2.3125$; by Figure 15, $K \cong 0.6$ at $n_s = 1,953$. Corrected $D - D' = (0.6)(2.3125) = 1.3875$ and corrected $D' = 16.3125 - 1.3875 = 14.93$ in.

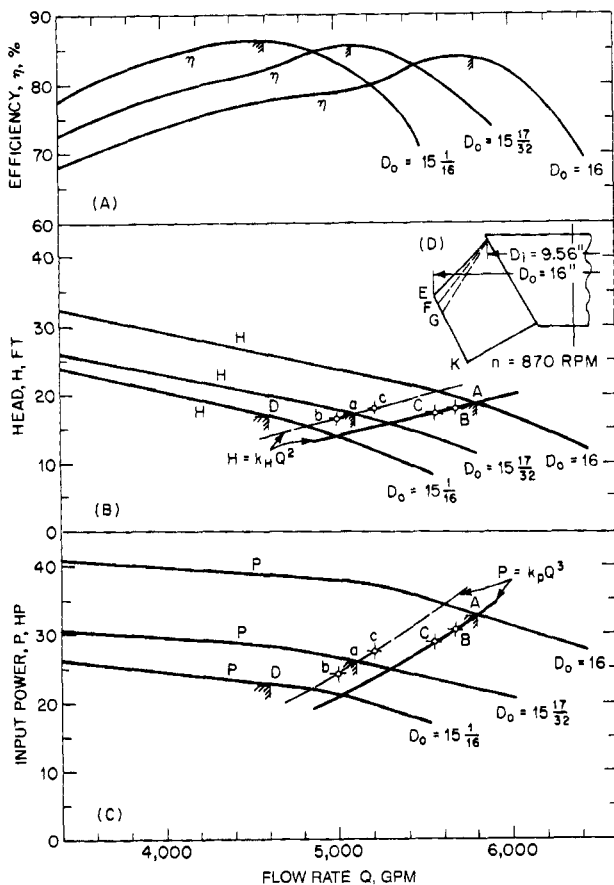


FIGURE 16A through C Diameter reduction of mixed-flow impeller (ft $\times 0.3048$ = m; in $\times 2.540$ = cm; gpm $\times 0.06309$ = l/s; hp $\times 0.7457$ = kW)

TABLE 6 Predicted characteristics at different impeller diameters on a mixed-flow pump

		Test values					Predicted from $D = 16.00$ (40.64 cm), $D_m = 13.17$ (33.45 cm)			Predicted from $D = 15.53$ (39.45 cm), $D_m = 12.89$ (32.74 cm)		
D , in (cm)	D_m , in (cm)	Q , gpm (l/s)	H , ft (m)	P , hp (kW)	n_s	$n_s D_m$	Q' , gpm (l/s)	H' , ft (m)	P' , hp (kW)	Q' , gpm (l/s)	H' , ft (m)	P' , hp (kW)
16 (40.64)	13.17 (33.45)	5800 (365.9)	18.6 (5.67)	32.5 (24.2)	7385 (2.702)	97,300 (90.39)	...	...	...	5210 (329)	17.9 (5.46)	27.4 (20.4)
										-11.3% error	-3.91% error	-18.6% error
15 $\frac{17}{32}$ (39.45)	12.89 (32.74)	5100 (321.8)	17.1 (5.21)	25.7 (19.2)	7385 (2.702)	95,200 (88.47)	5680 (358.4)	17.8 (5.43)	30.4 (22.7)			
							10.2% error ^a	4.50% error ^a	15.5% error ^a			
15 $\frac{1}{16}$ (38.26)	12.62 (32.05)	4600 (290.2)	16.8 (5.12)	22.6 (17.1)	7100 (2.598)	89,600 (83.26)	5560 (350.8)	17.1 (5.21)	28.6 (21.6)	5000 (315.5)	16.4 (5.00)	24.1 (18.2)
							17.3% error ^a	1.75% error ^a	21.0% error ^a	8.00% error ^a	-2.44% error ^a	6.22% error ^a

^aError in predicting best efficiency point

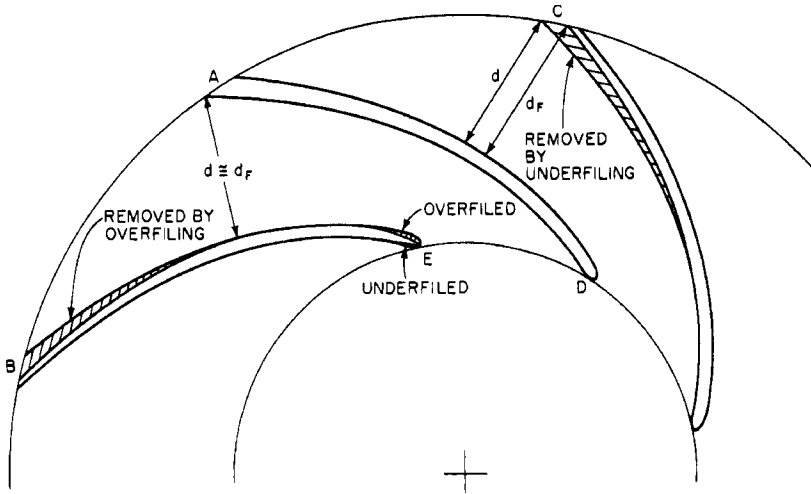


FIGURE 17 Underfiling and overfiling of blade tips

INLET BLADE TIPS If the inlet blade tips are blunt, as shown at *D* in Figure 17, the cavitation characteristics may be improved by sharpening them, as shown at *E*. In this case overfiling increases the effective flow area, which reduces c_{m1} for a given flow rate. If more area is needed, it may be advantageous to cut back part of the blade and sharpen the leading edge. Overfiling tends to increase β_1 , which is incompatible with a decrease in c_{m1} (Figure 19). The increase in β_1 increases the angle of attack of the liquid approaching the blade. In Figures 2 and 19, w_1 is tangent to the centerline of the blade at entrance and w_0

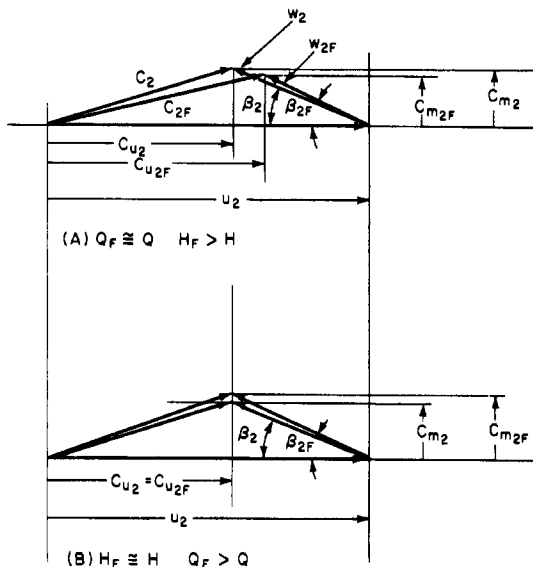


FIGURE 18 Discharge velocity triangles for underfiling blades (neglecting slip)

TABLE 7 Changes in performance when impeller blades are underfiled

Specific speed n_s	Ω_s	No. of stages	Impeller diameter D_2 in (cm)	Change in blade spacing d_F/d	$\frac{H_F - H}{H}$ % ^b	Changes at best efficiency point after filing, %				
						H_o^c	H	Q	P	η
862	(0.3154)	1	11½ (29.21)	1.13	4	1.5	4	0	9	-5
945	(0.3458)	2	8⅝ (21.27)	1.05	5.5	0	3	4	-0.5	7.5 ^d
1000	(0.3659)	4	9⅝ (23.18)	1.055	5.5	0	5.5	0	3	2.5
1080	(0.3952)	2	9⅓ ₁₆ (24.92)	1.08	10	2.5	10	0	6.8	3
1525	(0.5580)	2	10½ (26.67)	1.035	3.2	3	0.5	4.5	3.5	1.2
1950	(0.7135)	1	22 (55.88)	1.02	1.5	1	1.5	0	1.5	0
3300 ^a	(1.2075)	1	12 (30.48)	...	7.8	2.5	7.8	0	8.5	-0.6
3450 ^a	(1.2623)	1	30½ (77.47)	...	6.5	0.5	6.5	0	7.8	-2.2
4300	(1.5734)	1	41¼ (104.8)	...	5		5	0	0.5	4.5 ^e

^aDouble suction.
^b H_F is head after filing.
^cShutoff head.
^dDue in part to changes in pump casing.
^eDue in part to rounding of inlet blade tips.
Source: Flowserve Corporation

is the velocity of the approaching liquid as seen by an observer moving with the blade. The angle δ between w_o and w_1 is the angle of attack, which increases to δ_F after overfiling. Although the increase in δ tends to increase the opportunity for the liquid to separate from the low-pressure face of the blade, this disadvantage usually is outweighed by the improvement that results when c_{m1} is reduced.*

CASING TONGUE The casing tongue or cutwater forms part of the throat of the discharge nozzle of many volute casings (Figure 20a, Section 2.1.1). Frequently the throat area is small enough to act as a throttle and reduce the maximum flow rate otherwise obtainable from the impeller. Cutting back the tongue increases the throat area and increases the maximum flow rate. The head-versus-flow rate characteristic is then said to *carry out farther*. Shortening the discharge nozzle may increase the diffusion losses a little and result in a slightly lower efficiency.

CAVITATION

The formation and subsequent collapse of vapor-filled cavities in a liquid due to dynamic action are called *cavitation*. The cavities may be bubbles, vapor-filled pockets, or a combination of both. The local pressure must be at or below the vapor pressure of the liquid for cavitation to begin, and the cavities must encounter a region of pressure higher than the vapor pressure in order to collapse. Dissolved gases often are liberated shortly before vaporization begins. This may be an indication of impending cavitation, but true cavitation

*The distinction between a true velocity diagram formed just upstream of the impeller blades and the triangles shown just inside the blades in Figure 19 should be noted. w_1 in this figure is shown aligned with the blade, having undergone an adjustment in direction via incidence from that of the approaching w_o . Only by plotting the velocity diagram just upstream of this point can one determine the true value of the circumferential component c_{u1} of the absolute velocity being delivered to the impeller by the upstream piping, vanes, and so on. It is this delivered value of c_{u1} that must be used in computing input or ideal head of the impeller (Eq. 15b, Section 2.1.1 or, neglecting slip, Eq. 6 of this section). Thus w_o is properly the true w_1 vector and should be understood as such (see Section 2.1.1).

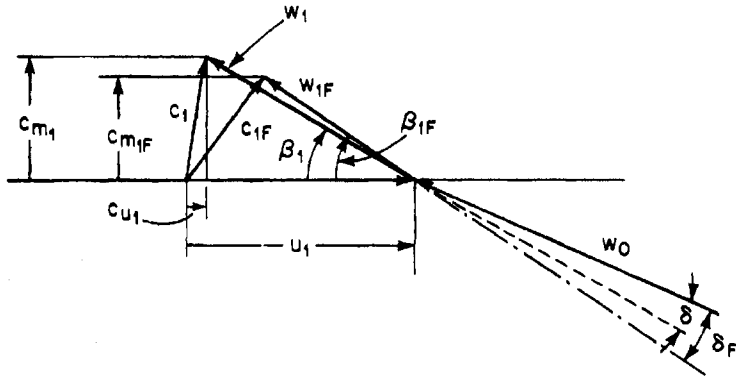


FIGURE 19 Effect of overfiling at inlet on the velocity triangle just inside the blades

requires vaporization of the liquid. Boiling accomplished by the addition of heat or the reduction of static pressure without dynamic action of the liquid is arbitrarily excluded from the definition of cavitation.

When a liquid flows over a surface having convex curvature, the pressure near the surface is lowered and the flow tends to separate from the surface. *Separation* and *cavitation* are completely different phenomena. Without cavitation, a separated region contains turbulent eddying liquid at pressures higher than the vapor pressure. When the pressure is low enough, the separated region may contain a vapor pocket that fills from the downstream end,¹⁵ collapses, and forms again many times each second. This causes noise and, if severe enough, vibration. Vapor-filled bubbles usually are present and collapse very rapidly in any region where the pressure is above the vapor pressure. Knapp¹⁶ found the life cycle of a bubble to be on the order of 0.003 s.

Bubbles that collapse on a solid boundary may cause severe mechanical damage. Shuttler and Mesler¹⁷ photographed bubbles that distorted into toroidal-shaped rings during collapse and produced ring-shaped indentations in a soft metal boundary. The bubbles rebounded following the initial collapse and caused pitting of the boundary. Pressures on the order of 10^4 atm have been estimated¹⁸ during collapse of a bubble. All known materials can be damaged by exposure to bubble collapse for a sufficiently long time. This is properly called *cavitation erosion*, or *pitting*. Figure 20 shows extensive damage to the suction side of pump impeller vanes after about three months' operation with cavitation. At two locations, the pitting has penetrated deeply into the $\frac{3}{8}$ -in (9.5-mm) thickness of stainless steel. The unfavorable inlet flow conditions, believed to have been the cause of the cavitation, were at least partly due to elbows in the approach piping. Modifications in the approach piping and the pump inlet passages reduced the cavitation enough to extend impeller life to several years.¹⁹

It has been postulated that high temperatures and chemical action may be present at bubble collapse, but any damaging effects due to them appear to be secondary to the mechanical action. It seems possible that erosion by foreign materials in the liquid and cavitation pitting may augment each other. Controlled experiments²⁰ with water indicated that the damage to metal depends on the liquid temperature and was a maximum at about 100 to 120°F (38 to 49°C). Cavitation pitting, as measured by weight of the boundary material removed per unit time, frequently increases with time. Cast iron and steel boundaries are particularly vulnerable. Controlled experiments have shown that cavitation pitting in metals such as aluminum, steel, and stainless steel depends strongly on the velocity of the fluid in the undisturbed flow past the boundary. On the basis of tests of short duration, Knapp¹⁵ reported that damage to annealed aluminum increased approximately as the sixth power of the velocity of the undisturbed flow past the surface. Hammitt²¹ found a more complicated relationship between velocity and damage, as shown in Figure 21. It seems clear that after cavitation begins, it will increase rapidly with increasing velocities. Frequently the rate of pitting accelerates with elapsed time. Comprehensive discussions of cavitation and erosion together with additional references are given by Preece.²²

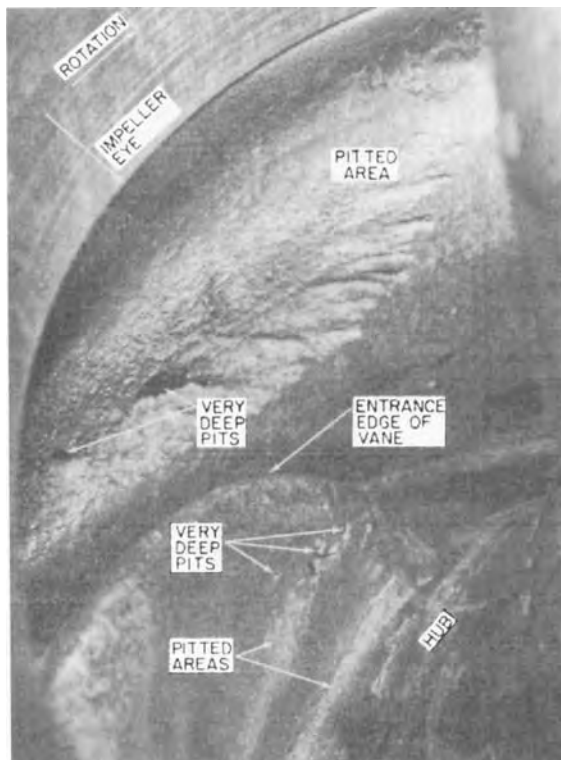


FIGURE 20 Impeller damaged by cavitation (Demag Delaval, Reference 19)

Centrifugal pumps begin to cavitate when the suction head is insufficient to maintain pressures above the vapor pressure throughout the flow passages. The most sensitive areas usually are the low-pressure sides of the impeller vanes near the inlet edge and the front shroud, where the curvature is greatest. Axial-flow and high-specific-speed impellers without front shrouds are especially sensitive to cavitation on the low-pressure sides of the blade tips and in the close tip-clearance spaces. Sensitive areas in the pump

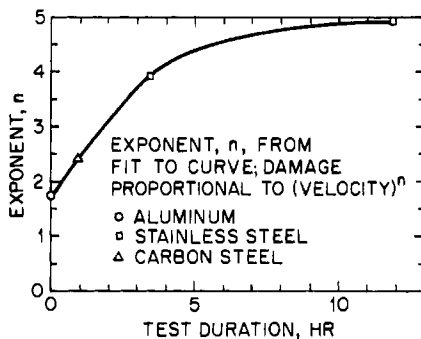


FIGURE 21 Cavitation damage exponent versus test time for several materials in water (Reference 21)

casing include the low-pressure side of the tongue and the low-pressure sides of diffusion vanes near the inlet edges. As the suction head is reduced, all existing areas of cavitation tend to increase and additional areas may develop. Apart from the noise and vibration, cavitation damage may render an impeller useless in as little as a few weeks of continuous operation. In multistage pumps cavitation usually is limited to the first stage, but Kovats²³ has pointed out that second and higher stages may cavitate if the flow is reduced by lowering the suction head (submergence control). Cavitation tends to lower the axial thrust of an impeller. This could impair the balancing of multistage pumps with opposed impellers. A reduction in suction pressure may cause the flow past a balancing drum or disk to cavitate where the liquid discharges from the narrow clearance space. This may produce vibration and damage resulting from contact between fixed and running surfaces.

Net Positive Suction Head (NPSH) Several criteria exist for establishing the NPSH of a pump. These are connected with a) inception of cavitation, b) loss of hydraulic performance, and c) protecting the pump against cavitation erosion or damage. These are defined and found as follows:

- a. **Inception NPSH.** Cavitation can be completely prevented so long as the static pressure within the pump is everywhere greater than the vapor pressure of the liquid. This can be achieved if the NPSH (also called h_{wv}), defined in Section 2.1.1 as the total head of the pumped liquid at the pump inlet datum or suction flange above that vapor pressure, is sufficiently large. The value of NPSH that achieves this is called $NPSH_i$, namely, the “inception NPSH.”
- b. **Performance NPSH.** In a typical NPSH-test at constant speed n and flow rate Q , a substantial reduction of NPSH below $NPSH_i$ is usually necessary to reach the value that produces an identifiable drop in performance—usually 3% of pump stage total pressure rise or pump head. This value is called the “required NPSH,” $NPSHR$ or $NPSH_{3\%}$. Table 1 in Section 2.1.1 provides empirical correlations for $NPSH_{3\%}$ for common pumps and inducers. Other criteria for $NPSH_{3\%}$ will be given further on. The domains of cavitation within a pump are widespread at this condition, and experience shows that a head drop in the neighborhood of 3% must occur in order to obtain a repeatable value of $NPSHR$ on test¹². Lower percentages have been demanded by users, but they invariably give rise to a large scatter in the measured $NPSHR$ for even small variations in any of the test variables.

It used to be thought throughout the pump community that there are no cavities or bubbles present at zero percent head drop due to cavitation activity within a pump. In recent decades, however, it has been proven through many observations that not only is the head drop equal to zero at inception (where $NPSH = NPSH_i$), it remains so for an enormous range of lesser NPSH-values over which extensive bubble activity is observed. In fact, $NPSH_i$ is commonly from two to five times the magnitude of the $NPSHR$ that is associated with any noticeable drop in pump head. At the 3% (or lesser percentage) head drop condition, actual observations of the cavitating flow show the cavities to extend all the way from the leading edge of each blade to the throat formed by the leading edge of the next blade²⁴. In the face of these learnings, it is evident that to demand a test for the NPSH required for a head drop of much less than 3% usually causes only needless misunderstandings and expenditure of time and money.

- c. **Damage-Limiting NPSH or Life NPSH.** If it is desired to substantially reduce or eliminate cavitation activity within the pump, an acceptance test should be conducted wherein the cavitating flow can be observed visually²⁵. This is indeed a serious issue if the pump energy level is high enough (and therefore the inlet pressure) for the collapsing cavities to do damage (as in Figure 20; see also Section 2.1.1). In such a case, “ $NPSH_{3\%}$ ” has virtually no meaning; rather, the truly *required* NPSH or “ $NPSHR$ ” is that *larger* value which satisfactorily limits the extent of this visually observed cavitation within the pump or pump model²⁶.

Therefore, the *available* NPSH at the installation must be at least equal to the *required* NPSH if the above consequences are to be avoided—be they significant loss of

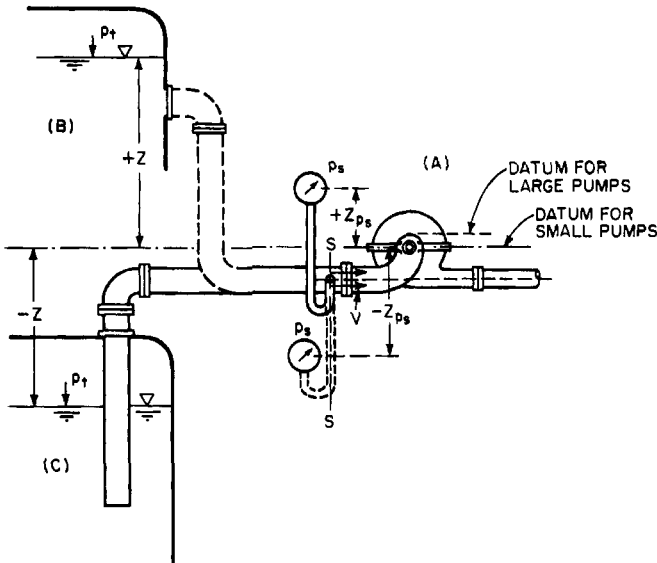


FIGURE 22 Definition sketch for computing NPSH

pump head (that is, loss of hydraulic performance) or damage due to collapsing cavities. Increasing the available NPSH above the required value provides a margin of safety against unpredictable variations of conditions, including transient behavior. Figure 22 and the following symbols will be used to compute the available NPSH:

p_a = absolute pressure in atmosphere surrounding gage, Figure 22

p_s = gage pressure indicated by gage or manometer connected to pump suction at section s-s; may be positive or negative

p_t = absolute pressure on free surface of liquid in closed tank connected to pump suction

p_{vp} = vapor pressure of liquid being pumped corresponding to the temperature at section s-s (if liquid is a mixture of hydrocarbons, p_{vp} must be measured by the *bubble point* method)

h_f = lost head due to friction in suction line between tank and section s-s

V = average velocity at section s-s

Z, Z_{ps} = vertical distances defined by Figure 22; may be positive or negative

γ = specific weight of liquid at pumping temperature

It is satisfactory to choose the datum for small pumps as shown in Figures 1 and 22, but with large pumps, the datum should be raised to the elevation where cavitation is most likely to start. For example, the datum for a large horizontal-shaft propeller pump should be taken at the highest elevation of the impeller-blade tips. The available NPSH is given by

$$h_{sv} = \frac{p_a - p_{vp}}{\gamma} + \frac{p_s}{\gamma} + Z_{ps} + \frac{V^2}{2g} \quad (15)$$

or

$$h_{sv} = \frac{p_t - p_{vp}}{\gamma} + Z - h_f \quad (16)$$

Consistent units must be chosen so that each term in Eqs. 15 and 16 represents feet (or meters) of the liquid pumped. Equation 15 is useful for evaluating the results of tests. Equation 16 is useful for estimating available NPSH during the design phase of an installation. In Eq. 15, the first term represents the height of a liquid barometer, h_b , containing the liquid being pumped and the sum of the remaining terms represents the suction head h_s ; that is, the total head H (Eq. 3, Section 2.1.1) evaluated at the suction flange S in Figure 22. Therefore

$$h_{sv} = h_b + h_s \quad (17)$$

Usually, a positive value of h_s is called a *suction head* and a negative value of h_s is called a *suction lift*.

Many pumps draw cold water from reservoirs exposed to atmospheric pressure, so the suction lift is limited if there is to be a reasonable value of h_{sv} (= NPSH) available at the pump suction flange or port. In normal practice, rather small values of suction lift or suction head are encountered in many installations, so one can depend on the value of h_{sv} being a substantial portion of h_b —say about 60% or 20 ft (6m) of water column. Recognizing that the suction specific speed capability S of most pumps falls within a rather small range has led to charts of recommended operating speeds such as those in Section 12.11 or the speed recommendations of Figure 23 for condensate pumps for varying amounts of NPSH available. In fact, $S = 8500$ ($\Omega_{ss} = 3.11$) is recommended by the Hydraulic Institute²⁷. S ($= N_{ss}$) and Ω_{ss} are defined in Eqs. 41 and 42 of Section 2.1.1. The relationship of suction-specific speed to other suction parameters is treated further on.

Cavitation Tests In addition to the constant- Q tests for $NPSHR$ that were described in the foregoing review of the various NPSH-criteria, one can establish the curve of

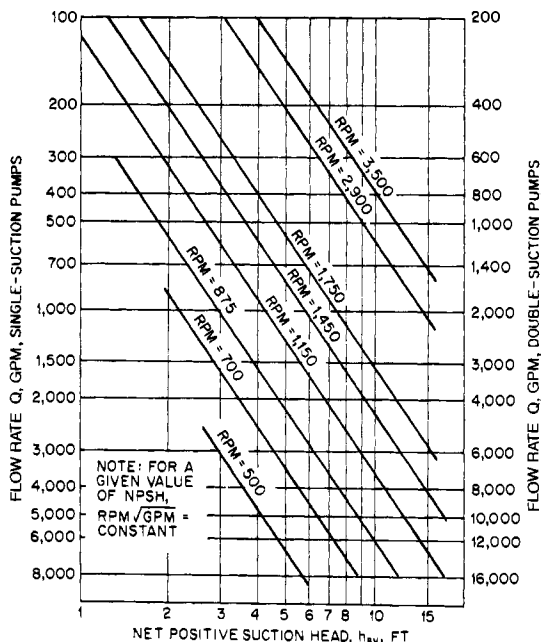


FIGURE 23 Capacity (flow rate) and speed limitations for condensate pumps with shaft through eye of impeller ($\text{ft} \times 0.3048 = \text{m}$; $\text{gpm} \times 0.06309 = \text{l/s}$) (adapted from Hydraulic Institute ANSI/HI 2000 Edition Pump Standards, Reference 27)

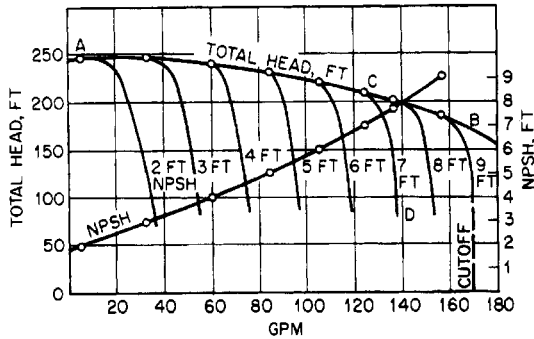


FIGURE 24 Test of a 1.5-in (3.81-cm) single-stage pump at 3470 rpm, on water, 70°F (21°C) (ft × 0.3048 = m; gpm × 0.06309 = l/s) (Reference 12)

performance NPSH versus flow rate by varying the NPSH available and seeing how much flow rate Q the pump can handle. This is illustrated in Figure 24, wherein a different head curve is found for each value of NPSH. For each such curve, as Q is increased, there is a point at which the head departs from the next one of higher NPSH. This, then, is the Q for which the given NPSH is that required to maintain head, essentially $NPSH_{3\%}$. At higher flow rate, performance is lost, each head curve possessing a “cutoff” flow rate. References 27 and 28 may be consulted for details of test procedures.

Thoma Cavitation Parameter σ All the terms in Eqs. 15 to 17 may be made dimensionless by dividing each by the pump head H . The resulting parameter

$$\sigma = \frac{h_{gv}}{H} \tag{18}$$

has proven to be useful especially for high-specific-speed pumps and turbines. The loading on the blades of a variable-pitch propeller, and therefore the minimum suction-side static pressure, is directly connected to the head developed. As this head increases with increasing blade setting angle, the loading increases, the minimum pressure drops, and the $NPSHR$ goes up. Hence an identifiable limiting value of σ is found to exist for such a machine. σ -limits are presented for a range of specific speeds in Figure 25.

Performance-NPSH data are sometimes given in terms of σ ; for example, Rüttschi¹³ presents the effect of pump hydraulic efficiency on NPSH in these terms, a consistently higher value of σ being required for pumps with lower efficiency (Figure 26).

Suction-Specific Speed For pumps in which the eye diameter of the impeller is smaller than that at exit, the pressures on the suction-sides of the blades that are associated with most of the head addition are in a region of higher pressure than in the vicinity of the eye. Thus Wislicenus²⁹ was able to decouple the eye from outer diameter of the pump (where most of the head is generated in a radial-flow machine) and show that only the eye geometry determines performance-NPSH and not the magnitude of the head created by the impeller. He demonstrated that for most centrifugal pumps (except the propellers just described), there is a small range of suction specific speed—from which the performance-NPSH can be determined. As in Section 2.1.1, this parameter is defined as follows:

$$S = \frac{N\sqrt{Q}}{(h_{gv})^{3/4}} \tag{19}$$

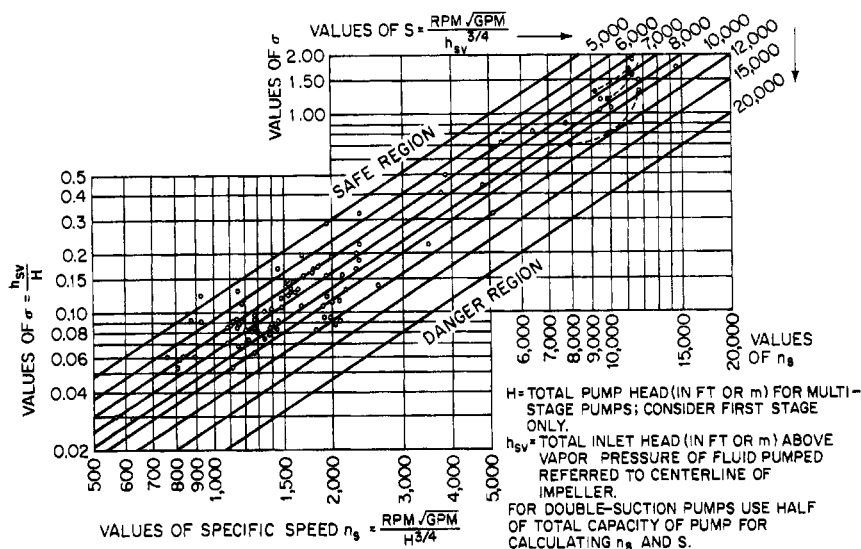


FIGURE 25 Cavitation limits of centrifugal and propeller pumps (Flowserve Corporation)

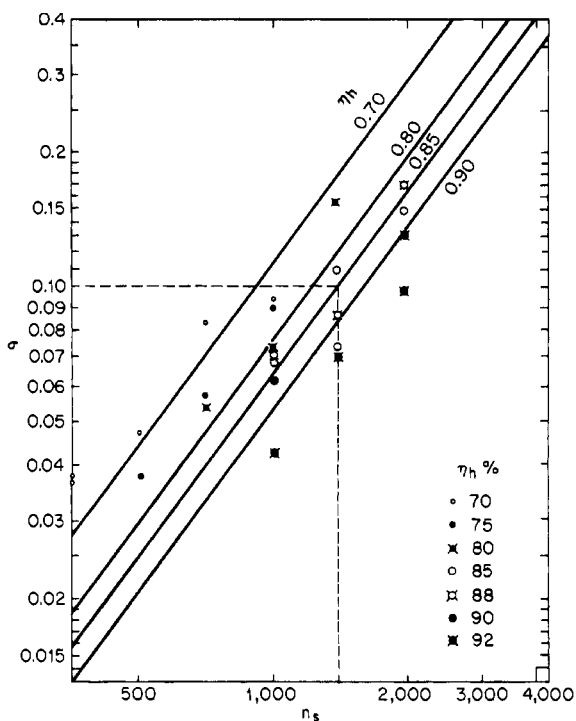


FIGURE 26 Cavitation parameter σ versus specific speed for different efficiencies ($\Omega_c = n_s/2733$) (Reference 13)

Note that Q = half the discharge of a double-suction impeller when computing S . Equations 18 and 19 and the definition of n_s may be combined to yield

$$\sigma = \left(\frac{n_s}{S} \right)^{4/3} \quad (20)$$

$$n_s = S(\sigma)^{3/4} \quad (21)$$

Figure 25 shows lines of constant S . For most pumps, $7500 < S < 11,000$ ($2.7 < \Omega_{ss} < 4.0$), $S = 8500$ (3.11) being recommended by the Hydraulic Institute ANSI/HI 2000 Pump Standards (Reference 27). Higher values may apply to special designs or service conditions, such as an inducer ahead of the first-stage impeller. For a given specific speed, the lower the value of S , the safer the pump against cavitation. Experience with large European pumped storage installations has shown that cavitation effects began at $S \approx 6000$ (2.2), and this value is recommended for large pumps. Figure 27 shows a summary of data and formulas that may be useful with commercial pumps.

German practice differs considerably from that in the United States in computing suction specific speed. Pfleiderer¹ defined a hub correction k as

$$k = 1 - \left(\frac{d_h}{D_o} \right)^2 \quad (22)$$

where d_h = the hub diameter and D_o = the diameter of the suction nozzle, in any consistent units. The suction specific speed S_G is defined as

$$S_G = \frac{(n/100)^2 Q}{k h_{sv}^{3/2}} \quad (23)$$

where n is measured in rpm, Q in cubic meters per second per impeller inlet, and h_{sv} in meters of liquid pumped. It follows that

$$S = 5164 \sqrt{S_G k} \quad (24)$$

NPSH for Liquids Other Than Cold Water Field experience, together with carefully controlled laboratory experiments, has indicated that pumps handling hot water or certain

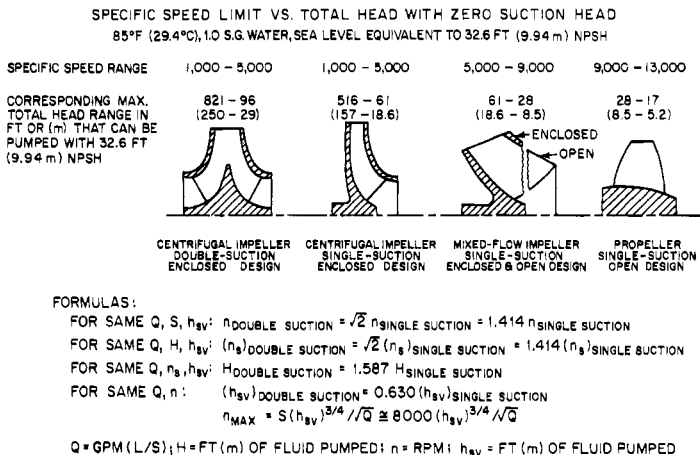


FIGURE 27 Commercial pumps applicable to zero suction head. Suction specific speed $S = 8000$ ($\Omega_{ss} = 2.93$). For centrifugal impellers, $n_s = 1,000$ –5,000 ($\Omega_s = 0.4$ –1.8); mixed-flow, 5,000–9,000 (1.8–3.3); propeller, 9,000–13,000 (1.3–4.8).

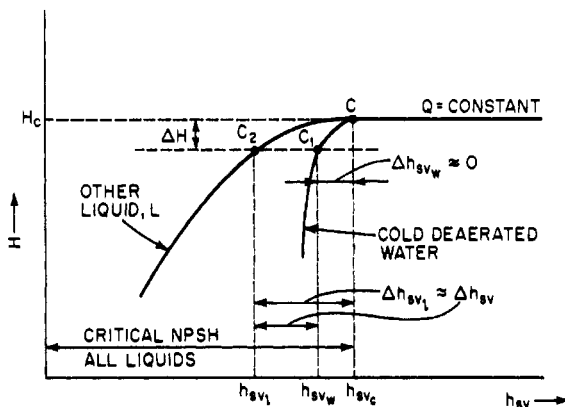


FIGURE 28 Cavitation tests with different liquids at constant speed and constant flow rate (Reference 34)

liquid hydrocarbons may be operated safely with less NPSH than would normally be required for cold water. This may lower the cost of an installation appreciably, particularly in the case of refinery pumps. A theory for this has been given by Stepanoff and others.^{14,30-33} Figure 28 shows the results of cavitation tests on two liquids for constant capacity and constant pump speed. No head loss due to cavitation is present at point C or at $h_{sv} > h_{svc}$. With cold deaerated water, lowering h_{sv} slightly below h_{svc} produces limited cavitation and a decrease in pump head ΔH to point C_1 , but Δh_{svw} usually is negligible. With hot water ($T \geq 100^\circ\text{F} = 37.8^\circ\text{C}$) or with many liquid hydrocarbons, a much larger decrease in h_{sv} will be required to produce the same drop in head ΔH that was shown by the cold water test. The NPSH reduction, or NPSH adjustment, is $\Delta h_{sv} \approx \Delta h_{svl}$. In practice, Δh has been limited to $\Delta h \leq 0.03H$, for which there is a negligible sacrifice in performance. The pumps usually are made of stainless steel or other cavitation-resistant materials, and the lower NPSH results in lower collapse pressure of the vapor bubbles, reducing the damage potential.

Chart for NPSH Reductions A composite chart of NPSH reductions for deaerated hot water and certain gas-free liquid hydrocarbons is shown in Figure 29. The curves of vapor pressure versus temperature and the curves of constant NPSH reduction were based on laboratory tests with the liquids shown and should be used subject to the following limitations:

1. No NPSH reduction should exceed 50% of the NPSH required by the pump for cold water or 10 ft (3.0 m), whichever is smaller.
2. NPSH may have to be increased above the normal cold-water value to avoid unsatisfactory operation when (a) *entrained* air or other noncondensable gas is present in the liquid or (b) *dissolved* air or other noncondensable gas is present in the liquid and the absolute suction pressure is low enough to permit release of the gas from solution.
3. The vapor pressure of hydrocarbon mixtures vary significantly with temperature and so should be determined at pumping temperature (see Reference 27).
4. If the suction system may be susceptible to transient changes in absolute pressure or temperature, a suitable margin of safety in NPSH should be provided. This is particularly important with hot water and may exceed the reduction that would otherwise apply with steady-state conditions.
5. Although experience has indicated the reliability of Figure 29 for hot water and the liquid hydrocarbons shown, its use with other liquids is not recommended unless it is clearly understood that the results must be accepted on an experimental basis.

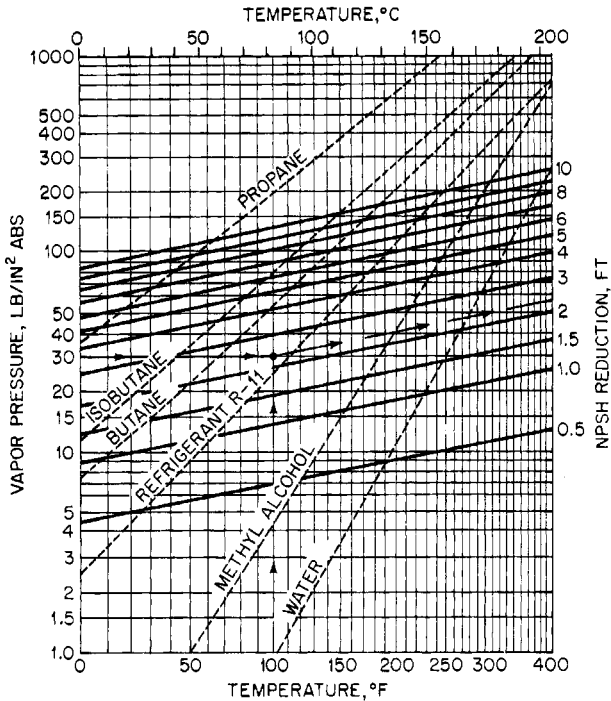


FIGURE 29 NPSH reductions for pumps handling liquid hydrocarbons and hot water. This chart has been constructed from test data obtained by using the liquids shown. For applicability to other liquids, refer to the text (ft $\times 0.3048$ = m; lb/in² $\times 6.895$ = kPa). (Hydraulic Institute ANSI/HI 2000 Edition Pump Standards, Reference 27)

USE OF FIGURE 29 Given a fluid having a vapor pressure of 30 lb/in² (210 kPa) abs at 100°F (37.8°C). Follow the arrows on the key shown on the chart and obtain an NPSH reduction of about 2.3 ft (0.70 m). Since this does not correspond to one of the liquids for which vapor pressure curves are shown on the chart, the use of this NPSH reduction should be considered a tentative value only. Given a pump requiring 16-ft (4.9-m) cold-water NPSH at the operating capacity, the pump is to handle propane at 55°F (12.8°C). Figure 29 shows the vapor pressure to be about 105 lb/in² (733 kPa) abs and the NPSH reduction to be about 9.5 ft (2.9 m). Since this exceeds 8 ft (2.44 m), which is half the cold-water NPSH, the recommended NPSH for the pump handling propane is half the cold-water NPSH, or 8 ft (2.44 m). If the temperature of the propane in the previous example is reduced to 14°F (−10°C), Figure 29 shows the vapor pressure to be 50 lb/in² (349 kPa) abs and the NPSH reduction to be about 5.7 ft (1.74 m), which is less than half the coldwater NPSH. The NPSH required for pumping propane at 14°F (−10°C) is then $16 - 5.7 = 10.3 \approx 10$ ft (4.87 − 1.74 = 3.13 ≈ 3 m).

Reduction of Cavitation Damage After the pump has been built and installed*, there is little that can be done to reduce cavitation damage. As previously mentioned, sharpening the leading edges of the blades by filing may be beneficial. Stepanoff² has suggested cutting back part of the blades in the impeller eye together with sharpening the tips, for low-specific-speed pumps, as a means of reducing the inlet velocity c_1 and thus lowering σ . Although a small amount of prerotation or prewhirl in the direction of impeller rotation

*Sometimes it is possible to lower the pump, and this should be considered before other alterations are made.

may be desirable⁸⁴, excessive amounts should be avoided. This may require straightening vanes ahead of the impeller and rearranging the suction piping to avoid changes in direction or other obstructions. The cavitation damage to the impeller shown in Figure 20 was believed to have been at least partly due to bad flow conditions produced by two 90° elbows in the suction piping. The planes of the elbows were at 90° to each other, and this arrangement should be avoided.

Straightening vanes in the impeller inlet may increase the NPSH requirement at all flow rates. Three or four radial ribs equally spaced around the inlet and extending inward about one-quarter of the inlet diameter are effective against excessive prerotation and may require less NPSH than full-length vanes. This is very important with axial-flow pumps, which are apt to have unfavorable cavitation characteristics at partial flow rates. Operation near the best efficiency point usually minimizes cavitation.

The admission of a small amount of air into the pump suction tends to reduce cavitation noise.⁷ This rarely is done, however, because it is difficult to inject the right amount of air under varying head and flow rate conditions and frequently there are objections to mixing air with the liquid pumped.

If a new impeller is required because of cavitation, the design should take into account the most recent advances described in the literature. Gongwer⁸⁵ has suggested (1) the use of ample fillets where the blades join the shrouds, (2) sharpened leading edges of blades, (3) reduction of β , in the immediate vicinity of the shrouds, and (4) raking the leading edges of the blades forward out of the eye. Increasing the number of blades for propeller pumps lowers σ for a given submergence. A change in the impeller material may be very beneficial, as described below.

Resistance of Materials to Cavitation Damage Table 8 shows the relative resistance of several metals to cavitation pitting produced by magnetostriction vibration. It will be seen that cast iron, the most commonly used material for impellers, has relatively little pitting resistance relative to bronze and stainless steel, which are readily cast and finished.

Damage due to cavitation erosion is commonly assessed in terms of the depth of penetration. The life of an impeller is generally considered to be the time required for cavitation erosion to reach a depth of 75% of the blade thickness at any point⁸⁶. The life of any material in years can be expressed as the product of the mean depth of penetration rate

TABLE 8 Cavitation erosion resistance of metals

Alloy	Magnetostriction Weight Loss after 2 h, mg
Rolled stellite ^a	0.6
Welded aluminum bronze	3.2
Cast aluminum bronze	5.8
Welded stainless steel (2 layers, 17 Cr-7 Ni)	6.0
Hot rolled stainless steel (26 Cr-13 Ni)	8.0
Tempered rolled stainless steel (12 Cr)	9.0
Cast stainless steel (18 Cr-8 Ni)	13.0
Cast stainless steel (12 Cr)	20.0
Cast manganese bronze	80.0
Welded mild steel	97.0
Plate steel	98.0
Cast steel	105.0
Aluminum	124.0
Brass	156.0
Cast iron	224.0

^aDespite the high resistance of this material to cavitation damage, it is not suitable for ordinary use because of its comparatively high cost and the difficulty encountered in machining and grinding.

Source: Reference 69.

(*MDPR*, in mm per year) times the thickness of the material in mm. Values of *MDPR* have been deduced³⁷ from the weight loss of test samples of known diameter in a magnetostriction test (Table 8)³⁸. Unfortunately, *MDPR*-values obtained in such tests are not the same as those of actual pumps, as the mode of cavitation varies with pump operating conditions and is generally different from the laboratory results. Nevertheless, laboratory results for *MDPR* have been used to rank the ability of various materials to resist cavitation erosion in pumps. See Section 8.1 for material selection guidelines.

Cavitation-resistant coatings, either metallic or nonmetallic, have found some niche applications. Elastomeric coatings are resilient and resist cavitation through a different erosion mechanism than that of metal. As such, they can be very effective⁶⁹. At least two considerations are involved in the use of coatings for resisting cavitation damage: 1) even if a contemplated coating demonstrates a reduced damage or erosion rate, this reduction must be enough to justify the cost of establishing a satisfactory bond between the coating and the base metal; 2) erosion of both the coating and the base material must be considered in determining the life according to the above 75%-depth criterion. Therefore, the life in this case would be equal to the *MDPR* of the coating times the coating thickness plus the *MDPR* of the base material times the allowable erosion depth of that material.

Inducers It is sometimes difficult or impossible to provide the required NPSH for an otherwise acceptable pump. Besides normal industrial situations that might produce a very low available NPSH, the need to keep the weight down in aircraft and rocket liquid-propellant pumps has led to high rotative speeds, which, for typical values of NPSH, produce extremely high suction specific speeds. The performance-NPSH required by the impeller under these circumstances can be provided by a small, axial-flow booster pump, called an inducer, placed ahead of the first-stage impeller³⁹. Inducers are designed to operate with very low NPSH and to provide enough head to satisfy the NPSH required by the impeller. In fact, long stable cavities are established on the suction sides of the long, lightly-loaded blades of an inducer, which enable it to operate at about twice the suction specific speed of a conventional impeller^{40,41}. At lower than normal flow rates, however, inducers readily produce swirling, destabilizing backflow at the inlet, which can cause excessive pump vibration in high-head pumps. These instabilities can be overcome by various passive design features, such as that described in Reference 39.

The inducers described in Reference 42 (Figure 30) have "constant lead" helical blades. They contribute not more than 5% of the total pump head. Although the efficiency of the inducer alone is low, the reduction in overall pump efficiency is not significant. Because this type of inducer causes prerotation, a careful match between inducer and suction impeller is required. In vertical multistage pumps, where a long shaft can be better supported, a vaned diffuser may be inserted between the inducer and the first-stage impeller. Such an arrangement is very beneficial for operation at reduced flow rate. Reference 42 shows that a suitable inducer-impeller combination can operate at about 50% of the NPSH required for the impeller alone at flow rates not exceeding the normal value. The NPSH requirement increases rapidly for flow rates above normal. Unless a variable-lead inducer is used^{32, 39}, operation in this range should be avoided.

Entrained Air Air or other gases may enter the impeller inlet from several sources. The immediate effect usually will be a drop in pump pressure rise, flow rate, and power. This will be followed by loss of prime if more gas is present than the impeller can handle. A typical limit for commercial industrial pumps is an inlet gas-to-liquid volume fraction (GVF) of 0.03, although specialty pumps such as those used in aircraft (Section 12.18.1) can handle higher GVF. See also Reference 10, Section 2.1.1. Air may be released from solution or enter through leaks in the suction piping. Stuffing box air leakage may be prevented by lantern rings supplied with liquid from the pump discharge. If the pump takes water from a sump with a free surface, a vortex may form from the free surface to the impeller inlet. The remedy may be the introduction of one or more baffle plates or even major changes in the sump. For information on proper sump design and the prevention of air-entraining vortices, see Sections 13.1 and 13.2, pp. 457 and 460 of Reference 7, and Reference 43. It is sometimes permissible to inject a small amount of air into the pump

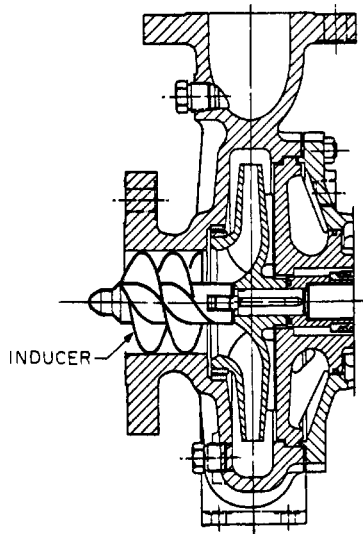


FIGURE 30 Pump fitted with inducer (Reference 42)

suction to reduce the noise and damage from cavitation caused by inadequate NPSH or recirculation in the impeller.

STARTING CENTRIFUGAL PUMPS

Priming Centrifugal pumps usually are completely filled with the liquid to be pumped *before starting*. When so filled with liquid, the pump is said to be *primed*. Pumps have been developed to start with air in the casing and then be primed.⁴⁴ This procedure is unusual with low-specific-speed pumps but is sometimes done with propeller pumps¹². In many installations, the pump is at a lower elevation than the supply and remains primed at all times. This is customary for pumps of high specific speed and all pumps requiring a positive suction head to avoid cavitation.

Pumps operated with a suction lift may be primed in any of several ways. A relatively inexpensive method is to install a special type of check valve, called a *foot valve*, on the inlet end of the suction pipe and prime the pump by filling the system with liquid from any available source. Foot valves cause undesirable frictional loss and may leak enough to require priming before each starting of the pump. A better method is to close a valve in the discharge line and prime by evacuating air from the highest point of the pump casing. Many types of vacuum pumps are available for this service. A priming chamber is a tank that holds enough liquid to keep the pump submerged until pumping action can be initiated. Self-priming pumps usually incorporate some form of priming chamber in the pump casing. Section 2.2.2 and Reference 7 may be consulted for further details.

Torque Characteristics of Drivers Centrifugal pumps of all specific speeds usually have such low starting torques (turning moments) that an analysis of the starting phase of operation seldom is required. Steam and gas turbines have high starting torques, so no special starting procedures are necessary when they are used to drive pumps. If a pump is directly connected to an internal combustion engine, the starting motor of the engine should be made adequate to start both driver and pump. If the starter does not have enough torque to handle both units, a clutch must be provided to uncouple the pump until the driver is started.

Electric motors are the most commonly used drivers for centrifugal pumps. Direct current motors and alternating current induction motors usually have ample starting torque for all pump installations, provided the power supply is adequate. Many types of reduced voltage starters are available⁷ to limit the inrush current to safe values for a given power supply. Synchronous motors are often used with large pumps because of their favorable power-factor properties. They are started as induction motors and run as such up to about 95% of synchronous speed. At this point, dc field excitation is applied and the maximum torque the motor can then develop is called the *pull-in torque*, which must be enough to accelerate the motor and connected inertia load to synchronous speed in about 0.2 s if synchronous operation is to be achieved. Centrifugal pumps usually require maximum torque at the normal operating point, and this should be considered in selecting a driver, particularly a synchronous motor, to be sure that the available pull-in torque will bring the unit to synchronous speed.

Torque Requirements of Pumps The *torque*, or turning moment, for a pump may be estimated from the power curve in USCS units by

$$M = \frac{5252P}{n} \quad (25)$$

and in SI units by

$$M = \frac{9549P}{n}$$

where M = pump torque, lb · ft (N · m)

P = power, hp (kW)

n = speed, rpm

Equation 25 makes no allowance for accelerating the rotating elements or the liquid in the pump. If a 10% allowance for accelerating torque is included, the constant should be correspondingly increased. The time Δt required to change the pump speed by an amount $\Delta n = n_2 - n_1$ is given by

$$\Delta t = \frac{I \Delta n}{k(M_m - M)} \quad (26)$$

where Δt = time, s

I = moment of inertia (flywheel effect) of all rotating elements of driver, pump, and liquid, lb · ft (kg · m²)

Δn = change in speed, rpm

k = 307 in USCS (9.549 in SI)

M_m = driver torque, lb · ft (N · m)

M = pump torque, lb · ft (N · m) (Eq. 25)

The inertia I of the driver and pump usually can be obtained from the manufacturers of the equipment. The largest permissible Δn for accurate calculation will depend on how rapidly M_m and M vary with speed. The quantity $M_m - M$ should be nearly constant over the interval Δn if an accurate estimate of Δt is to be obtained. Torque-speed characteristics of electric motors may be obtained from the manufacturers.

Horizontal-shaft pumps fitted with plain bearings and packed glands require a *break-away torque* of about 15% of M_n , the torque at the normal operating point, to overcome the static friction. This may be reduced to about 10% of M_n if the pump is fitted with antifriction bearings. The breakaway torque may be assumed to decrease linearly with speed to nearly zero when the speed reaches 15 to 20% of normal. Construction of torque-speed curves requires a knowledge of the pump characteristics at normal speed as well as details of the entire pumping system. Some typical examples taken from Reference 7 are given

below. The following forms of Eqs 12 are useful in constructing the various performance curves:

$$Q_2 = Q_1 \frac{n_2}{n_1} \quad (27)$$

$$H_2 = H_1 \left(\frac{n_2}{n_1} \right)^2 = H_1 \left(\frac{Q_2}{Q_1} \right)^2 \quad (28)$$

$$P_2 = P_1 \left(\frac{n_2}{n_1} \right)^3 \quad (29)$$

$$M_2 = M_1 \left(\frac{n_2}{n_1} \right)^2 \quad (30)$$

where Q = flow rate

n = speed

H = head

P = power

M = torque

in any consistent units of measure. After speeds n_1 and n_2 are chosen, the subscripts 1 and 2 refer to corresponding points on the characteristic curves for these speeds.

Low-Specific-Speed Pumps Figure 31 shows the constant-speed characteristics of a pump having $n_s \approx 1740$ (0.64) at best efficiency. This pump usually would be started with a valve in the discharge line closed. During the starting phase, the pump operates at shut-off with $P_1 = 25.8$ hp (19.2 kW) and $n_1 = 1770$ rpm. Then, by Eq. 25, $M_1 = 76.6$ lb · ft (103.9 N · m). These values may be used in Eq. 30 to evaluate starting torques M_2 at as many speeds n_2 as desired and plotted in Figure 32 as curve BCD . Section AB of the starting torque curve is an estimate of the breakaway torque. If the discharge valve is now opened, the speed remains nearly constant but the torque increases as the flow rate and power increase. If the normal operating point is $Q = 1400$ gpm (88.3 l/s) and $P_n = 53.2$ hp (39.7 kW), the motor torque will be $M_n = 158$ lb · ft (214 N · m) by Eq. 25. The vertical line DE in Figure 32 shows the change in torque produced by opening the discharge valve.

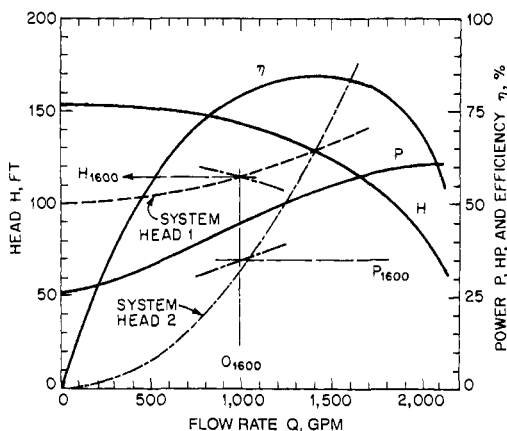


FIGURE 31 Characteristics of a 6-by-8 double-section pump at 1770 rpm ($\text{ft} \times 0.3048 = \text{m}$; $\text{gpm} \times 0.06309 = \text{l/s}$; $\text{hp} \times 0.7457 = \text{kW}$) $n_s = 1,740$ ($\Omega_s = 0.64$) (Reference 7)

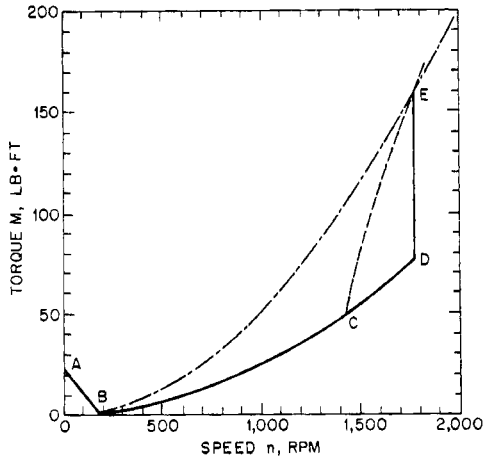


FIGURE 32 Torque characteristics of 6-by-8 pump shown in Figure 31 ($\text{lb} \cdot \text{ft} \times 1.356 = \text{N} \cdot \text{m}$) (Reference 7)

Instead of starting the pump with the discharge valve closed, let the pump be started with a check valve in the discharge line held closed by a static head of 100 ft (30.5 m). The frictional head in the system may be represented by kQ^2 . The value of k may be estimated from the geometry of the system or from a frictional-loss measurement at any convenient flow rate Q , preferably near the normal capacity Q_n . In this example, $Q_n = 1400$ gpm (88.33 l/s) and $k = 14.4/10^6$ (0.00109). The curve labeled system head 1 in Figure 31 was computed from $H = 100 + (14.4/10^6)Q^2$ ft ($H = 30.48 + 0.00109Q^2$ m) and intersects the head curve at $H = 128$ ft (39.01 m) and $Q_n = 1400$ gpm (88.33 l/s). The normal shutoff head is $H_1 = 153$ ft (46.6 m) at $n_1 = 1770$ rpm. By Eq. 28, the pump will develop a shutoff head $H_2 = 100$ ft (30.5 m) at $n_2 = 1430$ rpm. By Eq. 30, the torque at 1430 rpm will be $50 \text{ lb} \cdot \text{ft}$ ($68 \text{ N} \cdot \text{m}$), corresponding to point C in Figure 32. The portion ABC of the starting torque curve has already been constructed. Trial-and-error methods must be used to obtain the portion CE of the starting torque curve.

The auxiliary curves in Figure 33 are useful in constructing the CE portion of the starting torque curve. Select a value of n_2 intermediate between 1427 and 1770 rpm, say $n_2 = 1600$ rpm. In Figure 31, read values of Q_1 , H_1 , and P_1 for speed $n_1 = 1770$ rpm. By Eqs. 27 and 28, determine values of Q_2 and H_2 and plot as shown in Figure 31 until an intersection with the system-head 1 curve is obtained that provides Q_{1600} corresponding to $n = 1600$ rpm. By Eq. 29, determine value of P_2 and plot as shown in Figure 31 until an intersection is obtained with the Q_{1600} line which provides P_{1600} corresponding to $n = 1600$ rpm. Eq. 34 is now used to obtain M_{1600} , which is one point on the desired starting torque curve. The process is repeated for various speeds n_2 until the curve CE in Figure 32 can be drawn. The complete starting torque curve for this example is ABCE in Figure 31, with steady-state operation at point E.

Assume that the pump of the preceding examples is installed in a system having zero static head but a long pipeline with friction head given by $H = (65.4/10^6)Q^2$ ft ($H = 0.00500Q^2$ m), as shown in Figure 31 by the curve labeled system head 2. The valve in the discharge line is assumed to open instantaneously when power is first applied to the pump. The procedure described to construct curve CE of the preceding example must now be used together with the system-head 2 curve of Figure 31 to obtain the curve BE of Figure 32. The complete starting torque curve for this example is ABE in Figure 32, with steady-state operation at point E.

The inertia of the fluid in the system was neglected in solving the previous examples. Some of the power must be used to accelerate the liquid, and this may be appreciable in the case of a long pipeline. Low-specific-speed pumps, which are used with long pipelines, have rising power curves with minimum power at shutoff and maximum power at normal flow

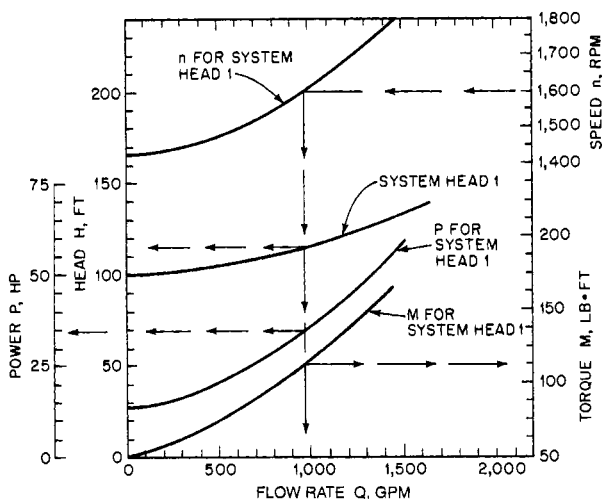


FIGURE 33 Analysis of 6-by-8 pump shown in Figure 41 ($\text{ft} \times 0.3048 = \text{m}$; $\text{gpm} \times 0.06309 = \text{l/s}$; $\text{lb} \cdot \text{ft} \times 1.356 = \text{N} \cdot \text{m}$; $\text{hp} \times 0.7457 = \text{kW}$) (Reference 7)

rate. Experience has shown that the starting torque-speed curves computed by neglecting the inertia of the liquid are conservative, so inertia effects need not be included. The inertia effect of the liquid does slow the starting operation. If the time required to reach any event, such as a particular speed or flow rate, is required, the inertia of the liquid should be considered, but including it greatly increases the difficulty of computation. References 45 through 48 give general methods for handling problems involving liquid transients.

High-specific-speed pumps have falling power flow rate curves with maximum power at shutoff and minimum power at normal flow rate. Neglecting the inertia of the liquid probably will result in too low a value for the computed starting torque for such pumps. If liquid inertia is to be included, consult References 44 to 47 and Section 11.1.

MEDIUM- AND HIGH-SPECIFIC-SPEED PUMPS Figure 34 shows constant-speed characteristics for a medium-specific-speed pump, $n_s = 4570$ (1.672) at best efficiency. The shutoff power

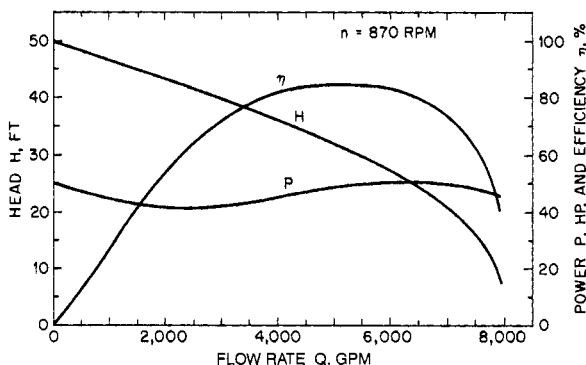


FIGURE 34 Characteristics of a 16-in (40.6cm) volute pump with mixed-flow impeller with flat power characteristic ($\text{ft} \times 0.3048 = \text{m}$; $\text{gpm} \times 0.06309 = \text{l/s}$; $\text{hp} \times 0.7457 = \text{kW}$). $n_s = 4570$ ($\Omega_s = 1.672$) (Reference 7)

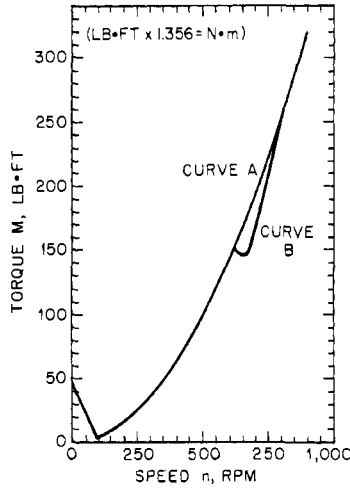


FIGURE 35 Torque characteristics of pump shown in Figure 34 (Reference 7)

is the same as the power at best efficiency, and the starting torque-speed curve is but little affected by the method of starting, as shown by Figure 35.

Figure 36 shows the constant-speed characteristics of a high-specific-speed propeller pump, $n_s \approx 12,000$ (4.391) at best efficiency. Figure 37 shows the starting torque-speed curve when the pump is started against a static head of 14 ft (4.3 m) and a friction head of 1 ft (0.3 m) at $Q_n = 12,500$ gpm (789 l/s). The system was assumed full of water with a closed check valve at the outlet end of the short discharge pipe. The methods of computation for Figures 35 and 37 were the same as for Figure 32.

Sometimes propeller pumps are started with the pump submerged but with the discharge column filled with air. In such a case, the torque-time characteristic for the driver must be known and a step-by-step calculation carried out. If the discharge column is a siphon initially filled with air, the starting torque may exceed the normal running torque during some short period of the starting operation. If the pump is driven by a synchronous

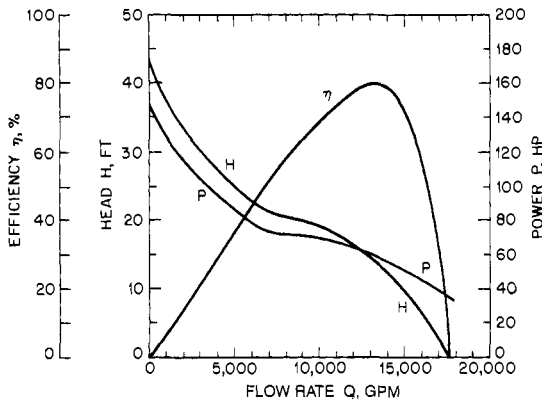


FIGURE 36 Characteristics of a 30-in (76.2-cm) discharge propeller pump at 700 rpm ($\text{ft} \times 0.1048 = \text{m}$; $\text{gpm} \times 0.06309 = \text{l/s}$; $\text{hp} \times 0.7457 = \text{kW}$). $n_s = 12,000$ ($\Omega_s = 4.391$) (Reference 7)

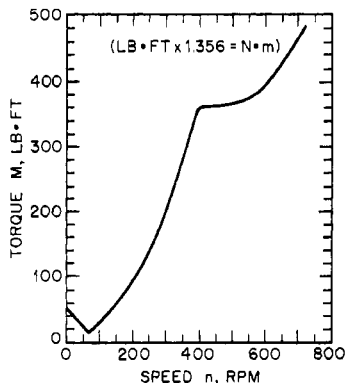


FIGURE 37 Torque characteristics of pump shown in Figure 36 (Reference 7)

motor, it is particularly important to investigate the starting torque in the range of 90 to 100% of normal speed to make sure that the pull-in torque of the motor is not exceeded. For additional information regarding starting high-specific-speed pumps discharging through long and large diameter systems, see Section 11.1.

Miscellaneous Requirements Pumps handling hot liquids should be warmed up to operating temperature before being started unless they have been especially designed for quick starting. Failure to do this may cause serious damage to wearing rings, seals, and any hydraulic balancing device that may be present. A careful check of the installation should be made before starting new pumps, pumps that have had a major overhaul, or pumps that have been standing idle for a long time. It is very important to follow the manufacturer's instructions when starting boiler-feed pumps. If these are unavailable, Reference 7 may be consulted. Ascertain that the shaft is not frozen, that the direction of rotation is correct, preferably with the coupling disengaged, and that bearing lubrication and gland cooling water meet normal requirements. Failure to do this may result in damage to the pump or driver.

A pump may run backwards at runaway speed if the discharge valve fails to close following shutdown. Any attempt to start the pump from this condition will put a prolonged overload on the motor. Figure 38 shows one example of the torque-speed transient for a pump, $n_s = 1700$ (0.622), started from a runaway reversed speed while normal pump head was maintained between the section and discharge flanges. In most practical cases, water hammer effects would make this transient even more unfavorable than Figure 38 indicates.

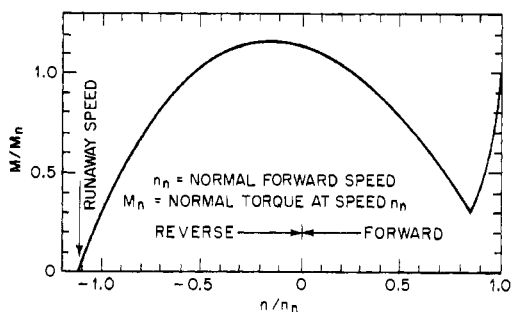


FIGURE 38 Torque characteristics of a double-suction pump, $n_s \approx 1700$ (0.64), from reversed runaway speed to normal forward speed (Reference 7)

The duration of such a transient will always be much longer than the normal starting time, and so protective devices would probably disconnect the motor from the power supply before normal operation could be achieved. Consult Section 11.1 for additional information on this subject.

REGULATION OF FLOW RATE

Flow rate variation ordinarily is accomplished by a change in pump head, speed, or both simultaneously. The flow rate and power input of pumps with specific speeds up to about 4000 (1.464) double suction increase with decreasing head, so the drivers of such pumps may be overloaded if the head falls below a safe minimum value. Increasing the head of high-specific-speed pumps decreases the flow rate but increases the power input. The drivers of these pumps should either be able to meet possible load increases or be equipped with suitable overload protection. Flow rate regulation by the various methods given below may be manual or automatic (see also References 1, 7, 12, 34 and 49).

Discharge Throttling This is the cheapest and most common method of flow rate modulation for low- and medium-specific-speed pumps. Usually its use is restricted to such pumps. Partial closure of any type of valve in the discharge line will increase the system head so the system-head curve will intersect the pump head curve at a smaller flow rate, as shown in Figure 40. Discharge throttling moves the operating point to one of lower efficiency, and power is lost at the throttle valve. This may be important in large installations, where more costly methods of modulation may be economically attractive. Throttling to the point of cutoff may cause excessive heating of the liquid in the pump. This may require a bypass to maintain the necessary minimum flow or use of different method of modulation. This is particularly important with pumps handling hot water or volatile liquids, as previously mentioned. Refer to Section 11.2 for information regarding the sizing of a pump bypass.

Suction Throttling If sufficient NPSH is available, some power can be saved by throttling in the suction line. Jet engine fuel pumps frequently are suction throttled⁶ because discharge throttling may cause overheating and vaporization of the liquid. At very low flow rate, the impellers of these pumps are only partly filled with liquid, so the power input and temperature rise are about one-third the values for impellers running full with discharge throttling. The capacity of condensate pumps frequently is submergence-controlled,⁷ which is equivalent to suction throttling. Special design reduces cavitation damage of these pumps to a negligible amount, the energy level (Section 2.1.1) being quite low.

Bypass Regulation All or part of the pump flow may be diverted from the discharge line to the pump suction or other suitable point through a bypass line. The bypass may contain one or more metering orifices and suitable control valves. Metered bypasses are commonly used with boiler-feed pumps for reduced-flow operation, mainly to prevent overheating. There is a considerable power saving if excess capacity of propeller pumps is bypassed instead of using discharge throttling. See Section 11.4.

Speed Regulation This can be used to minimize power requirements and eliminate overheating during flow rate modulation. Steam turbines and internal combustion engines are readily adaptable to speed regulation at small extra cost. A wide variety of variable-speed mechanical, magnetic, and hydraulic drives are available, as well as both ac and dc variable-speed motors. Usually variable-speed motors are so expensive that they can be justified only by an economic study of a particular case. Figure 39 shows a study by Richardson⁵⁰ of power requirements with various drivers wherein substantial economies in power may be obtained from variable-speed drives.

Regulation by Adjustable Vanes Adjustable guide vanes ahead of the impeller have been investigated and found effective with a pump of specific speed $n_s = 5700$ (2.086).

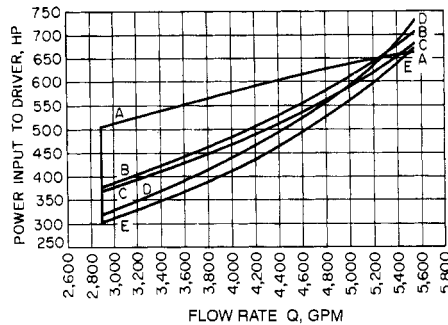


FIGURE 39 Power requirements of two double-suction pumps in series operated at constant head and variable flow rate. Total $H_n = 382$ ft (116 m) for both pumps at 1800 rpm ($\text{gpm} \times 0.06309 = \text{l/s hp} \times 0.7457 = \text{kW}$) (Reference 50). Curve AA: constant speed with discharge throttling. Curve BB: synchronous motor with variable-speed hydraulic coupling on each pump. Curve CC: variable-speed wound-rotor induction motor. Curve DD: dc motor with rectifier and shunt field control. Curve EE: synchronous motor with variable-speed constant-efficiency mechanical speed reducer

The vanes produced a positive prewhirl that reduced the head, flow rate, and efficiency. Relatively little regulation was obtained from the vanes with pumps having $n_s = 3920$ (1.204) and 1060 (0.39). Adjustable outlet diffusion vanes have been used with good success on several large European storage pumps for hydroelectric developments. Propeller pumps with adjustable-pitch blades have been investigated with good success. Wide flow rate variation was obtained at constant head and with relatively little loss in efficiency. These methods are so complicated and expensive that they have very limited application in practice. Reference 34 may be consulted for further discussion and bibliography.

Air Admission Admitting air into the pump suction has been demonstrated as a means of flow-rate regulation, with some savings in power over discharge throttling. Usually air in the pumped liquid is undesirable, and there is always the danger that too much air will cause the pump to lose its prime. The method has rarely been used in practice but might be applicable to isolated cases.

PARALLEL AND SERIES OPERATION

Two or more pumps may be arranged for parallel or series operation to meet a wide range of requirements in the most economical manner. If the pumps are close together, that is, in the same station, the analysis given below should be adequate to secure satisfactory operation. If the pumps are widely separated, as in the case of two or more pumps at widely spaced intervals along a pipeline, serious pressure transients may be generated by improper starting or stopping procedures. The analysis of such cases may be quite complicated, and References 46 to 48 should be consulted for methods of solution.

Parallel Operation Parallel operation of two or more pumps is a common method of meeting variable-flow-rate requirements. By starting only those pumps needed to meet the demand, operation near maximum efficiency can usually be obtained. The head-flow characteristics of the pumps need not be identical, but pumps with unstable characteristics may give trouble unless operation only on the steep portion of the characteristic can be assured. Care should be taken to see that no one pump, when combined with pumps of different characteristics, is forced to operate at flows less than the minimum required to prevent recirculation. See the discussion that follows on operation at other than normal flow rate. Multiple pumps in a station provide spares for emergency service and for the downtime needed for maintenance and repair.

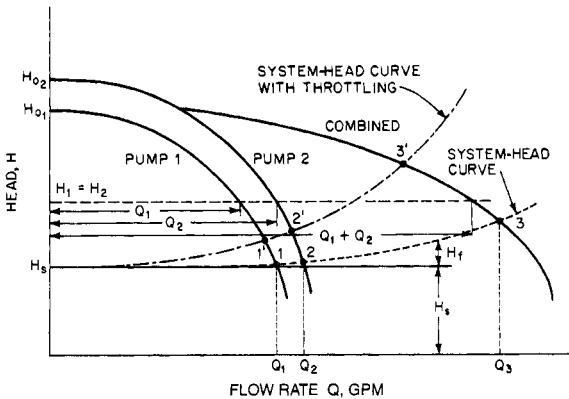


FIGURE 40 Head-flow curves of pumps operating in parallel

The possibility of driving two pumps from a single motor should always be considered, as it usually is possible to drive the smaller pumps at about 40% higher speed than a single pump of twice the capacity. The saving in cost of the higher-speed motor may largely offset the increased cost of two pumps and give additional flexibility of operation.

One of the first steps in planning for multiple-pump operation is to draw the system-head curve, as shown in Figure 40. The system head consists of the static head H_s and the sum H_f of the pipe-friction head and the head lost in the valves and fittings (see Sections 11.1 and 11.2). The head curves of the various pumps are plotted on the same diagram, and their intersections with the system-head curve show possible operating points. *Combined pump head curves* are drawn by adding the flow rates of the various combinations of pumps for as many values of the head as necessary. The intersection of any combined H - Q curve with the system-head curve is an operating point. Figure 40 shows two pump head curves and the combined curve. Points 1, 2, and 3 are possible operating conditions. Additional operating points may be obtained by changing the speed of the pumps or by increasing the system-head loss by throttling. Any number of pumps in parallel may be included on a single diagram, although separate diagrams for different combinations of pumps may be preferable.

The overall efficiency η of pumps in parallel is given by

$$\eta = \frac{H(\text{sp. gr.})}{k} \times \frac{\Sigma Q}{\Sigma P} \quad (31)$$

where H = head, ft (m)

sp. gr. = specific gravity of the liquid

k = 3960 USCS (0.1021 SI)

ΣQ = sum of the pump flow rates, gpm (l/s)

ΣP = total power supplied to all pumps, hp (W)

Series Operation Pumps are frequently operated in series to supply heads greater than those of the individual pumps. The planning procedure is similar to the case of pumps in parallel. The system-head curve and the individual head-flow curves for the pumps are plotted as shown in Figure 41. The pump heads are added as shown to obtain the combined pump head curve. In this example, Pump 2 operating alone will deliver no liquid because its shutoff head is less than the system static head.

There are two possible operating points, 1 and 2, as shown by the appropriate intersections with the system-head curve. As with parallel operation, other operating points

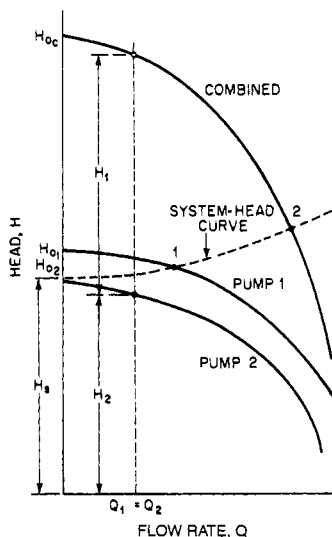


FIGURE 41 Head-flow curves of pumps operating in series

could be obtained by throttling or by changing the pump speeds. The overall efficiency of pumps in series is given by

$$\eta = \frac{Q(\text{sp. gr.})}{k} \times \frac{\sum H}{\sum P} \quad (32)$$

wherein the symbols are the same as for parallel operation. It is important to note that the stuffing box pressure of the second pump is increased by the discharge pressure of the first pump. This may require a special packing box for the second pump with leakoff to the suction of the first pump. The higher suction pressure may increase both the first cost and the maintenance costs of the second pump.

OPERATION AT OTHER THAN THE NORMAL FLOW RATE

Centrifugal pumps usually are designed to operate near the point of best efficiency, but many applications require operation over a wide range of flow rates, including shutoff, for extended periods of time. Pumps for such service are available but may require special design and construction at higher cost. Noise, vibration, and cavitation may be encountered at low flow rates. Large radial shaft forces at shutoff as well as lack of through flow to provide cooling may cause damage or breakage to such parts as shafts, bearings, seals, glands, and wearing rings of pumps not intended for such service. Some of the phenomena associated with operation at other than normal flow rate are described below.

Recirculation There is a small flow from impeller discharge to suction through the wearing rings and any hydraulic balancing device present. This takes place at all flow rates, but does not usually contribute to raising the liquid temperature very much unless operation is near shutoff.

When the flow rate has been reduced by throttling (or as a result of an increase in system head), a secondary flow called *recirculation* begins. Recirculation is a flow reversal due to separation at the suction and at the discharge tips of the impeller vanes. All impellers

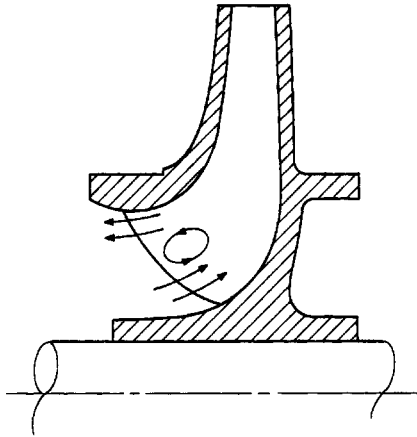


FIGURE 42 Suction recirculation

have a critical flow rate at which recirculation occurs. The flow rates at which suction and discharge recirculation begin can be controlled to some extent by design, but recirculation cannot be eliminated (see Figure 6 in Section 2.1.1).

Suction recirculation is the reversal of flow at the impeller eye. A portion of the flow is directed out of the eye at the eye diameter, as shown in Figure 42, and travels upstream with a rotational velocity approaching the peripheral velocity of the diameter. A rotating annulus of liquid is produced upstream from the impeller inlet, and through the core of this annulus passes an axial flow corresponding to the output flow rate of the pump. In pumps equipped with long, straight suction nozzles but no suction elbow, this rotating fluid has been detected over considerable distances upstream from the impeller eye. Suction pressures measured at wall taps where this phenomenon is present are always higher than the true average static pressure across the measuring section. This means that the pump head as determined from wall taps is less than it would be if true average static pressures were measured. The high shear rate between the rotating annulus and the axial flow through the core produces vortices that form and collapse, producing noise and cavitation in the suction of the pump.

Discharge recirculation is the reversal of flow at the discharge tips of the impeller blades, as shown in Figure 43. The high shear rate between the inward and outward relative velocities produces vortices that cavitate and can attack the pressure side of the blades. This phenomenon, which tends to occur at a lower flow rate than the highest Q for suction recirculation, also involves stalled flow from the diffuser vanes or volute tongue(s). Separated reversed flow recirculates and emerges from these vane systems back into the impeller with negative swirl (that is, swirl opposite to the direction of rotation). The impeller must expend significant power to redirect the portion of this fluid (that reenters it) out again—with positive swirl. As discussed in Section 2.1.1, the portion of this backflow from the diffuser or volute that enters the spaces outside the impeller shrouds and adjacent to the casing walls has the potential to reverse the axial thrust of the impeller, and this reversal can fluctuate if the backflow is unsteady (as separated, recirculating flow normally is) and not always feeding the same side of the impeller.

DETECTING RECIRCULATION The presence of suction or discharge recirculation can be determined by monitoring the pressure pulsations in the suction and in the discharge areas of the casing. Pressure transducers, installed as close to the impeller as possible in the suction and discharge areas of the pump, can be used to detect pressure pulsations. Figure 44 is a typical plot of the amplitudes of these pulsations against flow rate—with reference to the accompanying pump head and efficiency. As the flow rate is reduced, a sudden increase in the magnitude of pressure pulsations indicates the onset of recirculation.

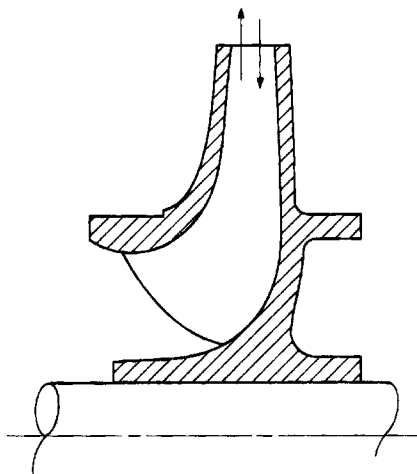


FIGURE 43 Discharge recirculation

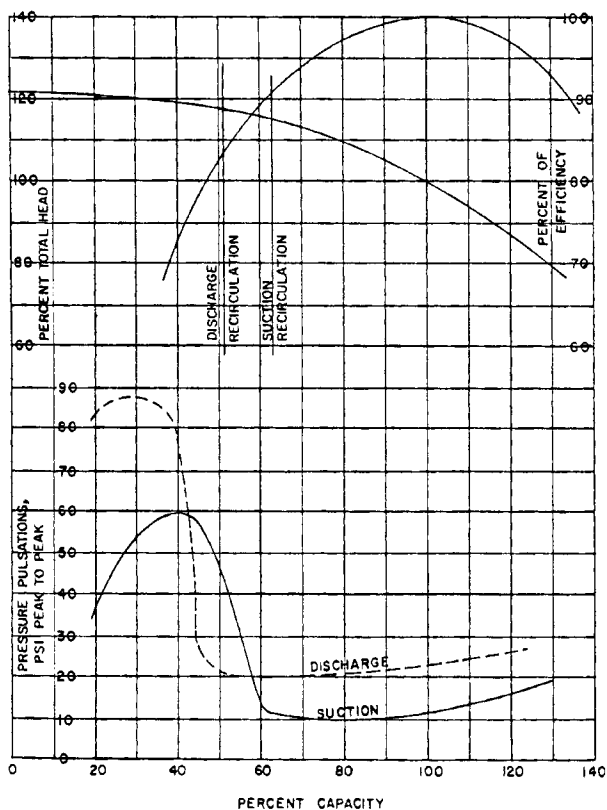


FIGURE 44 Detecting recirculation by means of pressure pulsations. As flow rate (capacity) is reduced, the initiation of recirculation is indicated by a marked increase in amplitude of the pressure pulsations.

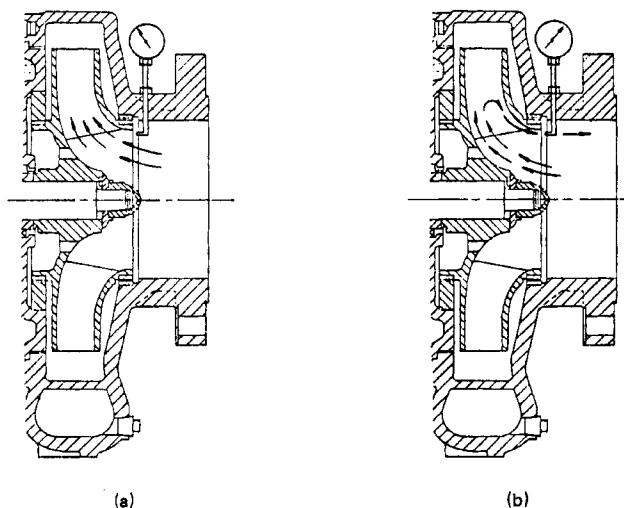


FIGURE 45 Detecting suction recirculation by means of a downstream-facing pitot tube at the impeller eye. As the flow rate is decreased, normal flow (a) gives way to recirculation (b) and the higher pressure reading due to the impact of the backflow

Also, suction recirculation can be detected by means of a pitot tube installed at the impeller eye, as shown in Figure 45. With the tube directed into the eye, the reading in the normal pumping range is the suction head minus the velocity head at the eye. At the point of recirculation, however, the flow reversal from the eye impinges on the opening of the pitot tube—with a rapid rise in the gage reading.

CORRECTIVE PROCEDURES Every impeller has specific recirculation characteristics, which usually includes the presence of cavitation bubbles and the consequent damage as they collapse on the impeller surfaces. An analysis of the symptoms associated with recirculations leads to suggested corrective procedures for avoiding damage, such as the following:

1. Increase the delivered flow rate of the pump.
2. Install a bypass between the discharge and suction of the pump. (See Section 11.4.)
3. Substitute an improved material for the impeller that is more resistant to cavitation. (See Section 8.1.)
4. Modify the impeller design. (See Section 2.1.1 and Section 12.5.)

QUANTIFYING THE ONSET OF RECIRCULATION A method for computing the flow rates Q_{SR} and Q_{DR} below which suction and discharge recirculation respectively occur is presented in Table 16 of Section 2.1.1, under the subheading “Minimum flow limits.” Another method is illustrated in Figures 38 and 39 of that section. In general, these methods reveal that Q_{SR} approaches the BEP flow rate Q_{BEP} as the eye diameter of the impeller approaches that of the discharge; that is, as D_e approaches D_2 . Thus, high-specific-speed (high- n_s) impellers and inducers are subject to suction recirculation over most of the flow range below BEP. However, even for lower- n_s impellers, as can be seen in the first design example of Section 2.1.1, D_e/D_2 increases with the design value of suction specific speed S , leading to the conclusion that high- S (or high- Ω_{ss}) impellers can experience the deleterious effects of recirculation at flow rates not far below BEP.

OPERATION WITH RECIRCULATION Despite the existence of suction and discharge recirculation, the mechanical response of the pump will not be serious unless the energy level is high.

In other words, most pumps can indeed be operated at $Q < Q_{SR}$. The minimum flow rate or simply "minimum flow" Q_{min} is quantified in Section 2.1.1. As energy level is increased, Q_{min} approaches Q_{SR} in the limit. Examples of the difference between Q_{min} and Q_{SR} are as follows: For water pumps rated at 2500 gpm (158 l/s) and 150 ft (45.7 m) total head or less, the minimum operating flows can be as low as 50% of the suction recirculation values shown for continuous operation and as low as 25% for intermittent operation. For hydrocarbons, the minimum operating flows can be as low as 60% of the suction recirculation values shown for continuous operation and as low as 25% for intermittent operation.^{51, 52, 53}

Temperature Rise Under steady-state conditions, friction and the work of compression increase the temperature of the liquid as it flows from suction to discharge. A further temperature increase may arise from liquid returned to the pump suction through wearing rings, a balancing device, or a minimum-flow bypass line that protects the pump when operating at or near shutoff.

Assuming that all heat generated remains in the liquid, the temperature rise is

$$\Delta T = \frac{gH \times (1 - \eta)}{g_o C_p \eta J} + \Delta T_c \quad (33)$$

where $g/g_o = 1$ lbf/lbm; but when using SI units, g/g_o is replaced by 9.80665 m/s^2 ($= g$ in the SI system). ΔT_c is due to the compression of the liquid and is not a consequence of loss or dissipation as is the term involving the pump efficiency η (see Section 2.1). As shown in Reference 1 of Section 2.1, ΔT_c is 3°F per 1000 psi (0.24°C per MPa) of pump pressure rise for hydrocarbon fuels. For boiler feedwater at 350°F (177°C), $\Delta T_c = 1.6^\circ\text{F}$ per 1000 psi (0.129°C per MPa), but it is much smaller for cold water. By consulting tables of properties for the liquid phase of the fluid being pumped and assuming the compression process between the actual inlet and discharge pressures to be isentropic, ΔT_c can be determined. This is important if Eq. 33 is used to evaluate overall pump efficiency from temperature rise measurements. ΔT and ΔT_c are often of the same order of magnitude at BEP, and serious errors have been made by excluding ΔT_c from the efficiency computation. At very low, off-BEP flow rates, ΔT will be high in comparison to ΔT_c ; so, the latter can be safely ignored in temperature rise calculations at such low-efficiency conditions.

In practice, determination of efficiency from ΔT -measurements is accomplished by the direct thermodynamic method⁵⁴, rather than by the ΔT_c -method. Both approaches are based on the definition of pump efficiency as the ratio of an isentropic rise of total enthalpy ($= g\Delta H$) to the actual rise of total enthalpy (Eq. 1 of Section 2.1.1), allowances being made for the usually small external power losses that do not appear in the pumped fluid (such as bearing drag) and the similarly small effects of heat transferred between pump and surroundings. In the direct thermodynamic method, the enthalpy rise Δh is found from the chain rule,

$$\Delta h = \int dh = \int [(\partial h / \partial p)_T dp + (\partial h / \partial T)_p dT] = a \Delta p + C_p J \Delta T$$

the coefficients a and $C_p J$ being average values of the two partial derivatives as found from tables of thermodynamic properties of the fluid. Values of these partial derivatives are conveniently tabulated for water in Reference 54.

General service pumps handling cold liquids may be able to stand a temperature rise as great as 100°F (56°C). Most modern boiler-feed pumps may safely withstand a temperature rise of 50°F (28°C). The NPSH required to avoid cavitation or to prevent flashing of hot liquid returned to the pump suction may be the controlling factor. Minimum flow may be dictated by other factors, such as recirculation and unbalanced radial and axial forces on the impeller. Axial forces can be the controlling factor with single-stage double-suction pumps.

It is especially important to protect even small pumps handling hot liquids from operation at shutoff. This is usually done by providing a bypass line fitted with a metering orifice to maintain the minimum required flow through the pump. In the case of boiler-feed pumps, the bypass flow usually is returned to one of the feed-water the water heaters. Unless especially designed for cold starting, pumps handling hot liquids should be warmed up gradually before being put into operation.

Radial Thrust Ideally, the circumferential pressure distribution at the impeller exit is uniform at the design condition (as explained in Section 2.1.1); however, it becomes nonuniform at off-BEP flow rates. An exception is that concentric collecting configurations will produce nonuniform pressure distributions at the BEP. Any nonuniformity leads to a radial force on the pump shaft called the radial thrust or radial reaction. The radial thrust F_r , in pounds (newtons) is

$$F_r = kK_r(\text{sp. gr.})HD_2b_2 \quad (34)$$

where $k = 0.433$ USGS (9790 SI)

K_r = experimentally determined coefficient

sp. gr. = specific gravity of the liquid pumped (equal to unity for cold water)

H = pump head, ft (m)

D_2 = outside diameter of impeller, in (m)

b_2 = breadth of impeller at discharge, including shrouds, in (m)

Values of K_r determined by Agostinelli et al.⁵⁵ for single-volute pumps are given in Figure 46 as functions of specific speed and flow rate. The magnitude and direction of F_r on the pump shaft may be estimated from Figure 47, but Eq. 34 probably will be more accurate for determining the magnitude of the force. The radial thrust usually is minimum near $Q = Q_n$, the flow rate at best efficiency, but rarely goes completely to zero. Near shutoff, F_r usually is maximum and may be a considerable force on the shaft in high-head pumps.

The radial thrust can be made much smaller throughout the entire flow-rate range by using a double volute (twin volute) or a concentric casing. These designs should be considered, particularly if the pump must operate at small flow rates. Figures 48 to 50 compare radial forces generated by three types of casings: a standard volute, a double volute, and a modified concentric casing. The latter casing was concentric with the impeller for 270° from the tongue and then enlarged in the manner of a single volute to form the discharge nozzle. The magnitude and direction of F_r on the pump shaft for the modified concentric casing may be estimated from Figure 51. The direction of F_r on the pump shaft with a double volute was somewhat random but in the general vicinity of the casing tongue. Radial forces on pumps fitted with diffuser vanes usually are rather small, although they may be significant near shutoff due to stall in some of the passages and not in others.

EXAMPLE Consider a single-stage centrifugal pump, $n_s = 2000$ (0.732) at best efficiency, handling cold water, sp. gr. = 1.0. Estimate the radial thrust on the impeller at half the normal flow rate when fitted with (a) a single volute, (b) a modified concentric casing, and (c) a double volute. Impeller dimensions are $D_2 = 15.125$ in (38.4 cm) and $b_2 = 2.5$ in (6.35 cm). The shutoff head is $H = 252$ ft (76.8 m), and the head at half capacity is $H = 244$ ft (74.4 m).

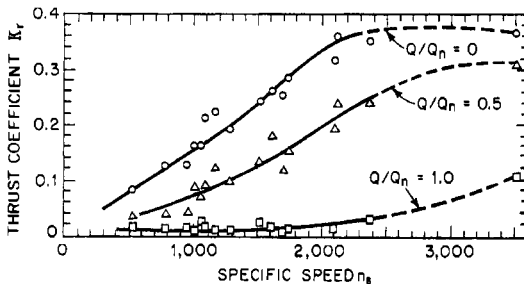


FIGURE 46 K_r as a function of specific speed and flow rate for single-volute pumps (to obtain Ω_n , divide by 2733.) (Reference 55)

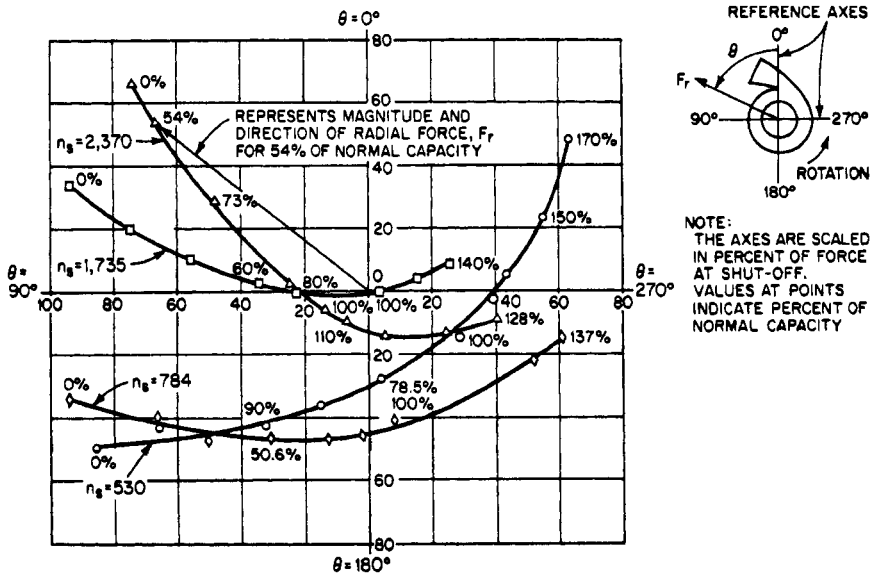


FIGURE 47 Polar plot showing direction of resultant radial forces for single-volute pumps at various flow rates ("capacities") and specific speeds. To obtain Ω_s , divide by 2733. (Reference 55)

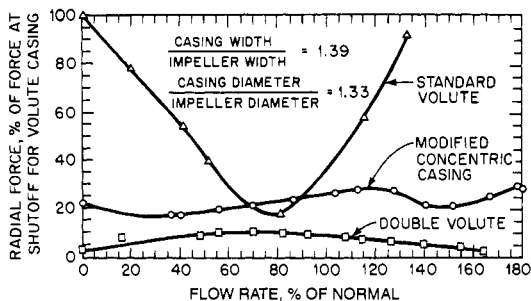


FIGURE 48 Comparison of the effect of three casing designs on radial forces for $n_s = 1165$ (0.426) (Reference 55)

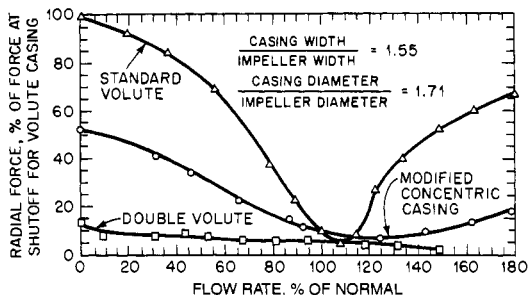


FIGURE 49 Comparison of the effect of three casing designs on radial forces for $n_s = 2120$ (0.776) (Reference 55)

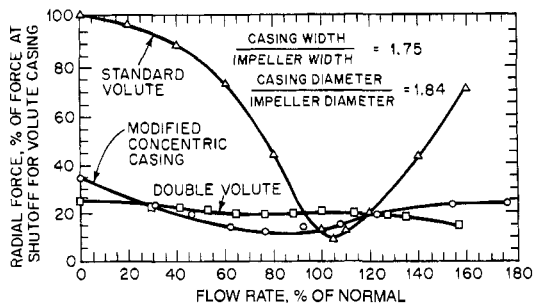


FIGURE 50 Comparison of effect of three casing designs on radial forces for $n_s = 3500$ (1.281) (Reference 55)

Solution

(a) $K_r = 0.2$ from Figure 46, by Eq. 34.

$$F_r = (0.433)(0.2)(1.0)(244)(15.125)(2.5) = 799 \text{ lb (3554 N)}$$

Estimating between the curves for $n_s = 2370$ (0.867) and 1735 (0.635) in Figure 47, the direction of F_r on the shaft should be about 65° to 70° from the casing tongue in the direction of rotation.

(b) Use Figure 49, $n_s = 2120$ (0.776), which is nearest to $n_s = 2000$ (0.730), to find the radial force for a modified concentric casing at half flow, which is about 33%

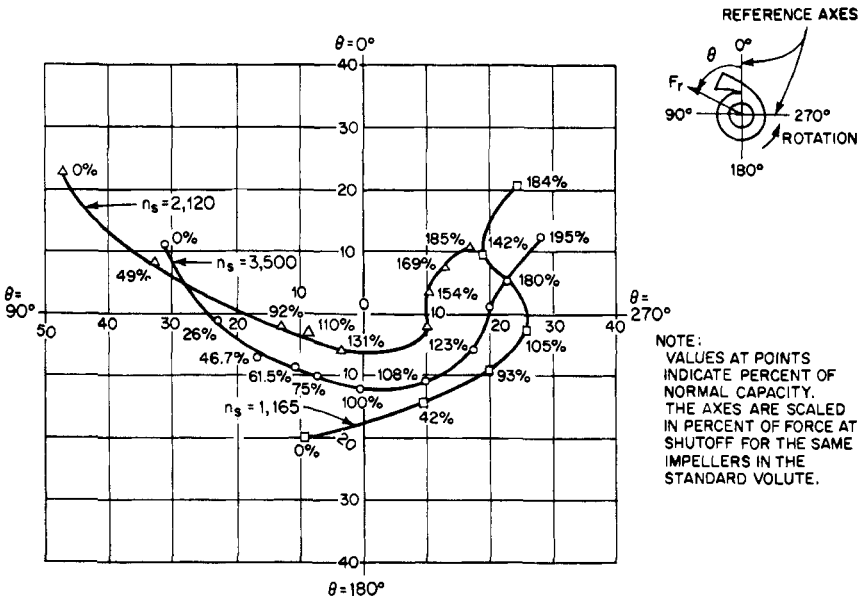


FIGURE 51 Polar plot showing direction of resultant radial forces for modified concentric casings at various flow rates and specific speeds; namely, 1,165 (0.426), 2,120 (0.776), and 3500 (1.281). The casings were concentric for 270° from the tongue. (Reference 55)

of shutoff value for a single-volute casing. From Figure 46, for a single-volute casing at shutoff, $K_r = 0.34$ and

$$F_r = (0.433)(0.34)(1.0)(252)(15.125)(2.5) = 1403 \text{ lb (6241 N)}$$

Then, for a modified concentric casing at half flow,

$$F_r = (0.33)(1403) = 463 \text{ lb (2059 N)}$$

From Figure 51, the direction of F_r should be about 75° to 80° from the casing tongue in the direction of rotation.

- (c) From Figure 49, the radial force for a double-volute casing is about 8% of the shutoff value for the single-volute casing, and so

$$F_r = (0.08)(1403) = 112 \text{ lb (499 N)}$$

According to Agostinelli et al.,⁵⁶ the direction of the radial thrust in double-volute casings was found to be generally toward the casing tongue. Stepanoff¹² has found this direction to follow approximately that in single-volute casings (see also Biheller⁵⁶).

Axial Thrust See Section 2.1.1 and 2.2.1.

ABNORMAL OPERATION

Complete Pump Characteristics Many types of abnormal operation involve reversed pump rotation, reversed flow direction, or both, and special tests are required to cover these modes of operation. Several methods of organizing the data have been proposed, and each has certain advantages. The Thoma diagrams shown in Figures 52 and 53⁴⁶ are easily understood and are truly *complete characteristics diagrams* because all possible modes of operation are covered (see Sections 11.3 and 12.14 and also Reference 57).

Figure 54 shows schematic cross-sections of the two pumps tested by Swanson⁵⁸ for which characteristics are given in Figure 53.

The *Karman circle diagram* (Reference 59) attempted to show the complete characteristics as a four-quadrant contour plot of surfaces representing head and torque with speed and flow rate as base coordinates. Because the head and torque tend to infinity in two zones of operation, another diagram would be required to show the complete pump characteristics. The data presented in such a diagram are, nevertheless, adequate for almost all requirements. One example of a circle diagram is given in Figures 55 and 56. Other examples may be found in References 12, 58, and 59.

Frequently, tests with negative head and torque have been omitted so that only half of the usual circle diagram could be shown. This has been called a *three-quadrant plot*, but the information necessary to predict an event, such as possible water-column separation following a power failure, is lacking.

Power Failure Transient A sudden power failure that leaves a pump and driver running free may cause serious damage to the system. Except for rare cases where a flywheel is provided, the pump and driver usually have a rather small moment of inertia, and so the pump will slow down rapidly. Unless the pipeline is very short, the inertia of the liquid will maintain a strong forward flow while the decelerating pump acts as a throttle valve. The pressure in the discharge line falls rapidly and, under some circumstances, may go below atmospheric pressure, both at the pump discharge and at any points of high elevation along the pipeline. The minimum pressure head which occurs during this phase of the motion is called the *downsurge*, and it may be low enough to cause vaporization followed by complete separation of the liquid column. Pipelines have collapsed under the external atmospheric pressure during separation. When the liquid columns rejoin, following separation, the shock pressures may be sufficient to rupture the pipe or the pump

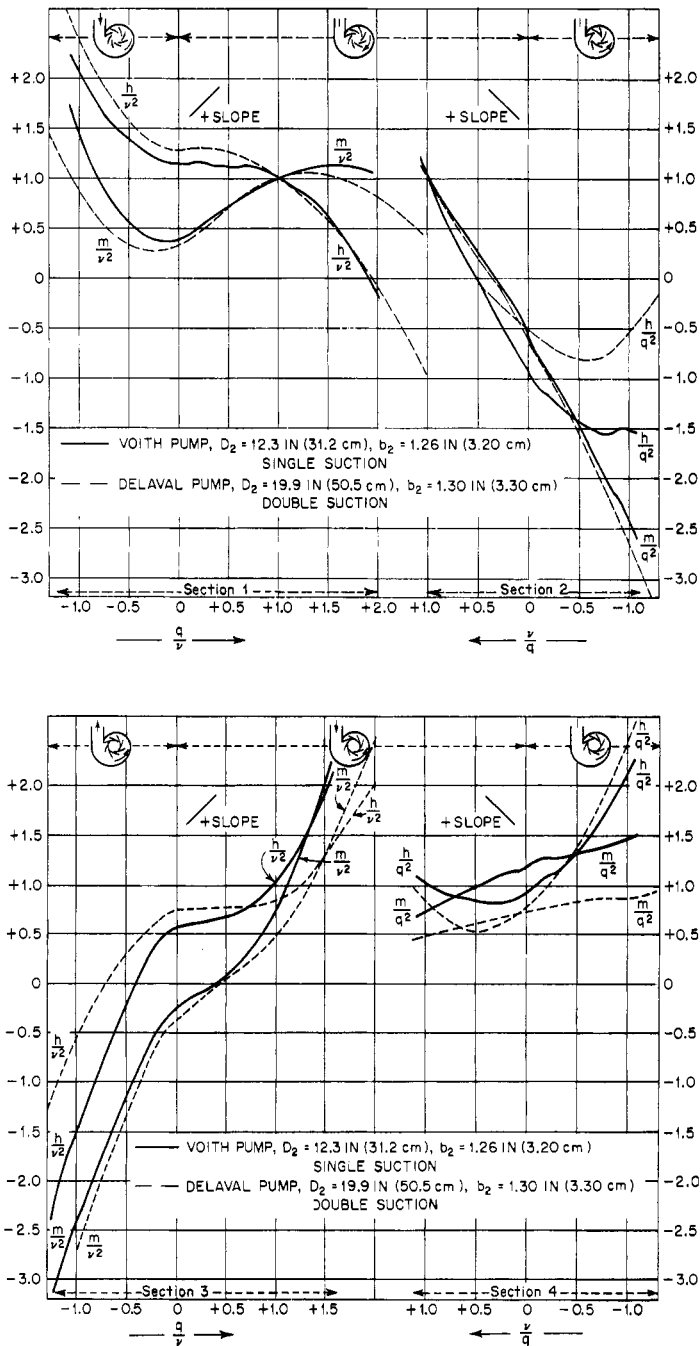


FIGURE 52 Complete pump characteristics. Specific speeds: Voith pump $n_s = 1,935$ (0.708); Delaval pump $n_s = 1,500$ (0.549). (Reference 45)

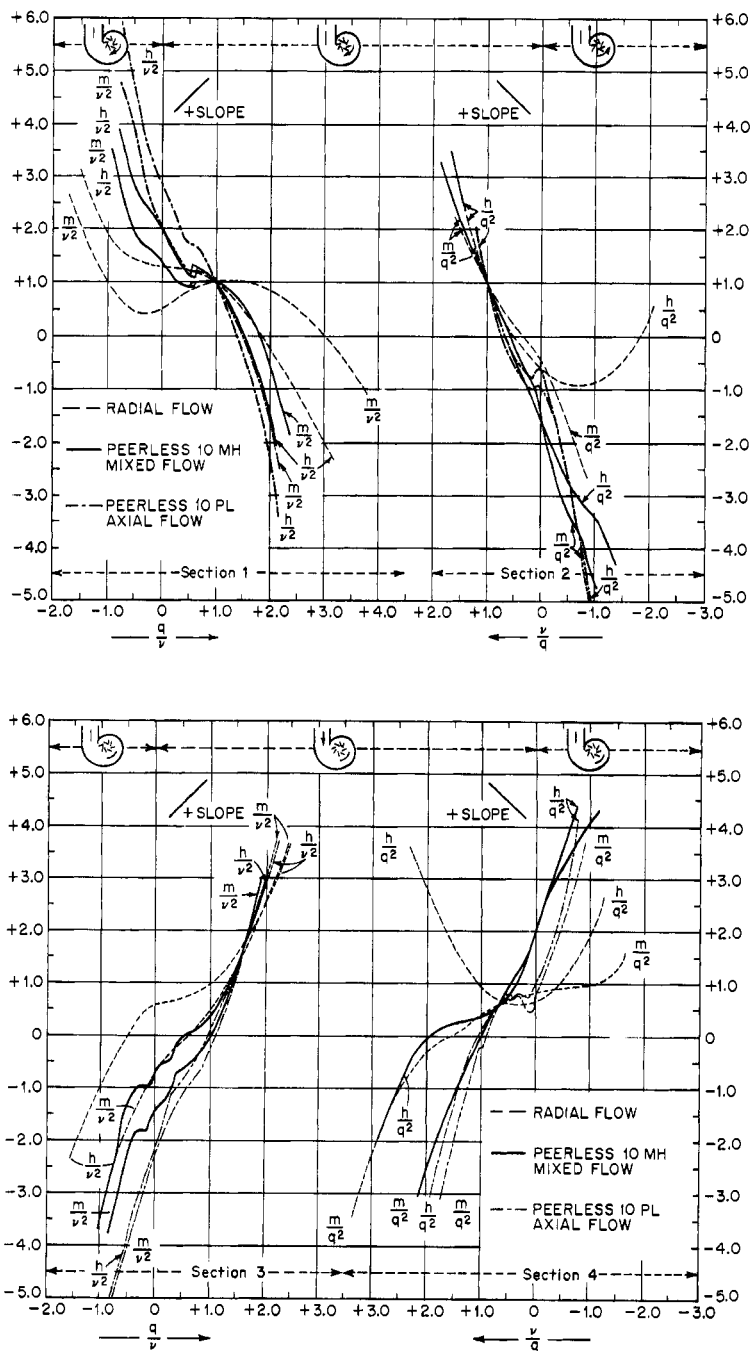


FIGURE 53 Complete pump characteristics. Specific speeds: Radial flow - $n_s = 1800$ (0.659); Peerless 10MH mixed flow - $n_s = 7,550$ (2.763); Peerless 10PL axial flow - $n_s = 13,500$ (4.940). (Reference 45)

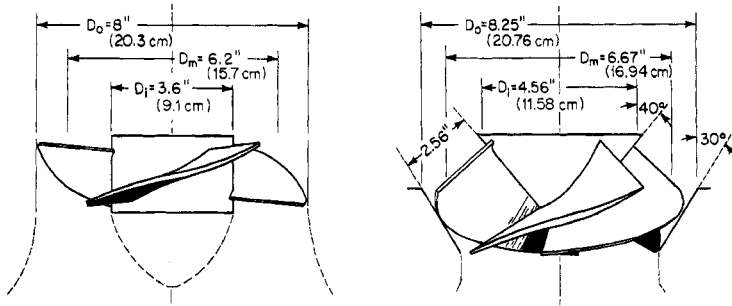


FIGURE 54 Schematic cross sections of high-specific-speed pumps, $n_s = 13,500$ (4.940) for axial flow pump; $n_s = 7,550$ (2.763) (Reference 45)

casing. Closing a valve in the discharge line will only worsen the situation, and so valves having programmed operation should be closed very little, if at all, before reverse flow begins.

Reversed flow may be controlled by valves or by arranging to have the discharge pipe empty while air is admitted at or near the outlet. If reversed flow is not checked, it will bring the pump to rest and then accelerate it with reversed rotation. Eventually the pump will run as a turbine at the runaway speed corresponding to the available static head diminished by the frictional losses in the system. However, while reversed flow is being established, the reversed speed may reach a value considerably in excess of the steady-state runaway speed. Maximum reversed speed appears to increase with increasing efficiency and increasing specific speed of the pump. Calculations indicate maximum reversed speeds more than 150% of normal speed for $n_s = 1935$ (0.708) and $\eta = 84.1\%$ ⁴⁵. This should be considered in selecting a driver, particularly if it is a large electric motor.

There will be a pressure increase, called the *upsurge*, in the discharge pipe during reversed flow. The maximum upsurge usually occurs a short time before maximum reversed speed is reached and may cause a pressure as much as 60% or more above normal at the pump discharge. A further discussion of power-failure transients is given in Section 11.3.

ANALYSIS OF TRANSIENT OPERATION The data of Figures 52 and 53 have been presented in a form suitable for general application to pumps having approximately the same specific speeds as those tested. The symbols are $h = H/H_n$, $q = Q/Q_n$, $m = M/M_n$, and $\nu = n/n_n$, wherein H , Q , M , and n represent instantaneous values of head, flow rate, torque, and speed respectively and the subscript n refers to the values at best efficiency for normal constant-speed pump operation. Any consistent system of units may be used. According to the affinity laws (Eqs. 12), q is proportional to ν , and h and m are proportional to ν^2 . Thus the affinity laws are incorporated in the scales of the diagrams. Figures 52 and 53 are divided into sections for convenience in reading data from the curves. The curves of sections 1 and 3 extend to infinity as q/ν increases without limit in either the positive or negative direction. This difficulty is eliminated by sections 2 and 4, where the curves are plotted against ν/q , which is zero when q/ν becomes infinite.

Usually any case of transient operation would begin at or near the point $q/\nu = \nu/q = 1$, which appears in both sections 1 and 2 of Figures 52 and 53. The detailed analysis of transient behavior is beyond the scope of this treatise. An analytical solution by the rigid column method, in which the liquid is assumed to be a rigid body, is given in Reference 45. Friction is easily included, and the results are satisfactory for many cases. The same reference includes a semigraphical solution to allow for elastic waves in the liquid, but friction must be neglected. Graphical solutions including both elastic waves and friction are discussed in References 45 to 48. Computer solutions of a variety of transient problems are discussed in Reference 47. These offer considerable flexibility in the analysis once the necessary programs have been prepared.

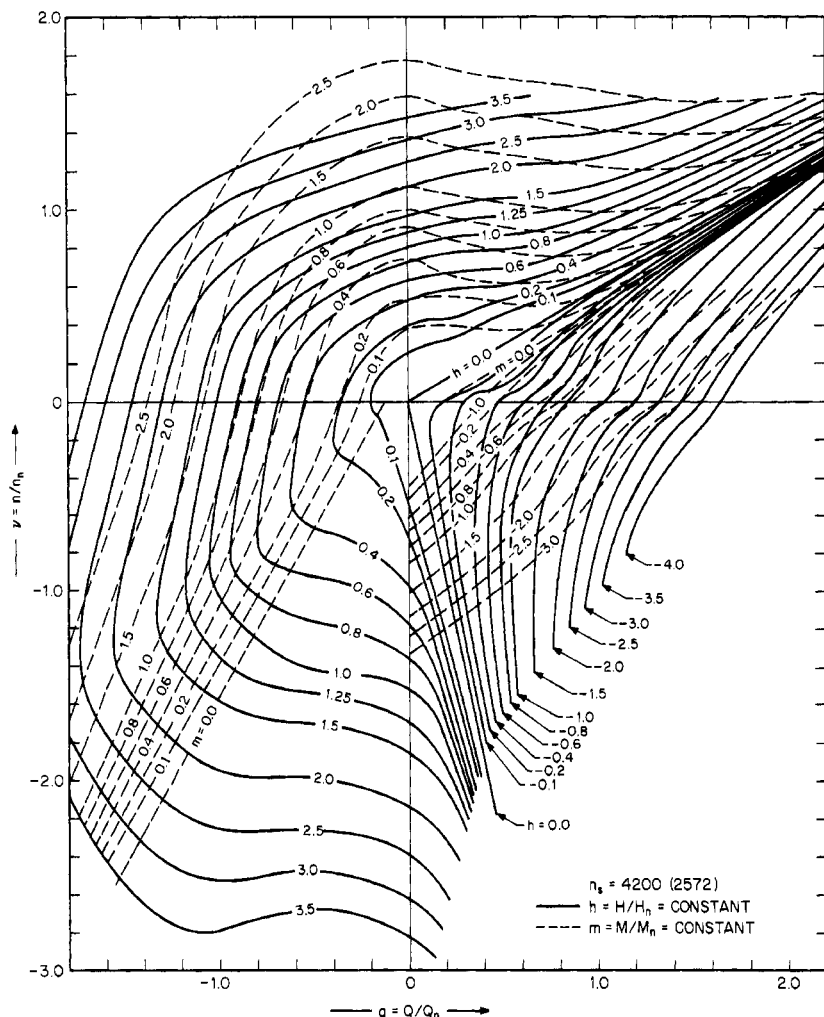


FIGURE 55 Circle diagram of pump characteristics. Specific speed $n_s = 4200$ (1.537). (Courtesy Combustion Engineering)

Some extreme conditions of abnormal operation can be estimated at points where the curves of Figures 52 and 53 cross the zero axes and are listed in Table 9. The data for Columns 2 to 4 of Table 9 were read from Sections 2 of Figures 52 and 53, and the data for Columns 6 and 7 were read from Sections 3. Column 8 was computed from Column 7 by assuming $h = 1$. Let the pump having $n_s = 1500$ (0.549) deliver cold water with normal head $H_n = 100$ ft (30.48 m), and let the center of the discharge flange be 4 ft (1.2 m) above the free surface in the supply sump. The discharge pressure head following power failure may be estimated by assuming the inertia of the rotating elements to be negligible relative to the inertia of the liquid in the pipeline. Then $q = 1$ and, from Column 2 of Table 9, the downsurge pressure head is $(-0.22)(100) - 4 = -26$ ft (-7.9 m), which is not low enough to cause separation of the water column.

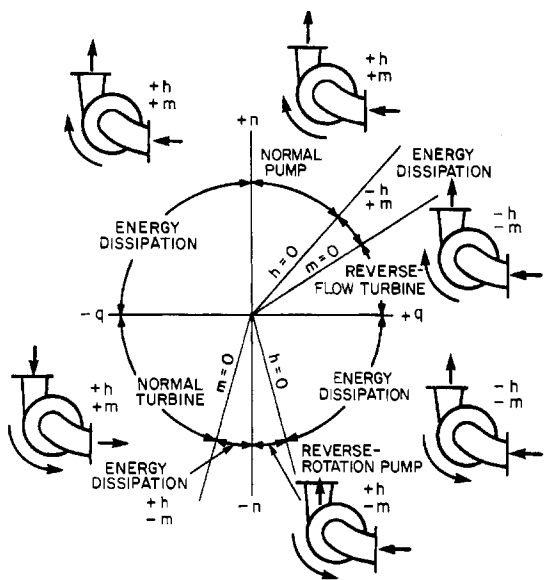


FIGURE 56 Explanatory diagram for Figure 55

Actually the downsurge would be less than this because of the effects of inertia and friction, which have been neglected. If this pump were stopped suddenly by a shaft seizure or by an obstruction fouling the impeller, Column 4 of Table 9 shows the downsurge to be $(-0.55)(100) - 4 = -59$ ft (-18 m), which would cause water column separation and, probably, subsequent water hammer. If, following power failure, the pump were allowed to operate as a no-load turbine under the full normal pump head, Column 8 of Table 9 shows the runaway speed would be 1.14 times the normal pump speed. The steady-state runaway speed usually would be less than this because the effective head would be decreased by friction, but higher speeds would be reached during the transient preceding steady-state operation. Column 8 shows that runaway speeds increase with increasing specific speed.

TABLE 9 Abnormal operating conditions of several pumps

Specific speed n_s (Ω_s)	Downsurge				Runaway turbine		
	Free-running $m/q^2 = 0$		Locked-rotor $v/q = 0$		$m/v^2 = 0$		$h = 1$
	h/q^2	v/q	h/q^2	m/q^2	q/v	h/v^2	v
1	2	3	4	5	6	7	8
1500 (0.549) ^a	-0.22	0.36	-0.55	-0.53	0.46	0.77	1.14
1800 (0.659)	-0.24	0.31	-0.60	-0.44	0.56	0.75	1.16
1935 (0.708)	-0.36	0.32	-0.94	-0.60	0.41	0.66	1.23
7550 (2.763)	-0.23	0.56	-1.57	-1.38	0.99	0.38	1.62
13,500 (4.940)	-0.12	0.67	-0.96	-0.60	1.08	0.33	1.73

^aDouble-suction.

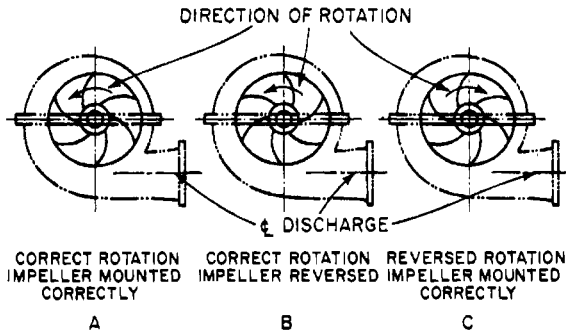


FIGURE 57A through C Pump assembly and rotation (Reference 7)

Incorrect Rotation Correct rotation of the driver should be verified before it is coupled to the pump (Figure 57). Sections 3 of Figures 52 and 53 show that reversed rotation might produce some positive head and flow rate with pumps of low specific speed, but at very low efficiency. It is unlikely that positive head would be produced by reversed rotation of a high-specific-speed pump.

Reversed Impeller Some double-suction impellers can be mounted reversed on the shaft. If the impeller is accidentally reversed, as at B in Figure 57, the flow rate and efficiency probably will be much reduced and the power consumption increased. Care should be taken to prevent this, as the error might go undetected in some cases until the driver was damaged by overload. Table 1 shows performance data for six pumps with reversed impellers. At least one of these would overload the driver excessively.

Further discussion of abnormal operating conditions may be found in References 7, 12, and 34.

Vibration Vibration caused by flow through wearing rings and by cavitation has been discussed in the foregoing and some remedies indicated. Vibration due to unbalance is not usually serious in horizontal units but may be of major importance in long vertical units, where the discharge column is supported at only one or two points. The structural vibrations may be quite complicated and involve both natural frequencies and higher harmonics. Vibration problems in vertical units should be anticipated during the design stage. If vibration is encountered in existing units, the following steps may help to reduce it: (1) dynamically balance all rotating elements of both pump and motor; (2) increase the rigidity of the main support and of the connection between the motor and the discharge column; (3) change the stiffness of the discharge column to raise or lower natural frequencies as required. A portable vibration analyzer may be helpful in this undertaking. Kovats⁶⁰ has discussed the analysis of this problem in some detail.

Structural vibrations can occur in most pump types. Typical sources are a) bearing housings—due to the commonly encountered cantilever construction, b) couplings, c) rotor instabilities stemming from excessive ring clearances and consequent loss of Lomakin stiffening of long-shaft multistage pumps, and d) hydraulic unbalance—due to dimensional variations in flow passages and clearances⁶¹.

PREDICTION OF EFFICIENCY FROM MODEL TESTS

Many pumps used in pumped storage power plants and water supply projects are so large and expensive that extensive use is made of small models to determine the best design. It is often necessary to estimate the efficiency of a prototype pump, as a part of the guarantee,

from the performance of a geometrically similar model. A model and prototype are said to operate under dynamically similar conditions when

$$Dn/\sqrt{H} = D'n'/\sqrt{H'},$$

where D = impeller diameter

n = pump speed

H = pump head

in any consistent units of measure. Primed quantities refer to the model and unprimed quantities to the prototype. Dynamic similarity is a prerequisite to model-prototype testing so that losses that are proportional to the squares of fluid velocities, called *kinetic losses*, will scale directly with size and not change the efficiency. Surface frictional losses are boundary-layer phenomena which depend on Reynolds number

$$Re = D\sqrt{H}/\nu,$$

where ν = kinematic viscosity of the liquid pumped.

(See also Section 2.1, Eqs. 36–40.) Reynolds numbers increase with increasing size, and, within limits, surface frictional-loss coefficients decrease with increasing Reynolds number. This leads to a gain in efficiency with increasing size. Computational difficulties have forced an empirical approach to the problem. Details of the development of a number of formulas are given in Reference 62.

Moody-Stauffer Formula In 1925, L. F. Moody and F. Stauffer independently developed a formula that was later modified by Pantell to the form

$$\left(\frac{1 - \eta_h}{1 - \eta'_h} \right) \left(\frac{\eta'_h}{\eta_h} \right) = \left(\frac{D'}{D} \right)^n \quad (35)$$

where η_h = hydraulic efficiency, discussed previously, and D = impeller diameter. Primed quantities refer to the model, and n is a constant to be determined by tests. The model must be tested with the same liquid that will be used in the prototype; that is, cold water in most practical cases. The original formula contained a correction for head, which is negligible if $H' \geq 0.8H$, and this requirement is now virtually mandatory in commercial practice. The meager information available indicates $0.2 \geq n \geq 0.1$ approximately, with the higher value currently favored. Improvements in construction and testing techniques very likely will move n toward lower value in the future. The Moody-Stauffer formula has been widely used since first publication. In practice, both η'_h and η_h are usually replaced by the overall efficiencies η' and η , respectively, because of the difficulty in determining proper values for the mechanical and volumetric efficiencies (see Eq. 11).

Rütschi Formulas The general form of several empirical formulas due to K. Rütschi⁶² and others was originally given as

$$\eta_h = \frac{f}{f'} \eta'_h \quad (36)$$

where η_h and η'_h are the hydraulic efficiencies of the prototype and model, respectively, and f and f' are values of an empirical f function for both the prototype and the model. The f function was obtained from tests of six single-stage pumps. $n_s < 2000$ (0.732) and is shown in Figure 58 based on the *eye diameters* D_o of the pumps in *millimeters*. Thus the f function depends on actual size in addition to scale ratio. The extrapolated portion of the curve, shown dashed in Figure 58, checked well with values for a model and large prototype, shown by E' and E , respectively. One of several formulas that have been proposed to fit the curve in Figure 58 is, in SI units,

$$f = 1 - \frac{3.15}{D_o^{1.6}} \quad (37)$$

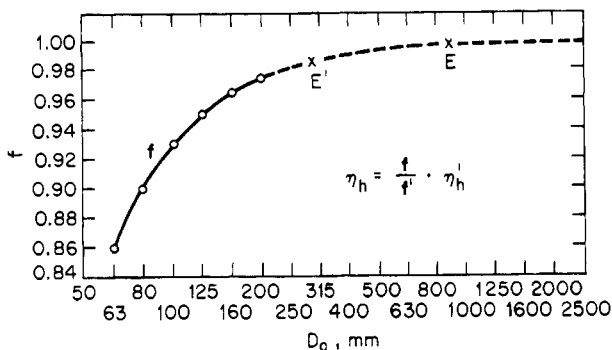


FIGURE 58 The f function for the Rüttschi formula (to obtain D_o in inches, multiply by 0.03937) (References 13 and 62)

where the eye diameter D_o is in centimeters or, in USCS units,

$$f = 1 - \frac{0.0133}{D_o^{1.6}} \quad (38)$$

where D_o is in feet. Figure 59 gives a graphical solution of Eq. 38. The Society of German Engineers (VDI) has adopted a slightly modified version of the Rüttschi formula as standard.

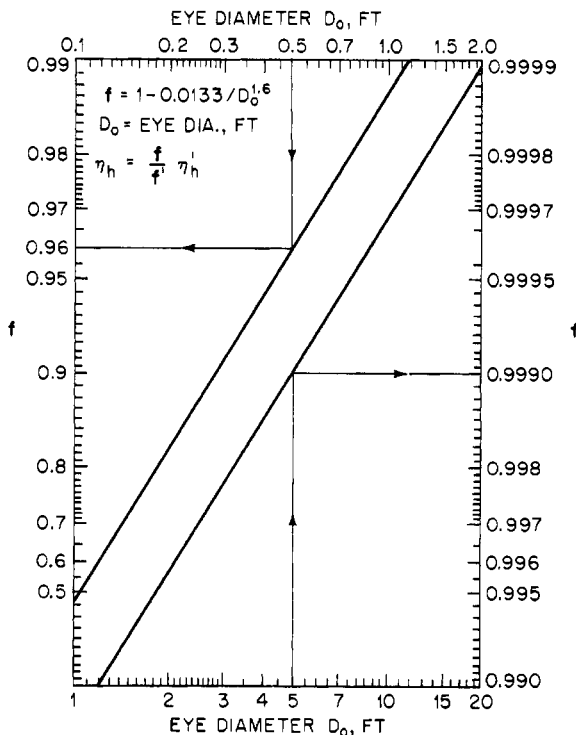


FIGURE 59 Chart for the solution of the Rüttschi formula (to obtain D_o in meters, multiply by 0.3048) (Reference 62)

Rütschi later recommended, in discussion of Reference 62, that the internal efficiency $\eta_i = \eta/\eta_m$ be used instead of the hydraulic efficiency η_h in Eq. 45. The mechanical efficiency η_m will probably be very high for both model and prototype for most cases of interest, so good results should be obtained if the overall efficiency is used in Eq. 36.

Model-prototype geometric similarity should include surface finish and wearing ring or tip clearances, but this may be difficult or impossible to achieve. Anderson (see Section 2.1.1: Figure 10 and Reference 6) proposed a method that includes a correction for dissimilarity in surface finish.

OPERATION OF PUMPS AS TURBINES

Centrifugal pumps may be used as hydraulic turbines in some cases where low first cost is paramount. Because the pump has no speed-regulating mechanism, considerable speed variation must be expected unless the head and load remain very nearly constant. Some speed control could be obtained by throttling the discharge automatically, but this would increase the cost, and the power lost in the throttle valve would lower the overall efficiency.

Pump Selection After the head, speed, and power output of the turbine have been specified, it is necessary to select a pump that, when used as a turbine, will satisfy the requirements. Assuming that performance curves for a series of pumps are available*, a typical set of such curves should be normalized using the head, power, and flow rate of the best efficiency point as normal values. These curves will correspond to the right part of Section 1 of either Figure 52 or 53. In normalizing the power P , let $p = P/P_n$ and the curve of p/ν^3 will be identical with the curve of m/ν^2 in Figure 52 or 53. The normalized curves may be compared with the curves in Sections 1 of Figures 52 and 53 to determine which curves best represent the characteristics of the proposed pump. When a choice has been made, the approximate turbine performance can be obtained from the corresponding figure of Figures 60 to 63.

EXAMPLE Assume that the turbine specifications are $H_T = 20$ ft, $P_T = 12.75$ hp and $n = 580$ rpm, and that the characteristics of the DeLaval L10/8 pump are representative of a series of pumps from which a selection can be made. The turbine discharge Q_T in gallons per minute after substituting the above values is

$$Q_T = \frac{3960 P_T}{H_T \eta_T} = \frac{2520}{\eta_T} \quad (39)$$

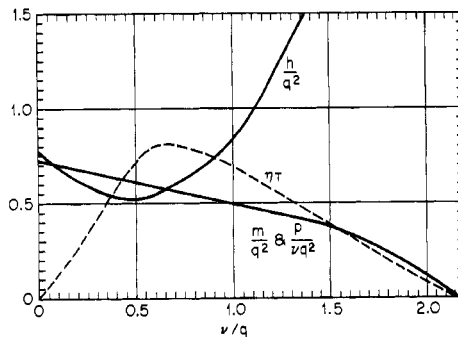


FIGURE 60 Dimensionless characteristic curves for DeLaval L 10/8 pump, constant-discharge turbine operation (Reference 63)

*It is assumed that turbine mode characteristics of the proposed pump are not available when the initial selection is made.

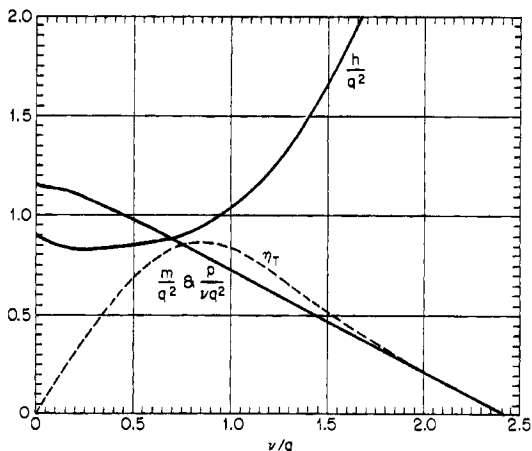


FIGURE 61 Dimensionless characteristic curves for Voith pump, constant-discharge turbine operation (Reference 63)

where η_T is turbine efficiency, shown in Figure 60. Only the normal values Q_n , H_n , P_n , and so on, are common to the curves of both Figures 52 and 60, so these alone can be used in selecting the required pump. Values of v/q , h/q^2 , and η_T are read from the curves of Figure 60 and corresponding values of Q computed by Eq. 39. Values of Q_n and H_n are then given by

$$Q_n = Q(v/q) \quad (40)$$

$$H_n = \frac{H(v/q)^2}{h/q^2} = \frac{20(v/q)^2}{h/q^2} \quad (41)$$

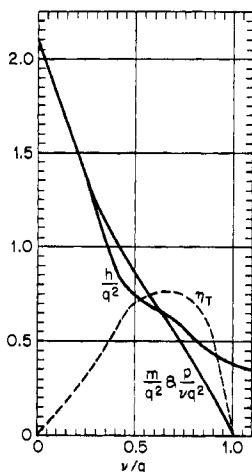


FIGURE 62 Dimensionless characteristics curves for Peerless 10MH pump, constant-discharge turbine operation (Reference 63)

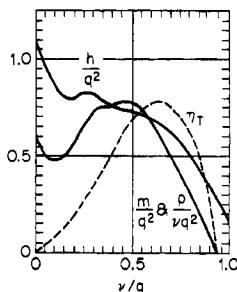


FIGURE 63 Dimensionless characteristic curves for Peerless 10PL pump, constant-discharge turbine operation (Reference 63)

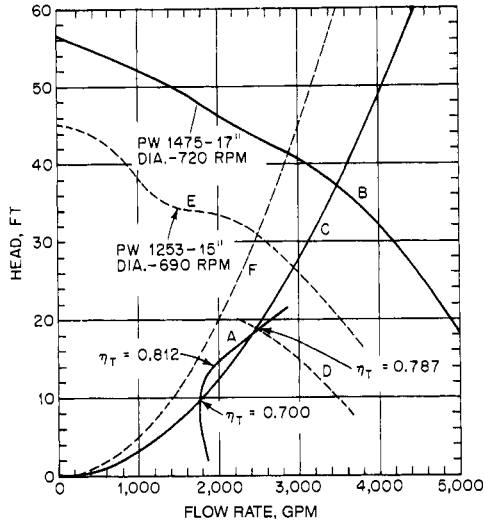


FIGURE 64 Head-flow curves of the pump selection (ft $\times 0.3048$ = m; gpm $\times 0.06300$ = l/s) (Reference 63)

For example, in Figure 60 at $v/q = 0.700$, read $h/q^2 = 0.595$ and $\eta_T = 0.805$. By Eq. 39, $Q_T = 2520/0.805 = 3130$ gpm; by Eq. 40, $Q_n = (3130)(0.700) = 2190$ gpm; and by Eq. 41, $H_n = (20)(0.700)^2/0.9595 = 16.5$ ft. In a similar manner, the locus of the best efficiency points for an infinite number of pumps, each having the same characteristics as shown in Figures 52 and 60, is obtained, and each pump would satisfy the turbine requirements. This locus of best efficiency points is plotted as curve A in Figure 64.

The head curve for a DeLaval L 16/14 pump having a 17-in-diameter impeller tested at 720 rpm is shown as curve B in Figure 64. The best efficiency point was found to be at $Q_n = 3500$ gpm and $H_n = 37.2$ ft. The locus of the best efficiency points for this pump for different speeds and impeller diameters is given by Eqs. 12 as

$$H_n = 37.2 \left(\frac{Q_n}{3500} \right)^2 = \frac{3.04}{10^6} Q_n^2 \quad (42)$$

and is shown by curve C in Figure 64. Curve C intersects curve A at two points, showing that the L 16/14 pump satisfies the turbine requirements. Only the intersection at the higher turbine efficiency is of interest. At this point, $Q_n = 2490$ gpm and $H_n = 18.8$ ft. Because the turbine speed was specified to be 580 rpm, the required impeller diameter is given by Eqs. 12 as

$$D = \frac{(17)(2490/3500)}{580/720} = 15 \text{ in}$$

or by

$$D = \frac{17\sqrt{18.8/37.2}}{580/720} = 15 \text{ in}$$

The computed head curve for the 15-in-diameter impeller at 580 rpm is shown as curve D in Figure 64. The turbine discharge is 3200 gpm from Eq. 39 with $\eta_T = 0.787$.

The optimum solution would be to have Curve C tangent to Curve A at the point corresponding to maximum turbine efficiency, in this case $\eta_T = 0.812$. Because this

would require a smaller pump, Curve *E* in Figure 64 shows a head curve for a K 14/12 pump, which was the next smaller pump in the series. Curve *F*, the locus of the best efficiency points, does not intersect Curve *A*, showing that the smaller pump will not satisfy the turbine requirements. It is important to note that the turbine head, 20 ft, specified for this example was assumed to be the net head from the inlet to outlet flange of the pump when installed and operated as a turbine.

The procedure outlined above should lead to the selection of a pump large enough to provide the required power. However, it probably will be necessary to apply the affinity laws over such wide ranges of the variables that the usual degree of accuracy should not be expected. Considerable care should be exercised if it becomes necessary to interpolate between the curves of Figures 60 to 63. The computed performance will very likely differ from the results of subsequent tests. The curves of Figure 60 may be converted to show the constant-head characteristics of the L 16/14 pump when installed and operated as a turbine. Details of the method of computation are given in Reference 63, and the computed characteristics are shown in Figure 65.

A comprehensive study by Acres American^{64,65} led to a computer program to aid in selecting a pump to meet specific requirements when operating in the turbine mode. Known pump characteristics are entered in the program according to a specified format. The computer compares them with stored characteristics of pumps for which turbine mode characteristics are known and provides estimated turbine mode characteristics for the proposed pump. Vols. I and III of Reference 64 describe the method of computation and give complete instructions for using the program.

Optimized Hydraulic Turbines A logical outgrowth of applying pumps as hydraulic turbines is the optimization of these machines as turbines. Several points of efficiency improvement have been demonstrated over that of pumps running in reverse as turbines.⁶⁶ The fluid accelerates through both the stator or nozzles and the turbine wheel or runner, whereas it decelerates through the same elements when they are acting as impeller and diffuser or volute in the pumping mode. If the fluid never has to decelerate

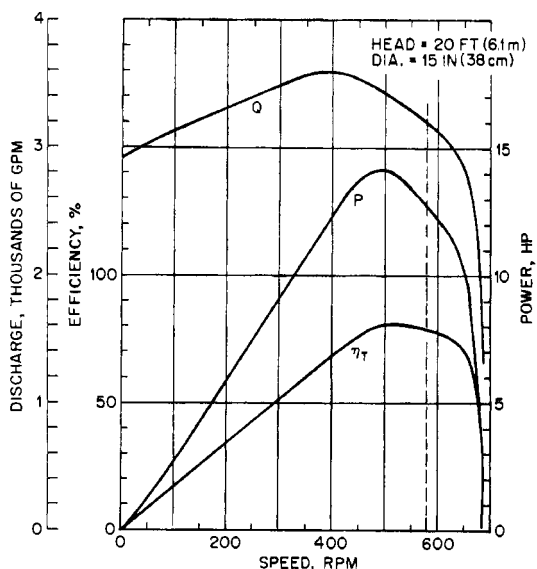


FIGURE 65 Computed constant-head turbine characteristics for DeLaval L 16/14 pump ($\text{gpm} \times 0.06309 = \text{l/s}$; $\text{hp} \times 0.7457 = \text{kW}$) (Reference 63)

because the turbine will never be used as a pump, the nozzles and runner can be designed to produce more aggressive and efficient acceleration in shorter distances than if the same machine is simply a pump being used as a turbine. If it is a radial-inflow runner, the resulting optimized turbine wheel employs a radial blade (with a blade angle $\beta = 90^\circ$ from the tangential direction) at the outer diameter (inlet). This outer diameter is about 75% of the diameter of the impeller of the pump-as-turbine, which further improves the efficiency by reason of the reduction of the disk friction drag of the runner. Design and application of both approaches is explained and compared in Reference 67. Many of the applications of these hydraulic turbines are for power recovery in the pressure let-down processes that occur in petroleum refining. Another variable is introduced in such processes; namely, the evolution of large volumes of dissolved gas as the pressure decreases through successive stages or portions of the turbine. This phenomenon affects the performance of hydraulic turbines for such applications because somewhat more power is produced when gas evolves from the liquid than when a single-phase liquid flows through the turbine at the same pressure drop and mass flow rate.⁶⁸

VORTEX PUMPS

A typical vortex pump is shown in Figure 66.* The ability of this type of pump to handle relatively large amounts of suspended solids as well as entrained air or gas more than offsets the relatively low efficiency. Table 10 lists performance data for four typical vortex pumps and four radial-flow centrifugal pumps of nearly the same head and flow rate. Figure 67 shows the head characteristics of a typical vortex pump with impellers of different diameters together with curves of constant efficiency and constant NPSH. Power curves for the same impellers are shown in Figure 68.

Curves for a conventional radial-flow pump have been added for comparison in Figures 67 and 68. Note that the head of the vortex pump does not decrease as rapidly with an increasing flow rate as does the head of the conventional pump. The power requirement of the vortex pump increases almost linearly with an increasing flow rate, whereas the power required by a conventional pump of about the same specific speed reaches a maximum and then decreases with the increasing flow rate.

Thus if the motor of the vortex pump has been selected to match the power required at the normal flow rate for best efficiency, it will be overloaded if the pump operates much beyond that point.

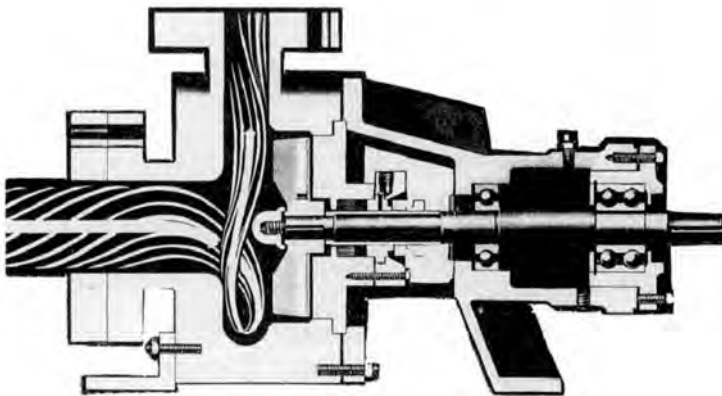


FIGURE 66 Vortex pump (courtesy Fybroc Division, METPRO)

*See also Section 12.2.

TABLE 10 Characteristics of typical vortex pumps and comparable radial-flow centrifugal pumps at 1750 rpm

Impeller	Flow rate,		Total head, ft (m)	Specific speed,		Suction specific speed,		Impeller diameter, in (mm)	Sphere diameter, ^a in (mm)	Efficiency ^b at $Q = Q_n$, %	Shutoff power, ^b % P_n	Power ^b		Head ^b		NPSHR ^b		NPSHR ^b			
	gpm	(l/s)		n_s	(Ω_s)	S	(Ω_n)					at $1.5Q_n$, % P_n	Shutoff head, ^b % H_n	at $1.5Q_n$, % H_n	at Q_n , ft (m)	at $1.5Q_n$, ft (m)					
Vortex	100	(6.3)	24	(7.3)	1614	(0.591)	7700	(2.82)	5 $\frac{7}{8}$	(149)	2	(51)	39.5	37.5	131	120	71	3	(0.9)	8	(2.4)
Radial	108	(6.8)	26	(7.9)	1580	(0.578)	6430	(2.35)	5 $\frac{7}{8}$	(149)	$\frac{5}{8}$	(16)	59	50	108	144	50	4	(1.2)	7.5	(2.3)
Vortex	200	(12.6)	61	(18.6)	1134	(0.415)	7400	(2.71)	8 $\frac{3}{8}$	(213)	2	(51)	45	45	129	123	82	5	(1.5)	8	(2.4)
Radial	225	(14.2)	60	(18.3)	1218	(0.446)	6450	(2.36)	8 $\frac{3}{8}$	(213)	$\frac{3}{4}$	(19)	61	50	130	125	65	6.5	(2.0)	14	(3.4)
Vortex	850	(53.6)	108	(32.9)	1523	(0.557)	9820	(3.59)	11 $\frac{1}{2}$	(292)	4	(102)	59	49	134	120	83	9	(2.7)	15	(4.6)
Radial	900	(56.8)	102	(31.1)	1636	(0.599)	8690	(3.18)	11 $\frac{1}{2}$	(292)	$\frac{7}{8}$	(22)	76	47	113	135	44	11	(3.4)	21	(6.1)
Vortex	1050	(66.2)	150	(45.7)	1323	(0.484)	7088	(2.59)	13	(330)	3 $\frac{3}{4}$	(95)	56	48	137	115	87	16	(4.9)	20	(6.1)
Radial	1250	(78.9)	154	(46.9)	1415	(0.518)			13	(330)	1 $\frac{1}{32}$	(26)	83	52	123	116	45	11	(3.4)	-	(-)
Vortex average					1399	(0.512)							50	45	133	120	81				
Radial average					1462	(0.535)							70	50	119	130	51				

^aDiameter of the largest sphere that will pass through the pump^bSubscript n designates values at the best efficiency point

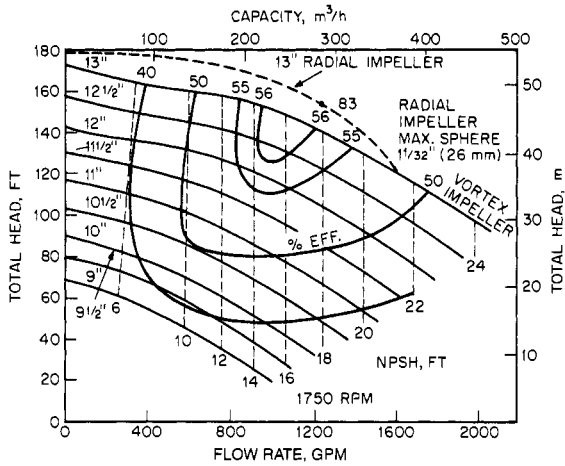


FIGURE 67 Head characteristics of a typical vortex pump. Curves show approximate characteristics when pumping clear water (in $\times 2.54 = \text{cm}$) (Flowsolve Corporation).

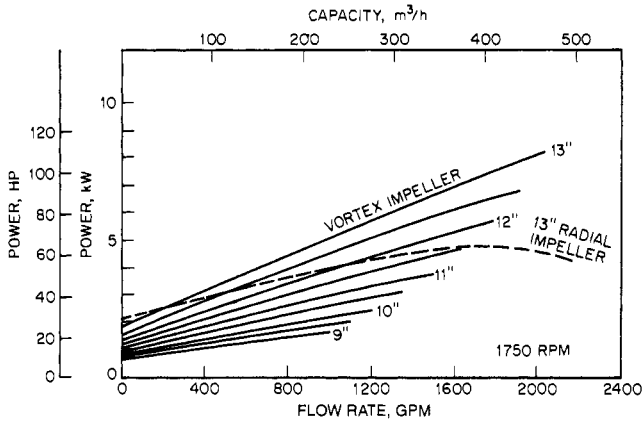


FIGURE 68 Power characteristics of a typical vortex pump. Curves show approximate characteristics when pumping clear water (in $\times 2.54 = \text{cm}$) (Flowsolve Corporation).

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2.1.4 CENTRIFUGAL PUMP MECHANICAL BEHAVIOR AND VIBRATION

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INTRODUCTION

Vibration is the fundamental measure of the mechanical waste energy within rotating machinery. In this respect it is a major performance phenomenon that is distinct from the hydraulic characteristics that the user demands of a pump, which must be so constructed that it will indeed perform as required. When failures occur in pumps and their associated systems, they generally are either caused by excessive vibration, or can be diagnosed by abnormalities in the vibration characteristics of the pump, which in turn are intimately connected with the design, construction, and application of the machine.

This section deals with the sources of pump mechanical distress and vibration, methods of vibration analysis and testing, and the diagnostic interpretation of pump vibration.

BASIC PRINCIPLES OF VIBRATION

Vibration consists of the oscillatory motion of one or more locations on a component. This motion can be considered as a function of time (this is how a person intuitively “sees” or “feels” vibration), or as a function of frequency, that is, how many oscillations per second, or hertz (Hz), are occurring. These are two methods of considering the same motion, and are related to each other through the Fourier Transform, often implemented by engineers and their instruments through use of the “Fast Fourier Transform,” or FFT. The FFT is able to decompose a complex signal into the various frequency components and phase relationships (timing) between these components that must be present in order for the time signal to have the waveform shape that it exhibits when the vibrational motion is plotted versus time. As will be seen, the FFT vibration versus frequency “spectrum” plot is useful in determining the probable source of vibrations, and whether or not the vibration level is unusual or signifies a problem requiring maintenance attention, or even immediate shutdown of the pump.

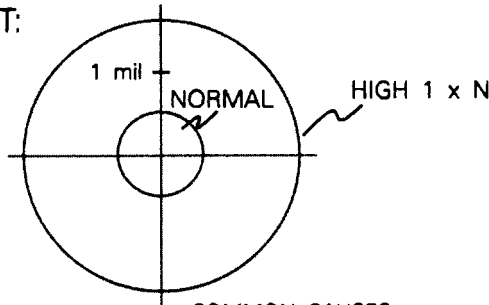
The vibrational response of a pump as a mechanical “system” is the result of the excitation forces (internally or externally imposed) imparted to the system, as amplified or attenuated by the inherent mechanical properties of the system itself. The methods discussed in this section allow differentiation between excitation-related and system-related causes. In general, for pump system diagnostic purposes, the most effective manner for evaluating mechanical vibration and pressure pulsation data is in the frequency domain (i.e., an amplitude versus frequency “spectrum”), because a force occurring at a specific frequency typically must be the result of some physical cause at that same frequency, for example, imbalance at $1 \times$ rotational speed (= running speed), and rotating impeller pulsation caused “vane-pass” frequency at the number of impeller vanes (also known as blades when on a rotating member) times running speed. Sometimes, time domain data contain more easily interpreted information, particularly when sporadic impulsive loads are present, such as during cavitation or rub events. A particularly useful form of time domain data is the “orbit,” in which the shaft motion versus time in two perpendicular lateral directions are plotted at a given axial location along the shaft, thereby showing the path of the centerline of the shaft during one of its revolutions.

Excitation Frequencies Some basic vibration concepts must be reviewed prior to detailed discussion of excitation forces and frequencies. The rotational speed of a pump is typically discussed in units of “revolutions per minute” or rpm. Vibration frequency data are typically discussed in terms of “cycles per second” or Hertz (Hz). A common means of describing vibration frequency is also in terms of multiples of running speed. For example, a $2 \times$ running speed vibration for a pump operating at 3600 rpm would be at a frequency of 120 Hz ($3600 \text{ rpm} \times [1 \text{ min}/60 \text{ s}] \times 2$). Pump oscillating excitation forces are characterized not just by their level, but also by the frequencies at which they impart energy to the system, as will be described for the various typical pump excitations in the following paragraphs.

1. **Rotating imbalance** Approximately 40% of all vibration problems in rotating machinery are caused by rotating imbalance, in the authors’ experience as well as in studies in various applications.¹ If the rotating components of the pump system are not properly balanced, and operate such that the center of gravity of the shaft or attached components is offset from the axis of rotation, the rotor will experience radial forces at the frequency of the pump operating speed N . This type of excitation would, therefore, be categorized as a $1 \times N$ vibration source. The greater the imbalanced mass (m) and associated offset (e) from the rotational axis, the greater the $1 \times$ vibration excitation force. Theoretically, the magnitude of the sinusoidal imbalance excitation force would be: $F = me(2\pi f)^2$ where f is the rotational frequency expressed in Hz. The rotating imbalance force will cause an orbiting motion of the shaft, and as the shaft comes into close proximity with its bearings will cause a portion of the imbalance force to be transmitted from the shaft, through the bearings, to the stationary structure where it can be readily measured as a lateral vibration. Some degree of imbalance will always be present. Methods for balancing the rotor system to remove this excitation source are discussed later in this section.

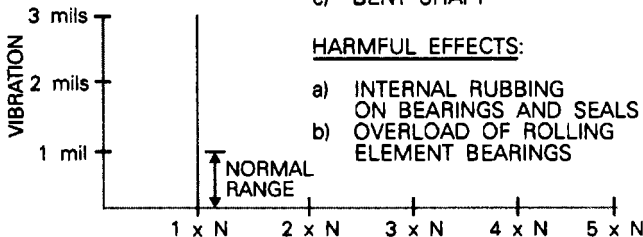
1. RUNNING SPEED ($1 \times N$)

ORBIT:

COMMON CAUSES:

- a) MECHANICAL UNBALANCE
- b) MISALIGNMENT
(USUALLY HIGH $2 \times$ ALSO)
- c) BENT SHAFT

SPECTRUM:

HARMFUL EFFECTS:

- a) INTERNAL RUBBING
ON BEARINGS AND SEALS
- b) OVERLOAD OF ROLLING
ELEMENT BEARINGS

FIGURE 1 Typical orbit and vibration spectrum for a pump with an imbalanced rotor (Mechanical Solutions, Inc.)

Signal processing the vibration data into the frequency domain would reveal a sharp spike at $1 \times$ running speed, proportional to the degree of imbalance, while considering it as an orbit in the time domain would result in a large ellipse, more elliptical than circular in shape if the bearing load is significant or if the bearing stiffness is different in the vertical versus horizontal direction, as is often the case. (See Figure 1.)

2. **Misalignment** In the authors' experience as well as in the same studies discussed with regard to balance issues (e.g., Makay¹), approximately 40% of all vibration problems in rotating machinery are caused by shaft misalignment. Misalignment may be caused by improper installation or extreme differential thermal expansions or pressure-related nozzle loads. The two main types of misalignment are angular and offset. The offset component is the amount that the two centerlines (of the driver and pump) are "offset" from each other (i.e., the distance between the centerlines when extended to be adjacent to each other). The angular component is the differential crossing angle that the two shaft centerlines make when projected into each other, when viewed from first the top, and then in a separate evaluation from the side. In general, misalignment is a combination of both offset and angular misalignment. The vibration data of a misaligned pump system are typically characterized by high $2 \times$ lateral vibration. Sometimes the misalignment load can cause higher harmonics (i.e., rotor speed integer multiples, especially $3 \times$), and may even decrease vibration, because it loads the rotor highly eccentrically against its bearing shell, distorting the otherwise perfect $1 \times$ sine wave that theoretically is caused by residual imbalance. Alternatively, misalignment may actually cause increased $1 \times$ vibration, by lifting the rotor out of its gravity-loaded "bearing pocket," to result in the bearing running relatively unloaded (this can also cause shaft instability, as discussed later). Shown in Figure 2 is a typical orbit and excitation FFT spectrum for misalignment, in which $2 \times$ running

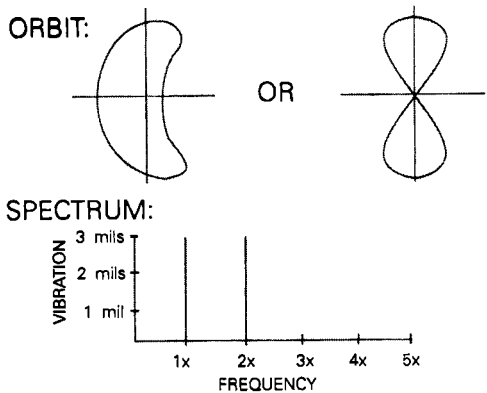
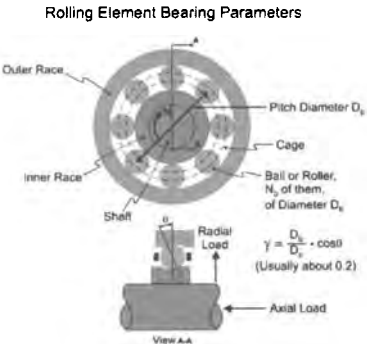


FIGURE 2 Excitation orbit and frequency spectrum for misalignment condition (Mechanical Solutions, Inc.)

speed is the dominant effect. This is often accompanied by relatively large axial motion, also at 2x, because the coupling experiences a nonlinear “crimp” twice per revolution.

- 3. Looseness, rubs** A common cause of pump system vibration is looseness in bearing supports or components attached to the shaft. The main symptoms of such a vibration are a number of harmonics at multiples of running speed, a pulsating orbit and potentially a substantial percentage of vibration at exact whole fractions of running speed, such as $\frac{1}{2}$, $\frac{1}{4}$, and so on. Rubs at a bearing, seal or other running clearance have similar symptoms.
- 4. Rolling element bearings** Rolling element, or “antifriction,” bearings have certain defect frequencies that indicate bearing imperfections. These frequencies are associated with the rate at which flaws in the bearing parts (the inner race, the outer race, the cage, and the rolling element such as ball or needle) interact with each other. Key bearing parameters include the pitch diameter D_p , which is the average of the inner and outer race diameters at the ball interface, the shaft rotational speed N , and the ball-to-race contact angle θ measured versus a plane running perpendicular to the shaft axis. The predominant defect frequencies are shown in Figure 3. Note that no bearing is perfect, and these frequencies are always present, to at least a minute degree, even in a healthy bearing, although the defect frequencies may be masked by other phenomena or by the measurement resolution of the probe or signal analyzer used.



Rolling Element Bearing Frequencies

$FTF = \frac{N}{2} (1 - \gamma)$	Fundamental Train Frequency
$BSF = \frac{D_p}{D_b} \cdot \frac{N}{2} (1 - \gamma^2)$	Ball Spin Frequency
$BPFO = N_b \cdot \frac{N}{2} (1 + \gamma)$	Ball Pass Frequency Outer Race
$BPFI = N_b \cdot \frac{N}{2} (1 - \gamma)$	Ball Pass Frequency Inner Race

FIGURE 3 Rolling element bearing excitation frequencies (Mechanical Solutions, Inc.)

$$F_{gm} = F_{bull} * T_{bull} = F_{pin} * T_{pin}$$

$$F_{pin} = F_{bull} * \frac{T_{bull}}{T_{pin}}$$

$$F_{app} = \frac{F_{gm}}{N_a}$$

$$F_{tr} = \frac{F_{gm} * N_a}{T_{bull} * T_{pin}} = \frac{F_{bull} * N_a}{T_{pin}} = \frac{F_{pin} * N_a}{T_{bull}}$$

where: F_{gm} = Gear Mesh Frequency (Cycles/Minute)
 F_{bull} = Rotational Speed of Bull Gear (Revolutions/Minute)
 F_{pin} = Rotational Speed of Pinion (Revolutions/Minute)
 T_{bull} = Number of Teeth on Bull Gear
 T_{pin} = Number of Teeth on Pinion
 F_{app} = Assembly Phase Passage Frequency (Cycles/Minute)
 F_{tr} = Tooth Repeat Frequency (Cycles/Minute)
 N_a = Number of Assembly Phases (Largest Number that Divides into T_{bull} and T_{pin} without remainder)

FIGURE 4 Gear box excitation frequencies (Mechanical Solutions, Inc.)

5. Gear boxes The primary excitation frequency caused by a gear box is the number of gear teeth times the running speed. In the case of a gear box problem, often highly increased vibration at multiples of the gear mesh frequency as well as $\pm 1 \times N$ sidebands (i.e., modulation) of the gear box harmonics may accompany severe vibration at the gear mesh frequency. A summary of nonmodulated excitation frequencies in gear boxes are shown in Figure 4.

6. Fluid forces

i. *Cavitation.* The phenomenon of cavitation occurs when vapor bubbles form, typically as a result of either insufficient suction pressure ("Net Positive Suction Head Available", $NPSH_A$) or a poor flow pattern at the suction, followed by movement of these bubbles into an area of higher pressure, such that they collapse with great force potentially against impeller or casing surfaces. The collapsing of the bubbles does not occur at a specific frequency, but rather, involves an impulsive load that acts as a broadband excitation to excite a very broad range of frequencies, exciting impeller and casing natural frequencies even into the ultrasonic range. Evidence of cavitation and its strength is therefore typically not evident as a spike on a frequency spectrum (instead it slightly raises the spectrum fairly evenly over a very broad range), but is clearly discernable as powerful random pulses in a pressure or vibration versus time high resolution plot. An example of cavitation pulses in the time domain as evidenced in such plots is given in Figure 5, for a double suction single stage pump in low suction pressure water service.

ii. *Fluid whirl.* Shaft part-speed (i.e., below running speed, or "subsynchronous") whirl is a forced response at a frequency typically below the first noncritically damped shaft bending natural frequency. It is driven by a moving static pressure field which rotates at some speed below running speed. This fluid rotational speed becomes the whirl speed. The most common cause of whirl is fluid rotation around the impeller front or back shrouds, or in the journal bearing, wear ring, or balance drum clearances. Such fluid rotation is typically about 45% of running speed,

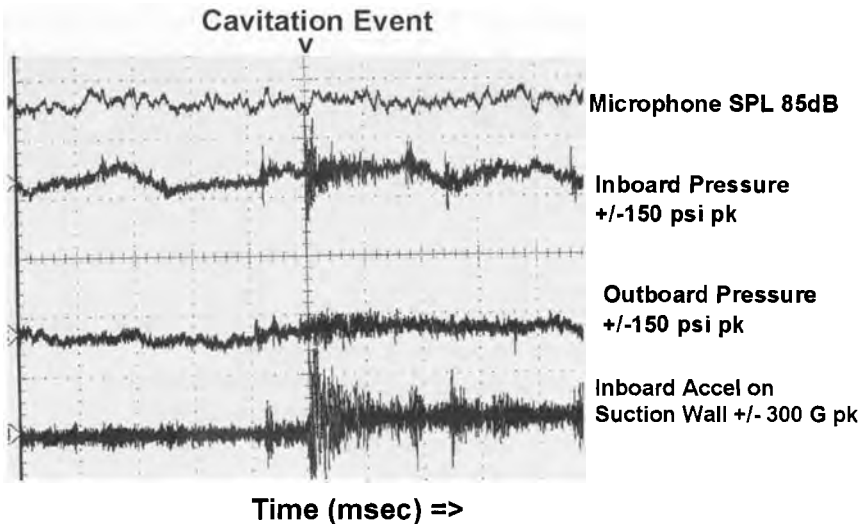


FIGURE 5 Cavitation pulses versus time (0 to 10 msec) in inboard and outboard suction bays of a double suction water pump. The pulse was in the inboard bay in this instance. The microphone, 6 ft away from the inboard suction wall, was not responsive at the high frequency (order of 50 kHz) of the pulse and its ring-down. Note the randomly occurring lower level pulses. In the case where a low pressure zone occurs at one clock position of the inlet, the pulses tend to cluster in groups associated with the impeller vane passing the low pressure zone. According to the authors' analysis, the amplitude and frequency of occurrence of the strongest pulses correlate with erosion damage potential. Bubble collapse in the middle of the passage, away from the walls, causes no damage, and only weak pulse transmission to the walls, as detected by a high frequency accelerometer. (Mechanical Solutions, Inc.)

because the fluid is stationary at the stator wall, and rotating at the rotor velocity at the rotor surface. Thus the bulk of the fluid in the running clearance rotates at roughly half speed.

If this whirl frequency increases to become coincident with a poorly damped natural frequency of the rotor/bearing system, the whirl can transform into a more dangerous motion called "whip." Once shaft whip starts, self-excitation occurs at the natural frequency of the shaft which is coincident with the whirl, such that the vibration response frequency "locks on" to the natural frequency, so that as speed is increased further, the whirling frequency no longer increases in proportion, but remains the same as the shaft natural frequency. During whip, the normally primarily $1\times$ running speed frequency spectrum and roughly circular shaft orbit show a strong pulsing peak at about 45% of the running speed at which whip initiated. In the orbit, whip shows up as a slowly rotating loop reflecting orbit pulsation every other revolution. In rotating machinery, whip is the most common form of "rotordynamic instability," in which dangerously high vibrations occur in spite of how well the machine is balanced, aligned, or otherwise designed. Fortunately, whip is far more common in compressors and turbines than in centrifugal pumps, although it can only be ruled out by either experience with the pump and system in question, or by careful rotordynamic analysis and/or testing.

- iii. *Vane pass.* Each time a pump impeller vane (or blade) moves past the volute "cut-water" or diffuser vane, a pressure pulse is imparted to it. The vane pass excitation frequency is therefore at the running speed times the number of impeller vanes, as indicated in Figure 6. This load condition can have a severe effect if any stationary structure or combined rotor and stationary structure natural frequencies happen to coincide with the vane pass frequency.

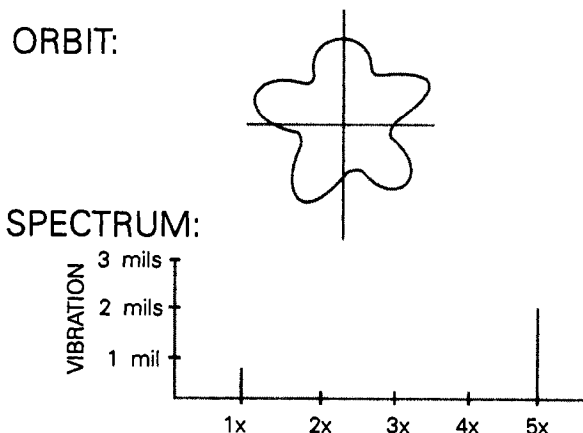


FIGURE 6 Excitation orbit and frequency spectrum for excessive vane pass pulsation (Mechanical Solutions, Inc.)

- iv. *Internal suction or discharge recirculation.* If a pump is operated too far away from its best efficiency point (BEP), the angle of attack of incoming flow on the rotating impeller vane is different from that of the optimal design. Vane stalling can occur at either the suction or discharge, leading to portions of the flow reversing direction. The reversing flow is termed internal recirculation (see Section 2.1.1), and can lead to strong vortices, which can develop very low pressure at their tips while they are in the flow passage, causing cavitation, and/or which buffet the pump as they rotate past the impeller vanes. This tends to cause amplitude modulation of the running speed and vane pass frequencies, with a broadband frequency range associated with the statistical average of the vortex rotating speeds (the vortex axes typically slip substantially relative to running speed).
- v. *Rotating stall.* Sometimes at the off-BEP flows associated with the internal recirculation condition, and due to the same principle of poor angle-of-attack on the impeller or diffuser vanes, rotating stall may occur, in which a recirculating stall cell (which is generally a recirculation vortex) will propagate circumferentially. This stall cell rotation in an impeller is opposite to the spin of the impeller, by 5 to 35% of rotating speed, resulting in stall cell rotation relative to ground of $0.65\times$ to $0.95\times N$. If the stall cell occurs in the diffuser, it rotates in the same direction as the impeller spin, at a frequency of about 0.05 to $0.35\times$. Although the rotating stall results in a "subsynchronous frequency" (i.e., below running speed), which few phenomena possess other than rotordynamic instability, they are not instabilities but rather forced vibrations, controllable by keeping their rotation frequency at least 10% away from any rotor or structural natural frequency. Typical orbits and spectra for these various types of subsynchronous vibration are shown in Figure 7, and are compared to orbits and spectra for rub or rotordynamic instability.

Natural Frequencies and Mode Shapes The inherent mechanical properties of a system important to its vibrational response include its stiffness, mass, and energy absorption as reflected in the natural frequencies, mode shapes and damping of the rotor and stationary structural system. If a cyclic load is applied to a structure, the response of that structure is dependent on the frequency at which it is excited. At specific frequencies, termed the natural frequencies, the response is dramatically increased, a phenomenon termed resonance. When designing and installing pumps and their systems it is important to ensure that the natural frequencies of the system do not coincide with anticipated excitation frequencies, such as those described above. If an appropriate separation margin

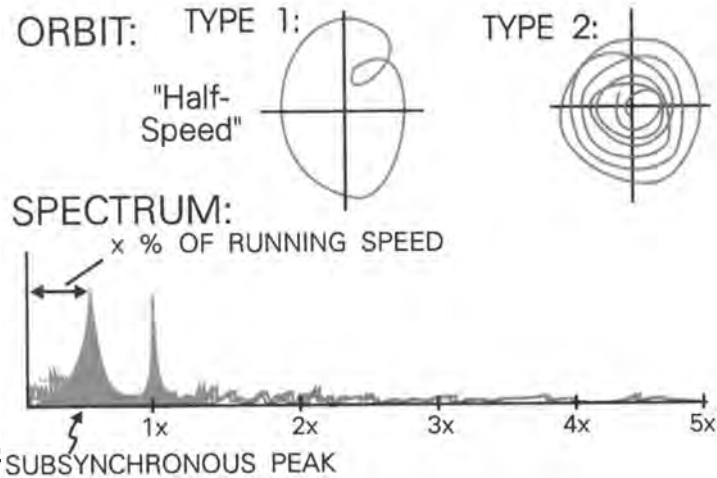


FIGURE 7 Excitation orbit and frequency spectrum for: (Type 1) Rub if $x = 0.5x$, (Type 1) rotordynamic instability if $x = 0.40$ to $0.49x$, or (Type 2) rotating stall in the diffuser if $x = 0.05$ to $0.35x$ or in the impeller if $x = 0.65$ to $0.95x$ (Mechanical Solutions, Inc.)

does not exist, excessive vibration can result, even for relatively low energy excitations. A resonance near running speed can result in a pump being chronically “balance sensitive” or “alignment sensitive.”

Each natural frequency has a deflection shape, or mode shape, associated with it. This is the shape that the structure takes on during vibration at that frequency. The mode shape also plays a role in the vibration response level. If the applied excitation forces occur at points in the structure which exhibit much movement, then a greater response is expected. Analytical methods for determining system natural frequencies and mode shapes are discussed later in this section.

Natural frequencies of rotating systems are termed critical speeds if their amplification factor is high enough (over 2.5, per API-610²). When a rotor's revolution is at a critical speed, the $1\times$ excitation that is always present to some degree will excite the corresponding natural frequency and can cause severe vibration even for normally acceptable levels of imbalance. The natural frequencies and mode shapes can be calculated using the principles of rotordynamics, which are discussed later in this section. Note that the natural frequencies of the rotating system will have dependence on the running speed. For example, at higher speeds, gyroscopic effects become significant and can stiffen lateral modes. Additionally, hydrodynamic effects caused by fluid passing through annular spaces around the shaft also vary with running speed and affect the natural frequencies through the Lomakin effect.³ Rotor critical speeds should be determined in the design stage of a pump system so that such conditions can be avoided.

As an example, consider the lateral vibration of an impeller-between-bearings multi-stage pump rotor system. One such typical rotor exhibited the natural frequencies and mode shapes shown in Figure 8.

Resonance, Damping, and Phase Angle When cyclic forces are applied to a system at a frequency equal to its natural frequency a resonant vibration occurs. Damping provides energy absorption and reduces the response. In a centrifugal pump, damping is provided from several sources such as viscous effects of hydrodynamic bearings, hydrostatic effects at annular seals, and structural damping inherent in the materials of construction. Figure 9 depicts the effects of damping on the frequency response function of a system.

**Typical rotor vibration response vs. speed
exhibits several natural frequencies**

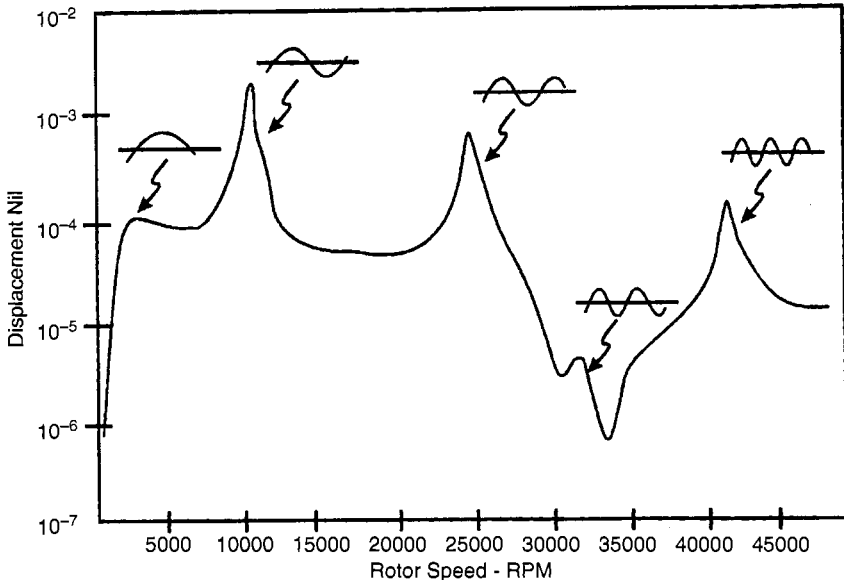


FIGURE 8 Typical mode shapes for a multistage centrifugal pump (Mechanical Solutions, Inc.)

In resonance, the vibration energy from the last peak of the force has come full cycle, and is reinforced when the next peak takes place. The vibration in the next cycle will then include movement due to both peaks, and will be higher than it would be for a single application of the force. The vibration motion keeps being amplified in this way until its large motion uses up more energy than the amount of energy that is being supplied by each additional peak. Unfortunately, the motion at that point is generally quite large, and damaging. To avoid resonance, the natural frequencies of the pump rotor and structure must be well separated from the frequencies at which such repetitive “dribbling” type forces will occur, which tend to be $1\times$ running speed (typical of imbalance), $2\times$ running speed (typical of misalignment), or at the number of impeller vanes time running speed (so-called “vane pass”).

In pumps, the vibration amplification (often termed “ Q ” as shown in Figure 9) due to resonance is usually between a factor of 2 and 25 times what it would be if the force causing the vibration were steady instead of oscillating. Q depends on the amount of energy absorption, called “damping,” which takes place due to oscillation. In an automobile suspension, this damping is provided by the shock absorbers. In a pump, it is provided primarily by the bearings and the liquid flowing axially between the rotor and stator in the “annular seals,” like the balance piston.

Another important concept in vibration is the “phase angle,” which measures the time lag between the application of a force and the vibrating motion which occurs in response to it. A phase angle of zero degrees means that the force and the vibration due to it act in the same direction, moving in step with one another. This occurs at very low frequencies, well below the natural frequency. An example of this is a force being slowly applied to a spring. Alternately, a phase angle of 180 degrees means that the force and the vibration due to it act in exactly opposite directions, so that they are perfectly out of step with each other. This occurs at very high frequencies, well above the natural frequency. At the point of resonance (frequency = natural frequency), phase angle is theoretically 90 degrees.

**“FFT” or Signature Plot:
Vibration vs. Speed (or vs. Frequency)**

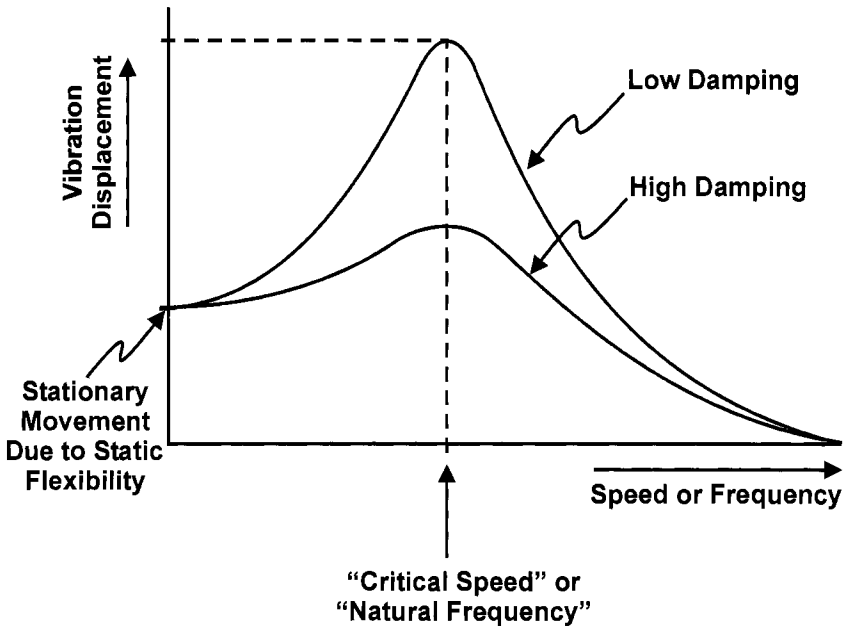


FIGURE 9 Vibration resonance spectrum, displacement x versus frequency f (Mechanical Solutions, Inc.)

Phase angle ϕ is important because it can be used together with peaks in vibration frequency field data to positively identify natural frequencies as opposed to excessive excitation forces. Such identification is useful in order to determine what steps should be taken to solve a vibration problem. The role of phase angle is also important in recognizing and solving rotordynamic instability problems, which typically require different solutions than resonance or excessive oscillating force problems.

These concepts can be described mathematically for simplified vibration systems. For example, consider the mechanical system displayed in Figure 10. The system is comprised of mass m , stiffness k , damping c , and is operated on by a sinusoidal force F as a function of time t : $F(t) = F_0 \sin(\omega_f t)$. The dynamic displacement response of such a system would be $x(t) = X \sin(\omega_f t + \phi)$.

For this so-called single degree of freedom (DOF) system, the natural frequency is $\omega_n = (k/m)^{0.5}$ rad/sec, which can be divided by 2π to be put in terms of cycles per second (Hz), and can be further multiplied by 60 to be put in terms of f_n with units of cycles per minute, the same units as pump rpm. Note that k is the effective stiffness of the system, and m is the effective mass of the system, which in USCS units must be divided by the gravitational constant g_o to get consistent units. (See the definition of g_o in the nomenclature of Section 2.1.1.)

For a static loading force equal to F_o , the static displacement S of the mass is equal to F_o/k as shown in Figure 8. For a cyclic force, the amplitude of vibration of the mass, designated by X , is equal to $F_o / ((k - m\omega_f^2)^2 + (c\omega_f)^2)^{0.5}$, where ω_f is the forcing frequency in rad/sec, and c is the damping constant. The maximum amplitude of vibration P occurs when the forcing function frequency ω_f is near the natural frequency. It can be shown that for small values of damping, the magnification factor $P/S = 1/(2\zeta)$, where the damping factor

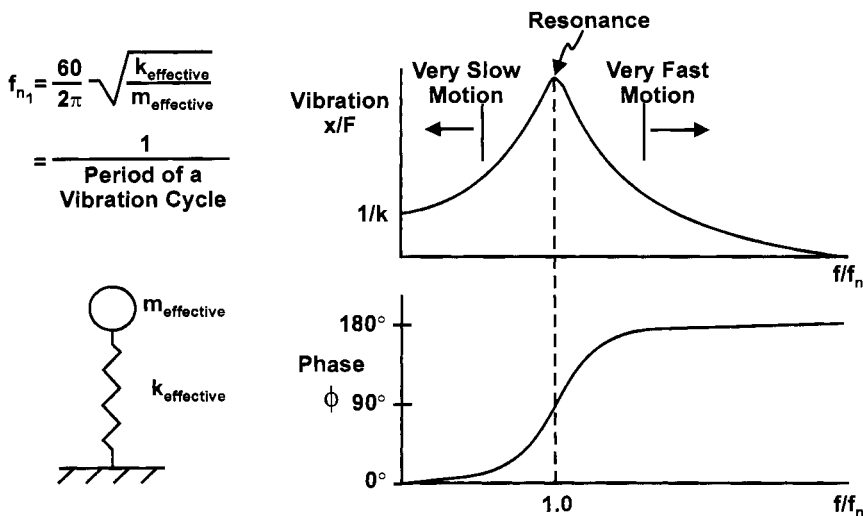
Natural Frequency**How Phase Relates to Natural Frequency:**

FIGURE 10 Single degree of freedom system (Mechanical Solutions, Inc.)

ζ is defined as the ratio of the damping c to the critical level of damping c_c . A critically damped system will not oscillate if perturbed momentarily from an at rest position. The critical damping level in terms of pump mechanical properties is $c = 2(km)^{0.5}$. Typical pumps have damping significantly less than critical (typically $\zeta = 0.02$ to 0.1).

Often mechanical aspects of a pump system can be reasonably simplified into a single DOF system such as the one described, and approximate natural frequencies and response levels can be estimated. For example, the behavior of the motor of a typical vertical turbine pump resting upon the discharge head of the pump may be very similar to a single DOF system in its first "above ground" dominated bending mode. Similarly, the effective mass and stiffness the rotor of an overhung impeller pump may be estimated to allow for a critical speed prediction. Even mechanical systems that are significantly more complicated than single degree of freedom systems can have their vibratory behavior described as a combination of natural frequencies, each with an effective mass (perhaps distributed in several locations), stiffness, and damping.

THE ROLE OF VARIOUS PUMP COMPONENTS

Impeller Effects on Vibration Impellers are typically the heaviest components and have the largest mass moments of inertia in the pump rotor system. The transverse moment of inertia of an impeller is sometimes neglected in lateral analyses, and this can result in significant errors, particularly for overhung impeller or multistage pumps, or in high-speed applications. With the advent of modern solid modeling software, not only weight, but also the polar and transverse moments of inertia, can be accurately determined from solids models of impellers.

Shafts: Lateral, Torsional, and Axial Stiffness and Motion In general, the lateral stiffness of shafts is directly proportional to the fourth power of the diameter and inversely proportional to the cube of the length. Typically the lengths that govern the lateral stiffness or flexibility of the shafts are the spans between bearings rather than the entire shaft

length. In the case of overhung pumps, the distance between the impeller(s) and the closest bearing can be more important than the distance between bearings. The torsional stiffness of shafts is directly proportional to the fourth power of the diameter and inversely proportional to the length of the shaft. The axial stiffness of the shaft is directly proportional to the square of the diameter and inversely proportional to the length of the shaft. Both the lateral stiffness and the axial stiffness of shafts are directly proportional to the modulus of elasticity, E , of the material, whereas the torsional stiffness is directly proportional to the modulus of rigidity, G . The selection of bearing span and shaft diameter will dictate the overall stiffness of the shaft and whether the pump will run above or below its first critical speed.

Bearings The purpose of bearings is to provide the primary support to position the rotor and maintain concentricity of the running clearances within reasonable limits. There are several types of pump bearings. The most common type is the plain journal bearing, in which a smooth, ground shaft surface rotates within a smooth surfaced circular cylinder. In this bearing, the load “bearing” effect is provided by a hydrodynamic wedge which builds between the rotating and stationary parts as rotating fluid flows through the narrow part of the eccentric gap between the shaft journal and the cylindrical bearing insert. The eccentricity of the shaft within the journal is caused by the net radial load on the rotor forcing it to displace within the fluid gap. The hydrodynamic wedge provides a reaction force which gets larger as the eccentricity of the shaft journal increases, similar to the buildup of force in a spring as it is compressed. The plain journal bearing has moderately high stiffness, and is relatively inexpensive, with good damping properties, but has high cross-coupled stiffness. The concept of how cross-coupled stiffness is created in a journal bearing or annular sealing cavity is shown in Figure 11. Cross-coupled stiffness can negate the damping, leading to rotordynamic instability. Putting axial grooves in the journal bearing significantly diminishes the cross-coupling. Other types of pump bearings include the rolling element (“antifriction”) bearing (inexpensive and stiff, but with little damping), the tilting pad bearing (moderate stiffness and damping, low cross-coupling, but expensive), the magnetic

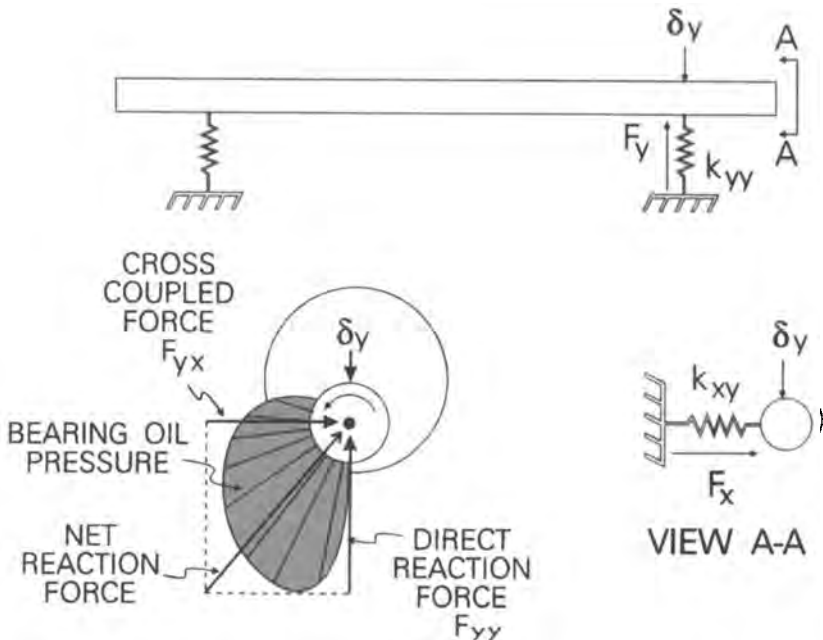


FIGURE 11 The concept of cross-coupled stiffness (Mechanical Solutions, Inc.)

bearing (low stiffness and expensive, but zero cross-coupling), and the hydrostatic bearing (high stiffness and damping, low cross-coupling, but expensive to install and operate).

Annular Seals and the Lomakin Effect One of the most common types of flow-path seals is the annular seal, with either smooth cylindrical surfaces (plain seals), stepped cylindrical surfaces of several different adjacent diameters (stepped seals), or multiple grooves or channels perpendicular to the direction of flow (serrated, grooved, or labyrinth seals). These areas include the impeller front wear ring, the rear wear ring or diffuser interstage bush rings, and the thrust balancing device leak-off annuli.

Annular seals can dramatically affect pump rotor dynamics, especially multistage pump rotor dynamics, in two ways :

1. By changing the rotor support stiffness and therefore the rotor natural frequencies, thereby either avoiding or inducing possible resonance between strong forcing frequencies at one and two times the running speed and one of the lower natural frequencies.
2. By increasing the rotor support cross-coupled stiffness or, as the seals wear, decreasing the rotor damping, allowing greater magnification factors during resonance, and in the extreme allowing destabilizing forces on the rotor to dominate those associated with stabilizing energy absorption.

These effects are particularly strong in multistage pumps, because multistage rotors are relatively long and flexible. Since the annular seals are primarily in the central portion of the rotor, where they exercise considerable leverage on the first bending mode of the rotor, the contribution of seal stiffness to the rotor support can be comparable to the rotor stiffness itself.

The stiffness and damping of an annular seal is provided in small part by the squeeze-film and hydrodynamic wedge effects well known to journal bearing designers. However, because of the high ratio of axial to circumferential flow rates in annular liquid seals relative to bearings, large forces can develop in the annular clearance space due to the circumferentially varying Bernoulli pressure drop induced as rotor eccentricity develops, as shown in Figure 12. This is known as the Lomakin effect and is the largest stiffness and damping force-generating mechanism within pump annular seals.³ The primary action of the Lomakin effect is beneficial, through increased system direct stiffness and damping which tend to increase the rotor natural frequency and decrease the rotor vibration response at that natural frequency.

The Lomakin effect depends directly on the pressure drop across the seal, which for constant system flow resistance results in a variation of the Lomakin support stiffness with roughly the square of the running speed. However, in cases of high static head compared to head rise at BEP, there is only a small variation of the Lomakin effect with pump speed. The other important parameters are a diminishing increase in stiffness with increasing annular sealing cavity length, stiffness increase in proportion to diameter, and stiffness decrease in inverse proportion to clearance. Fluid properties are of secondary importance unless very high viscosity liquids are involved.

The Lomakin effect can be calculated with reasonable accuracy through use of bulk flow computer programs. An approximate method for predicting Lomakin effect stiffness is though the following equation, applicable to short L/D sealing gaps in liquid water of any temperature:

$$K_L = k \Delta P R^2 L / C_R$$

Where K_L = Lomakin Stiffness in lbf/in USCS or kN/mm in SI units

k is a unitless constant equal to about 0.4

ΔP is pressure drop in psi in USCS or MN/m² in SI units

R is radius in inches in USCS and mm in SI units

L is length in inches in USCS and mm in SI units

C_R is average radial clearance in inches in USCS and mm in SI units

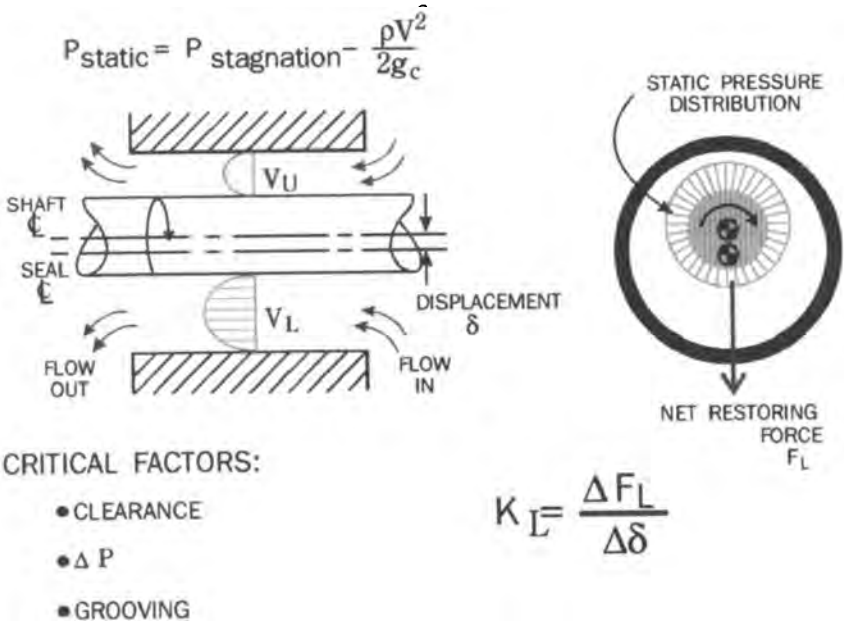


FIGURE 12 The concept of Lomakin effect stiffness K_L (Mechanical Solutions, Inc.)

The physical reason for the strong loss of the Lomakin effect with increased clearance is that increased clearance or circumferential grooving gives the opportunity for equalization of circumferential pressure distribution, which is behind the Lomakin effect. It is important that modern rotors be designed with sufficiently stiff shafts so that any natural frequency that starts above running speed with new clearances remains above running speed with clearances worn to the point that they must be replaced from a performance standpoint.

Mechanical Seals and Packing Mechanical seals do not affect vibration, or specifically pump rotordynamics, significantly. They do not add excitation forces, and unlike most wear rings provide negligible support stiffness and damping. Packing likewise usually has negligible stiffness, but can add considerable damping. When pump designs get changed from packing to mechanical seals, often rotor critical speed problems begin to surface that went unnoticed previously.

Sleeve Effects on Vibration Sleeve effects on vibration depend on the fit-up of sleeves to the shaft and the type of analysis being performed. For a lateral analysis, if the sleeves are press fit to the shaft with tight class of fit, then the added stiffness of the entire length of the sleeve should be included, in the authors' experience; (although, API-684 suggests taking no credit for sleeve stiffness in any circumstance.⁴) If the sleeve is lightly pressed or loose fit to the shaft then no added stiffness is included in the model, but the mass and transverse moment of inertia is to be included at the center of gravity of the sleeve. In a torsional analysis if the sleeve is press fit to the shaft, then it is treated in the same manner as in the lateral analysis, but if the sleeve is loose fit and keyed to the shaft, $\frac{1}{3}$ the diameter of the shaft where the sleeve is located is subtracted from each side of the sleeve and what is left is used as added stiffness to the shaft. When modeling coupling hubs, if the hub is press fit to the shaft then the entire length of the hub is included as added stiffness. If the hub is keyed on with a loose fit, then a $\frac{1}{3}$ penetration of the shaft into the hub is used to limit added stiffness. In both cases, the full mass and moment of inertia are to be included in the analysis.

Horizontal Pump Casings and Bearing Housings Horizontal pump casings are relatively massive compared to steam turbines and compressors, and historically have not played a strong role in pump rotordynamics, other than to act as a rigid reaction point for the bearings and annular seals. However, pressure on designers to save on material costs occasionally results in excessive flexibility in the bearing housings, which are cantilevered from the casing. The approximate stiffness of a bearing housing can be calculated from beam formulas given in Roark, such as $3EI/L^3$, where L is the cantilevered length of the bearing center from the casing end wall, E is the modulus of elasticity, and I can be determined from the cross section of the bearing housing. The bearing housing stiffness must be combined as a series spring with the bearing film stiffness to determine a total direct “bearing” stiffness for use in rotordynamics calculations. The following series spring formula may be used:

$$\frac{1/k}{\text{total}} = \frac{1/k}{\text{housing}} + \frac{1/k}{\text{bearing}}$$

If resources are available, an accurate determination of the bearing stiffness can be obtained through 3D finite element analysis of the bearing housing. A unit load can be applied to the bearing at the location of the internal journal or ball bearing race, and the applied load divided by the deflection at the load provides the stiffness of the bearing housing.

Vertical Turbine Pumps A vertical turbine pump (VTP) can be described as a vertically mounted pump where the impeller is located in a cylindrical casing mounted in-line to an attached vertical column pipe. The impeller takes fluid directly from the well through a suction bell and bowl assembly, and discharges coaxially into a multivane diffuser and then into the attached column pipe. The column pipe in turn is attached to a discharge head which in turn mounts the pump driver. The discharge head and pump casing may be separated by as little as a few inches (order of 100 mm) or as much as a few thousand feet (order of 1000 m). The pump impeller and driver are connected through intermediate “lineshaftering” running down the center of the column pipe and supported by lineshaft bearings spaced according to shaft diameter. If the pumped fluid is corrosive, abrasive, or contains hard particulate, the bearings and intermediate shafting may be enclosed within a pipe (“shaft tube”) run concentrically with the column pipe to protect them from damage. In such a case, this enclosing tube is tensioned between the discharge head and the pump casing.

VTPs generally have much more flexible motor and pump casings than comparable horizontal pumps, and more flexible attachment of these casings to the foundation. Due to the flexibility of the discharge heads used to support motors, many structural natural frequencies of vertical pumps tend to fall very close to the $1\times$ running speed of the pumps, and can cause very high vibrations to occur. In addition, the foundation that the pump is mounted on can play a significant role. Finally, the below-ground column piping and pump casing or “bowl assembly” as well as the water added-mass effect need to be assessed to determine if the below-ground components will have a vibration issue, or will combine with the above-ground components and foundation (and possibly piping) to cause a system-related problem. To adequately include discharge head, baseplate, piping, column/bowl assembly, and foundation effects in such pumps, a finite element analysis (FEA) model is required.

If a manual model of a VTP is attempted, the baseplate and foundation may be modeled as torsional springs in series. The spring constants can be based on force versus deflection formulas provided for plates with central moments (see Timoshenko) and for separate “half space” foundations.⁵ These hand calculations may produce large errors and should be verified using detailed finite element analysis or field “bump” test results.

The natural frequencies of concern in a VTP are the first several structural bending natural frequencies of the entire assembly, and the enclosing tube if one is used. Lineshaft natural frequencies are occasionally a problem, but generally present a “moving target” to excitations, in terms of their frequency value, because the lineshaft bearings provide very inconsistent support, and so the shaft natural frequencies shift from instant to instant like a guitar string as finger pressure is placed in different locations. The main source of excitation

for both the structure and the rotor occurs at $1\times$ running speed due to imbalance or shaft bow, and to a lesser extent misalignment. In particular, driver coupling misalignment can influence discharge head vibration at both $1\times$ and $2\times$ running speed. Vane pass excitation is normally not of concern since most VTP casings are designed with multivane diffusers as well as greater impeller vane/diffuser vane spacing, and this minimizes the effect of vane pass frequency in exciting natural frequencies.

The nominal lineshaft (or "intermediate shaft") bearings are normally spaced to place the first lineshaft natural frequencies above running speed, as per recommendations in API-610.² These nominal natural frequencies can be estimated if the shaft diameter and bearing spacing is known by using continuous beam equations, where the bearings act as simple supports. Axial loads on the shaft tend to straighten the shaft and raise the natural frequencies above that calculated particularly if the load is high or the shaft bearing spacing is long. In this case the axial stiffening of the shaft must be taken into account using equations developed to account for axial loading.⁶ The pump shaft bearings are typically spaced close together such that the pump shaft natural frequencies are well above $2\times$ running speed.

Water both internal and external to the pipe must be taken into account (e.g., the methods of Blevins⁷) or the calculations will be substantially in error. Column pipes are normally set so that a portion is above the water level and a portion below. The column pipe natural frequency calculation should account for the transition from water added mass being internal only to being both internal and external. Similar to the shafting, the enclosing tube natural frequency calculation needs to be adjusted due to axial stiffening of the tube if it is tensioned between the discharge head and the pump casing.

Natural frequencies for VTP discharge heads should be placed to avoid excitation by running speed and to a lesser extent $2\times$ running speed. Prior to the introduction of finite element analysis many manual calculation and simple spreadsheet methods were used to calculate the discharge head natural frequencies, for example using beam equations from Roark,⁵ or plate/shell calculations such as Timoshenko.⁶ These methods proved to be approximate and still may be suitable for low horsepower (less than 200 bhp or 150 kW) constant speed applications where a single running speed is to be avoided and calculation error can be absorbed. The horsepower limit is given to acknowledge that as the power goes up the overall stiffness of the structure may not increase as quickly as the imbalance force. Running at a natural frequency may produce acceptable vibration in a 10-bhp (7.5-kW) unit but most certainly will produce high vibration in a 100-bhp (75-kW) or more unit. Additionally, shifting the natural frequency of a 10-bhp (7.5-kW) unit to a more suitable location in the field is more easily accomplished if excessive vibrations are encountered. Outside these limits FEA has proven to be a highly accurate (within 5 to 10%) method when employed by personnel familiar with the characteristics of both VTPs and FEA.

Vertical Sewage Pumps Vertically mounted sewage pumps, while typically much more stiff than vertical turbine pumps, are also susceptible to natural frequencies that fall within the operating range of the pump. Like VTPs, vertical sewage pumps have a large potential for exciting a structural natural frequency at $1\times$ or $2\times$ running speed, or vane pass frequency. In addition these units are tall, which in itself makes them susceptible to higher vibration levels since the point of rotation for the unit center of gravity is about the base and allows a small force to create a larger moment. In general a structural lateral natural frequency analysis of the motor/pump assembly should be performed on all new designs and on all variable speed units larger than 100 bhp (75 kW). Up-front analysis work on units less than 100 bhp (75 kW) may be more expensive than field fixes and therefore may not make economic sense for all applications.

If the motor is directly mounted to the volute casing, there is much higher likelihood that a cantilever bending mode of the motor could coincide with the $1\times$ running speed of the pump. If the pump is driven by a long drive shaft with the motor well above the pump, there is less chance of a bending mode of the cantilevered motor, but the drive shaft natural frequencies need to be evaluated. In addition, the bearing towers used to support bearings on sewage pumps are often mistakenly considered as rigid, when in reality they contribute quite a bit of flexibility to the rotor system, and can lead to combined rotor/structural modes.

Typical nonclog pumps are a single-volute design utilizing an overhung impeller with 1 to 5 vanes, operating at synchronous speeds between 300 and 1800 rpm. Historically, in all but the largest stations, nonclog pumps have been operated start/stop at full and load speed as incoming flow demanded. However, recent trends in station design have shown an increased use of variable speed drives to match incoming and outgoing flows. Variable speed operation is now the norm in all but the smallest stations. With the advent of variable frequency drives (VFDs), in addition to the great operating cost savings provided by eliminating the need to throttle the pump discharge, the possibilities of exciting rotor and structural natural frequencies have increased dramatically, and the need to analyze these pumps has grown along with it.

The three key excitation frequencies of $1\times$, $2\times$, and vane pass appear in the vibration spectra even for properly designed nonclog pumps. $1\times$ occurs due to residual imbalance and coupling misalignment produces both $1\times$ and $2\times$. In 1- to 3-vane impellers, vane pass occurs as a strong excitation, greater than the imbalance force due to the low number of vanes used. In addition, the impellers are designed to pass solids and therefore have wide discharge passages, susceptible to excitation by pressure pulsations. Additionally the majority, by a large margin, of nonclog pumps are a single-volute casing design which produces an unbalanced impeller radial load; (see the work of Agostinelli et al in Section 2.1.3.⁹) Combining the vane pass frequency with $1\times$ and $2\times$ excitation produces a range of excitation sources which are difficult to avoid, and the possibility of exciting a natural frequency is very high if the unit has variable speed.

Most nonclog pump shafts use an overhung impeller design with bearings specified for a life in excess of 50,000 L-10 h and a shaft deflection at the stuffing box held to less than 0.002 in. The long bearing life and maximum shaft size (beyond that necessary for strength, in order to minimize shaft deflection) make it unusual to find a nonclog pump shaft that runs with a first mode lateral natural frequency below the operating rpm. At least 20% margin is recommended.

The operating rpm/power and impeller solid size capabilities combine to produce relatively large polar inertia for the impeller and motor rotor in nonclog pumps. It is not uncommon to find the first torsional natural frequency within the range of $1\times$, $2\times$, or vane pass frequency especially when the pump is operated at variable speed. Therefore, a torsional natural frequency analysis should be performed for all nonclog pump applications.

In addition to the pump rotor/bearing tower housing, the motor/discharge volute attachment stiffness, and the stiffness of the pump pedestal "milk stool" (preferably cross-braced in a truss fashion), installation specific characteristics for the foundation and attachment of the piping play a large role in determining the installed natural frequency, as does the effect of entrained water.

Motor stand design heavily influences the installed structural natural frequencies and can be used to shift them to more suitable locations. Changes in wall thickness, rib design, opening size, and mounting plate thicknesses have all been used to shift natural frequencies. In extreme cases running short struts from the foundation to the top of the motor stand can be used to significantly raise natural frequencies.

Natural frequencies for the vertical shafting for these units should be placed to avoid excitations by running speed and other sources by at least 20%. Excitations vary with the coupling used and the shaft location. Universal joint couplings produce $2\times$ and disc-pack couplings produce an excitation at a frequency equal to half the number of disc-pack bolts times rpm. Vane pass excitation can be transferred up the shafting and is a source of excitation for at least the first section of shafting attached to pump and may influence sections above this. It may not be possible to avoid excitation of all natural frequencies, in which case a forced response analysis may be required to ensure that shaft stresses and bearing loads are acceptable.

Motors A structural analysis requires the motor mass, center of gravity, and first reed critical frequency along with the motor mounting characteristics consisting of bolt circle diameter, number of bolts, bolt location, and OD, ID, and thickness of the motor mounting plate. A rotor torsional analysis requires the motor rotor polar inertia and torsional stiffness from rotor center of gravity to the coupling end of the shaft. A rotor lateral analysis (seldom required since motors are designed with their first critical speed at least 25%

above running speed) requires a motor shaft drawing noting diameters and lengths, bearing locations, bearing types, and mass amounts and locations, and polar inertias.

The authors have tracked motor data over the years and present typical data for evaluating motor data validity and for use in feasibility studies. As can be seen in Figure 13, motor data vary with frame and manufacturer, and actual motor data provided by the original equipment manufacturer (OEM) should always be used to minimize calculation error. The plots were developed from data for 72 motors used mostly in vertical applications. Eleven different vendors are represented with motors ranging from 15 to 6000 bhp (11 to 4500 kW) and 390 to 1200 rpm. Two synchronous motors, five horizontal motors, and three hollow shaft motors are represented. The remaining motors are vertical solid shaft induction motors. The data are plotted against torque recognizing that motors are torque producing machines and their data are best presented in this fashion.

The motor structural "reed" critical frequency data plot shows that the motors when mounted to a rigid structure have structural natural frequencies between 80 to 550% of running speed. The wide range in natural frequencies is mainly due to different construction techniques among vendors and the availability of different motor frame sizes. The plot shows that the average natural frequency decreases as a percent of running speed with increasing motor torque and corresponding motor size increase. It must be remembered that the data supplied by the motor vendor are almost always referenced to mounting the motor to a rigid structure. VTP discharge heads and frame mounted pumps cannot be considered "rigid" and therefore the data presented by the motor vendor should be considered the maximum theoretical natural frequency possible for that motor. Experience has shown that mounting the motor on top of a pump structure, whether it be a VTP or frame mounted unit, produces a motor/pump/pedestal system natural frequency approximately 50 to 80% of the motor/rigid-structure natural frequency. Therefore, the average motor mounted to the average pump produces an installed system natural frequency within +/- a factor of 2 of running speed, easily excitable by $1\times$, $2\times$, or vane pass frequency in VFD applications.

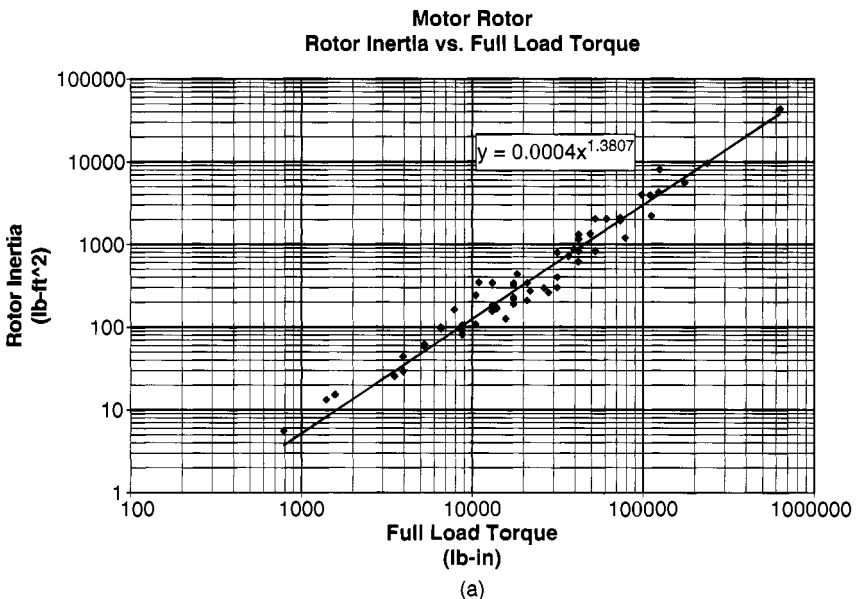


FIGURE 13A Typical motor inertia characteristics ($\text{kg-m}^2 = \text{lb-ft}^2/23.73$; $\text{N-m} = \text{lb-in}/8.851$) (Mechanical Solutions, Inc.)

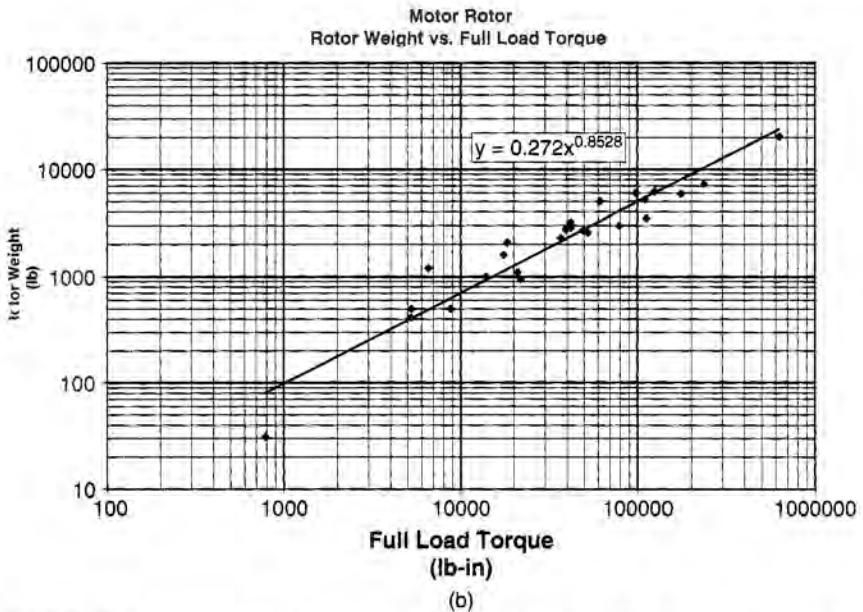


FIGURE 13B Typical motor weight characteristics ($\text{kg} = \text{lb}/2.205$; $\text{N-m} = \text{lb-in}/8.851$) (Mechanical Solutions, Inc.)

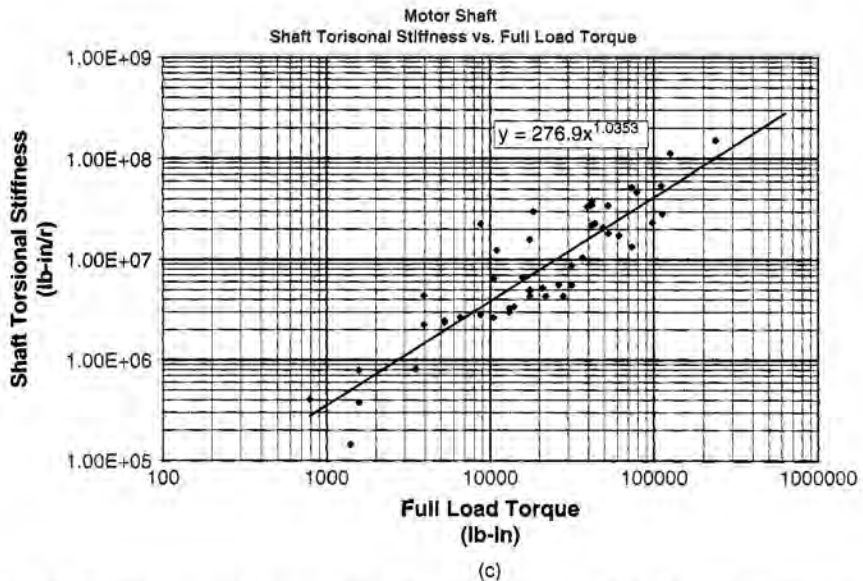


FIGURE 13C Typical motor shaft torsional stiffness characteristics ($\text{N-m} = \text{lb-in}/8.851$) (Mechanical Solutions, Inc.)

VFDs In variable speed installations it may not be possible to avoid exciting a natural frequency within the operating range. In this case the designer must place the natural frequency in a location (within the frequency domain) that will do the least harm recognizing the accuracy limitation of the natural frequency calculation method used. For example, assume a pump can be excited by $1\times$ (imbalance), $2\times$ (misalignment), and $3\times$ (vane pass) and the operating range is from 540 to 900 rpm; 60 to 100% speed @ 900 rpm typical for a VFD. Further assume that the specification has called for avoiding natural frequencies by $\pm 25\%$. The following results:

Excitation Frequencies, cpm						
Speed Range rpm	Nmin – 25%	Nmax + 25%	$2 \times$ Nmin – 25%	$2 \times$ Nmax + 25%	$3 \times$ Nmin – 25%	$3 \times$ Nmax + 25%
540 900	432	1125	864	2250	1296	3375

In this representative example, no safe haven for natural frequencies exists between 432 and 3375 cpm and the designer must choose a location to place the natural frequencies where they will do the least harm. The question becomes where does least harm exist, based on the nature of each excitation force. In this case all three excitation forces are proportional to the square of the speed. A potentially dangerous excitation force at full speed may have insufficient energy to excite excessive motion at reduced speed. Additionally, in this example $2\times$ misalignment most likely will produce the least force and the unit can be field aligned to minimize this harmonic. In this case the best location for natural frequencies is in the lower $2\times$ excitation range.

THE ROLE OF THE FLUID SYSTEM

Vibration versus Operating Point Ideally, a pump should operate close to its best efficiency point (BEP). Contrary to intuition, centrifugal pumps do not undergo less nozzle loading and vibration as they are throttled back, unless the throttling is accomplished by variable speed operation. Operation well below the BEP at any given speed, just like operation well above that point, causes a mismatch in flow incidence angles in the impeller vanes or in the diffuser vanes or volute tongues of the various stages. This loads up the vanes, and may even lead to “airfoil stalling,” with associated formation of strong vortices (miniature tornadoes) that can severely shake the entire rotor system, and can even lead to fatigue of impeller shrouds or diffuser plates or “strong-backs.” Such internal flow recirculation can cause cavitation on the *pressure side* of vanes, as opposed to cavitation on the nonpressure side which is caused by lack of sufficient NPSHR. The rotor impeller steady side-loads and shaking that occurs at flows below the onset of suction or discharge recirculation (see Sections 2.1.1 and 2.1.3) leads to the strong possibility of rubs, and excessive rotor loads that can damage bearings. Many plants buy equipment that has more capacity than is needed, to allow for future production expansion, but in doing so may ensure years of unreliable performance of potentially reliable machinery.

The typical effect on vibration of this practice is shown in Figure 14. Never run a pump for extended periods at flow rates below the “minimum continuous flow” provided by the manufacturer. Also, if this flow was specified prior to about 1985, it may be based only on avoidance of flashing and not on recirculation onset, and should be rechecked with the manufacturer.

A strong sign that recirculation is occurring is a “throbbing” vibration at part load, and strong vibration peaks showing up subsynchronously (i.e., below running speed), but not in the range of 0.4 to $0.5\times$ running speed. The 0.4 to $0.5\times$ range implies rotordynamic instability problems sourced in the bearings and annular seals (wear rings or balancing device), as discussed later.

Vibration & Pulsation vs. Flow

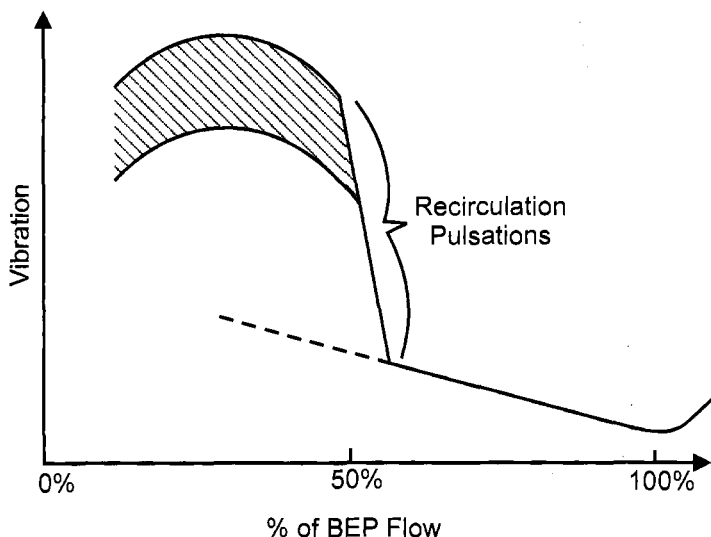


FIGURE 14 Effect on vibration of off-BEP operation (Mechanical Solutions, Inc.)

When stall vortices form, they often spit out the inlet to the impeller, and can cause “rotating stall” where the vortex formation is regular, and proceeds backward through the impeller as it spins forward. This leads to a net forward spin at some percentage of the rotational speed, typically 65 to 95% of rotational speed. This kind of stalling can also happen in a pump diffuser, but in that case typically rotates forward in the diffuser, at a rate of 5 to 35% of rotor speed. When these unexpected frequencies show up in the vibration spectrum, recirculation and rotating stall are often the cause. If such frequencies become equal to a critical speed, excessive vibrations at that frequency could result.

Cavitation Effects on Vibration The main issue of hydraulic concern is that sufficient suction static pressure be present to avoid cavitation. Today, it is understood that this means more than merely having sufficient net positive suction head available (NPSHA) to satisfy the 3% head drop NPSHR (NPSH Required) published by the manufacturer. (See Sections 2.1.1 and 2.1.3.) At NPSHA as much as $3\times$ the NPSHR, high frequency cavitation (sometimes inaudible) can cause serious erosion of the suction side of impeller vanes or wear ring exits. Most cavitation vibration and noise energy is in the 10 to 50 kHz range, and some energy all the way up to 120 kHz. Because cavitation excites a broad range of frequencies at once, as do other phenomena like bearing faults or flow passage turbulence, its presence and its damage potential are best determined with high speed oscilloscopes rather than with FFT analyzers. When significant cavitation is taking place, the shock wave of the bubble collapse shows up strongly (about 100 psi or more p-p) in the signal spike from a high frequency pressure transducer directly exposed to the flow, preferably with line-of-sight to the impeller. If the cavitation is damaging, bubble collapse is at a vane or wall, and the spike shows up as very strong acceleration (typically greater than 100 g’s p-p) on an accelerometer on the suction sidewall, and even (to a lesser degree) in other metal components such as bearing housings. The effect is hard to distinguish in the frequency spectrum, but shows up as spikes tens of microseconds long (with subsequent ringdowns) in an oscilloscope trace. g-levels in nondamaging cavitation situations are at least a factor of 10 less in spike strength.

A-Gap, B-Gap, and Overlap Dr. Elemer Makay introduced these famous concepts, relevant particularly to double suction pumps and high energy pumps such as boiler feed pumps and illustrated in Section 2.1.1. A-Gap is the radial annular width of the passage between the impeller OD and the volute or diffuser sidewall, and “Overlap” is the width (in the direction of leakage flow) of this close clearance area. Small A-Gaps and large overlaps decrease the communication between discharge pulsations and the impeller side passages, limiting the sensitivity of vibration to discharge recirculation, or discharge piping acoustic resonances. B-Gap is the radial distance from the impeller vane tip at the OD to start of the volute tongue or diffuser vane tip. If B-Gap is smaller than 3 to 5% of the impeller radius in diffuser pumps, or 5 to 8% of the impeller radius in volute pumps, then vane pass pulsations are often (not always) much stronger, possibly enough to cause vibration problems or even impeller hub or shroud fatigue cracking.

Hydraulic Instabilities Caused by Flat Pump Curves and Parallel Operation If the pump curve is flat and has a more positive slope than the system resistance curve, particularly when two or more pumps are run in parallel, the opportunity exists for the pump to get “confused” about what flow rate the system requires for a given head. This can lead to large, unstable flow surges. In a different but associated situation, if one of a pair of pumps deteriorates more than its mate when run in parallel, the stronger pump will set the head, and the weaker pump with a flat or positive head curve could be forced to run below minimum continuous flow or even at shutoff, although overall system flow remains high enough to satisfy the end user. This may lead to excessive vibration, and likely reliability problems in the weaker pump.

Thrust Reversals—Reasons and Ramifications Thrust reversals can occur in vertical pumps during start-up. If such upthrusting is greater than the weight of the motor and pump rotor system; uplift can occur, possibly damaging the mechanical seal or thrust bearing components not designed for upthrust. In multistage horizontal pumps, thrust reversal can happen if the flow rate becomes low enough so that flashing occurs, if the pump is run partially air-bound, or if the flow rate is outside the expected flow range so that the balance device overcompensates toward the suction.

Sources and Cures for Vane Pass Vibration Insufficient B-Gap can cause excessive vane pass vibration, in which case B-Gap should be increased under the supervision of a hydraulic designer. Suction or discharge recirculation can also lead to increased vane pass vibration, but usually this is less of a practical problem than the associated cavitation damage. Viable remedies are not to run at such low flows, to modify the impeller design to accommodate a lower flow rate (usually at some expense in efficiency), to put inlet guide vanes in front of the impeller to induce negative swirl, or to put “bulkhead rings” (sometimes called “choke rings,” which resemble large washers with ID a little less than the impeller inlet eye diameter) in front of the impeller to reduce the tendency for fluid to recirculate or flow upstream out of the impeller and to increase flow velocity and thereby improve incidence angle.

Piping and Acoustics

- 1. Check valve transients** The sudden closure of a check valve can cause a “thermal-hydraulic fluid transient,” or waterhammer (see Section 11.3). Large pressure pulsations can result, and can travel upstream through the pump, as well as through the column piping in a VTP. The authors have witnessed 300 psi (2 MPa) pressure pulses due to waterhammer.
- 2. Piping elbows and sump anomalies** Sump problems can result in unexpected flow swirl, even if air-sucking vortices are not visible. (See Chap. 13.) Sump vortices can form starting on the floor or walls of the sump, and can have cavitation or air (which comes out of solution in the submerged water) at the center, as illustrated in Section 13.2. Whether or not the vapor is at the center, the swirling flow can cause bad incidence angles at the impeller inlet, leading to unexpected suction recirculation, and resulting cavitation noise and excessive vibration. The remedy is to change the sump configuration

(e.g., simply adding an inlet cone or changing the suction bell setting), or to put straightening vanes and possibly a vapor trap in the inlet piping.

- 3. Piping acoustics** Acoustic resonances are familiar in the wind type of musical instruments. Acoustics of pump piping systems can be complex to analyze but for the most part can be broken down into two common situations, the quarter wave and the half wave. Quarter waves occur when the sound wave within a section of pipe is reflective at one end and open at the other. A good example of a quarter wave is a section of pipe coming from a sump, and attached at the other end to the suction inlet of a pump. A half wave occurs when the sound wave is either reflective at both ends or open at both ends, such as in a discharge header. Instances of half waves are less common than quarter waves. To calculate the acoustic natural frequency within a pipe it is necessary to know the speed of sound within the fluid, which is influenced by the wall thickness of the piping in which the fluid is enclosed. For practical purposes the speed of sound in water in steel piping can be estimated at about 4000 ft/sec (1200 m/s). The first fundamental frequency of a quarter wave is calculated as the fluid speed of sound divided by 4 times the length in feet (m) of the pipe section in question, while the half wave is calculated similarly, but the 4 is replaced with a 2.

It is important that an acoustic natural frequency not coincide with a source of excitation such as vane pass frequency. If this occurs, piping and headers may shake severely; and it is then necessary to detune the resonance, changing either the affected piping length or the impeller vane number. If neither of those is possible, in constant speed systems acoustic resonances can be diminished by installing a side-pipe "Helmholtz resonator" on the problem pipe wherein the pressure wave becomes strongest, and tuning its quarter wave frequency to the problem acoustic frequency.⁷

- 4. Allowable nozzle loads** Pumps are not normally designed to resist the very large force that the piping can impose due to thermal growth or pressurized "bourdon tube" type loading, and this load transfer can overstress the pump or baseplate. The resulting strain can move the pump into misalignment with the motor and can also create rubs in close running clearances. This is why pump manufacturers specify the maximum loading that may be applied to the pump nozzles.

Expansion joints can be used to isolate the pump from the piping, but these create separate concerns since without stiff axial attachment of the piping, the pump nozzle pressure times the nozzle area creates a longitudinal hydraulic thrust that otherwise could easily be dissipated by the piping wall. Therefore the joint should be fastened axially, using an appropriate tie rod system (including mounting plate) capable of carrying the load imposed while not overly constraining the pump to follow any piping movement in the lateral or torsional directions. The potential nozzle longitudinal thrust consists of the pressure force and the fluid momentum at the joint both of which can be substantial for a large pipe; for example, a 42-in (1067-mm) pipe transporting 30,000 gpm (6814 m³/h) of water at 10 psi (69 kPa) produces 13,854 lbf (61,626 N) due to the pressure and 901 lbf (4008 N) due to momentum. Lack of tie rods or insufficiently sized tie rods will allow transfer of part or all of the load back to the pump and create at the pump flange a pressure pulsation force at vane pass frequency. This pressure pulsation can create unacceptable vibration at vane pass frequency in an otherwise acceptable installation.

ROTORDYNAMICS

Computer Methods Rotordynamic analysis is typically performed by a specialized computer program, able to account for gyroscopics and cross-coupled forces. Older programs use the more computationally efficient Transfer Matrix method, although the massive increase in rapid and inexpensive computational capability has led to conversion to the use of the FEA method, due to its flexibility and its ability to handle more complex models. Traditionally, even using FEA the rotor model is a "stick figure," since greater detail

is typically not required for sufficient accuracy. In such simplified models, the following assumptions are made:

1. Linear bearing coefficients, which stay constant with deflection. This can result in significant errors for large rotor orbits.
2. Linear, monolithic bearing supports.
3. Perfect impeller and sleeve fits, except as accounted for as a worst-case unbalance.
4. If flexible couplings are used, shaft coupling coefficients are negligible in the radial deflection and shaft bending modes, with finite stiffness only in torsion.
5. It is assumed there is no feedback between vibration and resulting response forces, except during stability analysis.

Gyroscopic Stabilization Gyroscopic forces are important, and can either effectively stiffen or de-stiffen a rotor system. The key factor is the ratio of polar moment of inertia " I_p ," the second mass moment taken about the rotor axis, to transverse moment of inertia " I_t ," taken about one of the two axes perpendicular to the rotor axis, but passing through the center of mass. This ratio is multiplied by the ratio of the running speed divided by the orbit or "whirl" speed. The whirl speed is the rate of precession of the rotor, which can be "forward" (in the same direction as running speed) or "retrograde" or "backward" (opposite in direction to running speed). The whirl or precessional speed absolute value is generally less than the running speed. It is very difficult to excite backward whirl in turbomachinery, so evaluation is limited to the forward whirl modes. If the product of the inertia and speed ratio is less than 1.0, then the gyroscopic moment is de-stiffening relative to forward whirl, while if it is greater than 1.0, it tends to enhance the rotor spinning about its center axis (i.e., the principle of a gyroscope) and thus contributes apparent stiffness to the rotor system, raising its forward whirl natural frequencies.

Couplings Couplings may provide either a rigid or a pivoting ball-in-joint type connection between the pump and its driver. These are known as "rigid" and "flexible" couplings, respectively. Rigid couplings firmly bolt the driver and driven shafts together, so that the only flexibility between the two is in the metal bending flexure of the coupling shank, acting as a beam. This type of coupling is common in vertical and in small end-suction horizontal pumps. In larger horizontal pumps, especially multistage or high-speed pumps, flexible couplings are essential because they prevent the occurrence of strong moments at the coupling due to angular misalignment. Common types of flexible couplings include gear couplings and disc-pack couplings. Both gear and disc couplings allow the connected shafts to kink, but allow torsional transmission similar to rigid couplings.

In performing a rotordynamics analysis of a rigidly coupled pump and driver, the entire rotor—pump, coupling, and driver—must be analyzed together as a system. In such a model, the coupling is just one more segment of the rotor, with a certain beam stiffness and mass. In a flexibly coupled pump and driver, however, the entire rotor train usually does not need to be analyzed. Instead, the coupling mass can be divided in half, with half added to the pump shaft model, and the other half and the driver shaft ignored in the analysis.

Fluid "Added Mass" The fluid surrounding the rotor adds inertia to the rotor in three ways: the fluid trapped in the impeller passages adds mass directly, the fluid displaced by the presence of the impellers and shaft material adds its mass directly due to the rotor system, since as the rotor vibrates within the fluid it must displace this mass, and the fluid in close clearances must accelerate much faster than the rotor vibration acceleration to preserve continuity, and hence potentially adds mass many times its displaced mass (the so-called Stokes effect).

Impeller Forces As an impeller moves within the surrounding diffuser or volute, reaction forces are set up because of the resulting nonsymmetrical static pressure distribution around the periphery of the impeller. These forces are normally represented by coefficients which are linear with displacement. The primary reaction forces are a negative direct stiffness, and a relatively large cross-coupling stiffness. Both of these forces

tend to be destabilizing in situations where stability is an issue. Along with reactive forces, there are also active forces which exist independently of the impeller motion and are not affected substantially by it. These forces are primary “excitation forces” for the vibration.

Annular Seal Lomakin Effect Annular seals in pumps can greatly affect dynamics by changing the rotor support stiffness and therefore the rotor natural frequencies.³ The stiffness and damping of an annular seal is provided in small part by the squeeze-film and hydrodynamic wedge effects well known to journal bearing designers. However, because of the high ratio of axial to circumferential flow rates in annular liquid seals relative to bearings, large forces can develop in the annular clearance space due to the circumferentially varying Bernoulli pressure drop induced as rotor eccentricity develops. This is known as the Lomakin effect as discussed earlier, and is the largest stiffness and damping force generating mechanism within pump annular seals.

The primary action of the Lomakin effect is beneficial, through increased system direct stiffness and damping which tend to increase the rotor natural frequency and decrease the rotor vibration response at that natural frequency. However, overreliance on the Lomakin effect can put the rotor design in the position of being too sensitive to wear of operating clearances, resulting in unexpected rotor failures due to resonance. It is important that modern rotors be designed with sufficiently stiff shafts so that any natural frequency which starts above running speed with new clearances remains above running speed with clearances worn to the point that they must be replaced from a performance standpoint.

Nonlinear Vibration: Parametric Resonance and Fractional Frequencies Certain types of nonlinear vibration response have been found common in turbomachinery. They generally fall into the category of parametric resonance, the details of which are beyond the scope of this section. Typically, parametric resonances are caused by bearing support looseness or a rub at a bearing, seal or other running clearance, allowing a contact/loss-of-contact vibration excitation similar to dribbling of a basketball. The symptoms are a pulsating orbit, with a large amount of vibration at exact whole fractions of running speed, such as $\frac{1}{2}$, $\frac{1}{4}$, and so on.

Cross-Coupling versus Damping and “Log Dec” Cross-coupled stiffness originates from the way fluid films build up hydrodynamically in bearings and other close running clearances, as discussed earlier. The cross-coupling force vector acts in a direction directly opposite to the vector from fluid damping, and many think of it in terms of an effectively negative damping. The action of cross-coupling is very important to stability, in that if the cross-coupling force vector becomes greater than the damping vector, vibration causes reaction forces that lead to ever more vibration, in a feedback fashion.

Subsynchronous Whirl, Whip, and Rotordynamic Stability Rotordynamic stability refers to phenomena whereby the rotor and its system of reactive support forces are able to become self-excited, leading to potentially catastrophic vibration levels even if the active, stable excitation forces are quite low. The most common form of rotordynamic instability is known as shaft “whirl,” which at lower speeds begins as a more benign phenomenon known as shaft “part-speed whirl.” Shaft part-speed whirl is a forced response at a frequency below the first shaft bending natural frequency. It is driven by a fluid excitation force, such that the static pressure field which causes the force is rotating at some speed below running speed. The fluid rotational speed becomes the whirl speed. The most common cause of whirl is fluid rotation around the impeller front or back shrouds, in the wear rings, or in journal bearing clearances. Such fluid rotation is typically about 45% of running speed, because the fluid is stationary at the stator wall, and rotating at the rotor velocity at the rotor surface, such that the bulk of the fluid rotates at roughly half speed in the running clearance. The pressure distribution which drives this whirl is generally skewed such that the cross-coupled component is potentially strong and positive in sign, able to “pump” energy into the rotor whirling motion. If somehow clearance is decreased on one side of the gap, due to eccentricity for example, the resulting cross-coupled force increases further.

The cross-coupled force acts perpendicular to any clearance closure. In other words, the cross-coupling force acts in the direction that the whirling shaft minimum clearance will be in another 90 degrees of rotation. If the roughly half speed frequency at which the cross-coupled force and minimum clearance are whirling becomes equal to a natural frequency, a 90 degree phase lag shift occurs, which is typical when force frequency equals natural frequency. This phase shift means that the motion in response to the cross-coupling force is delayed from acting for 90 degrees worth of rotation. By the time it acts, therefore, the gap which caused the force has rotated 90 degrees such that the cross-coupled force tends to act in a direction to further close this minimum gap. As the gap closes in response, the cross-coupled force which is inversely proportional to this gap increases further. The cycle continues until all gap is used up, and the rotor is severely rubbing. This final process is shaft whip, and is a dynamic instability in the sense that the process is self-excited once it begins.

The characteristics of shaft whip are that once it starts, all self-excitation occurs at the bending natural frequency of the shaft, so the vibration response frequency "locks on" to the natural frequency. Since whip begins when whirl is close to half the running speed, and is equal to the shaft natural frequency, the normal $1 \times$ running speed frequency spectrum and roughly circular shaft orbit now show a strong component at about 45% of running speed, which in the orbit shows up as a loop reflecting orbit pulsation every other revolution. The "lock on" of vibration to the natural frequency means that vibration at speeds above whip initiation will no longer track the whirl's previous constant percentage of running speed.

One method of overcoming rotordynamic instability is to reduce the cross-coupling force which drives it. A complementary solution is to increase system damping to the point that the damping vector, which acts exactly opposite to the direction of the cross-coupling vector, overcomes the cross-coupling. The amount of damping required to do this is commonly measured in terms of "log dec," roughly $2 \times \pi \times$ damping ratio. For centrifugal pumps, it has been found that if the log dec is calculated to be greater than about 0.15 then it is likely to overcome any cross-coupling forces which are present, avoiding instability.

Typical design modifications which reduce the tendency to rotor dynamic instability involve bearing changes, to reduce cross-coupling and hopefully simultaneously increase damping. The most unfavorable type of bearing in this regard is the plain journal bearing, which has very high cross-coupling. Other bearing concepts, with elliptical or offset bores, fixed pads, or tilting pads, tend to reduce cross-coupling, dramatically so in terms of the axially grooved and tilting pad style bearings. Even more effective and controllable, but much more expensive, are the hydrostatic bearing and the actively controlled magnetic bearing. Another bearing effective in reducing cross-coupling relative to damping is the pressure dam bearing.

Shaft Press Fits API publishes a rotor dynamic guideline standard designated API-684.⁴ This guideline discusses the standard practices which API uses in establishing the vibration standards for its various machinery specifications such as API-610 for centrifugal pumps.² In API-684, the effects are discussed coming from shaft fits, such as occur in slip-fit or press-fit impellers and sleeves. API's opinion is that the effect of such fits is uncertain and generally not large, since asperities and ovality allow slight shaft bending within even a heavily press-fit sleeve or other component. Therefore, for consistency of analysis, API recommends that such fits be ignored, and that any slip or press-fit components be accounted for only in terms of their added mass, not in terms of any stiffness contributions they make to the rotor system. It is the authors' experience that press-fits do increase shaft critical speeds, as determined by test, but generally by less than half the amount of increase expected if the full added stiffness of the press-fit component were added to the rotor model.

Forced Response and Typical Excitation Levels A forced response analysis may be used to predict the vibrational displacement and stress at specific points in a model due to individual excitations. The analysis can be used to demonstrate compliance with vibration requirements when it is not possible to avoid the occurrence of natural frequencies

within the range of excitation. FEA is the preferred calculation method and the analysis proceeds by first calculating the natural frequencies and mode shapes, and selecting a representative modal damping value. The forced response analysis is then performed. The analyst must enter modal damping ratios, force inputs, and calculation output locations. Modal damping ratios and excitation forces can be difficult to quantify. If modal test data are available, the damping can be estimated from existing vibration spectra by measuring the width of the peaks at half power, such as discussed in API-541.¹⁰ Alternatively, for most pump structures modal damping of 1 to 3% critical damping has proven to be representative. Forces are input at specific locations and response to the individual force is measured at specific locations across the frequency range. The result is a plot of amplitude versus frequency for a specific force at a specific location. Since most loading varies with speed, output from the forced response analysis is often calculated using unit loads, and is post processed to determine amplitude. In this way not only can force be varied with speed, but responses to individual forces at a specific point can be combined using the square root of the sum of the squares (SRSS).

Effects of Bearing Housings and Casings The output in the example presented in Figure 15 shows the response at the top of the bearing housing for a vertically mounted sewage pump, due to a 100-lbf (445-N) unit load excitation at the impeller as well as at the line shaft couplings. Figure 16 shows the expected response at the top of the bearing housing due to $1\times$, $2\times$, $3\times$, and vane pass excitations determined by post processing the data used in Figure 15. The response for each excitation represents the SRSS combination of individual forces at a particular frequency, and considers how those forces vary in magnitude with speed. For $1\times$ the vibration frequency and running frequency coincide. For all other responses the vibration frequency is a multiple of the running frequency; for example, for $2\times$ the response amplitude when running at 10 Hz occurs at 20 Hz. For this example, imbalance force was estimated using the individual component ISO balance criteria limit, and misalignment was assumed equal to imbalance. Vane pass force was estimated by reviewing field vibration response spectra and, based on this, was set at 40% of the impeller radial load divided by the number of vanes.

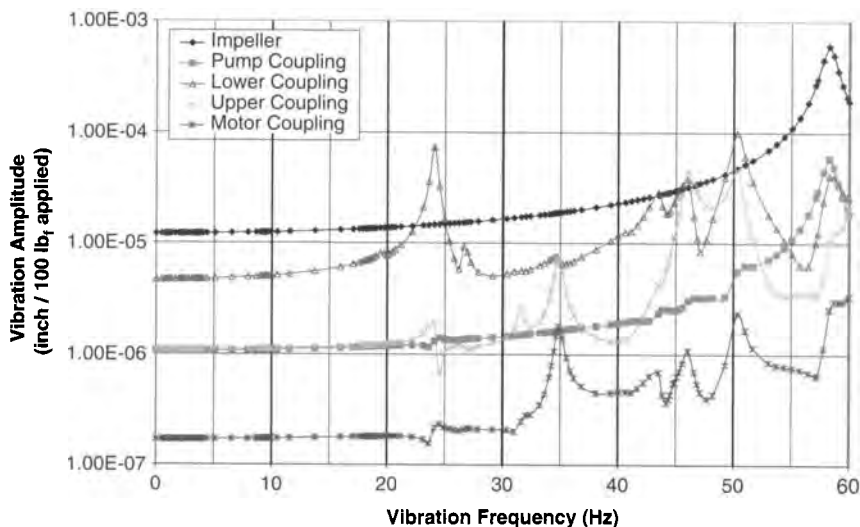


FIGURE 15 Horizontal vibration vs. frequency on pump upper bearing housing due to 100 lbf (445 N) applied separately at five excitation locations (mm = inch $\times$ 25.4) (Mechanical Solutions, Inc.)

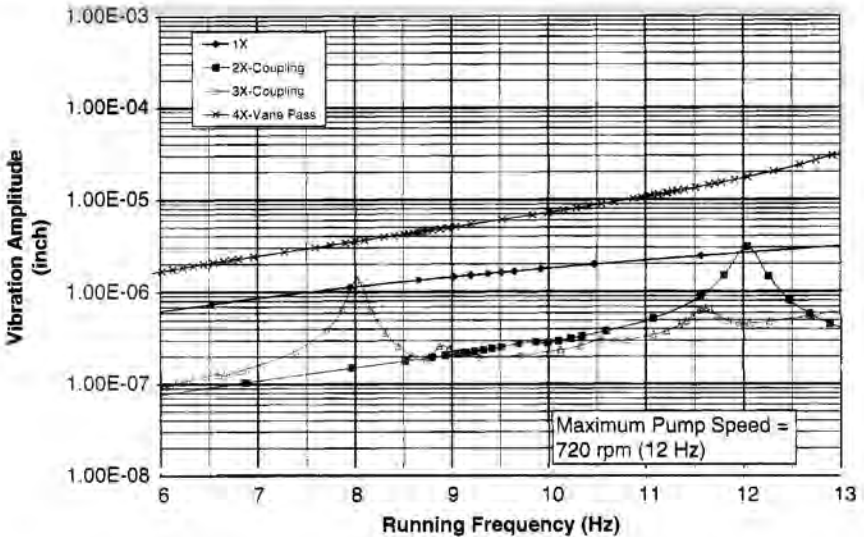


FIGURE 18 Horizontal vibration vs. speed (pump rpm) on pump upper bearing housing due to the combined effect of all excitations (mm = inch $\times$ 25.4) (Mechanical Solutions, Inc.)

TORSIONAL VIBRATION

It is important to predict torsional natural frequencies near the running speed of a drive train or at other important excitation frequency sources such as vane pass frequency, $1\times$ and $2\times$ electric motor line frequency, and motor rotating speed times number of poles. Although torsional vibration problems are not common in pump systems, they occur enough to require calculation. In this kind of vibration, the shaft rotates at different instantaneous speed and different phase (superimposed on the steady running speed), creating a torque oscillation along the shaft and creating twisting motion that can generate high destructive stresses. Accelerometers or proximity probes commonly used to measure lateral vibration will not usually detect this vibration. A severe torsional vibration can exhibit gear noise and tooth distress, coupling wear, keyway fatigue, or motor winding loosening.

Lateral rotor dynamics of pumps can typically be analyzed without including other rotor system components such as the driver or gear box. However, torsional vibration of the pump shaft and all types of vibration of the pump stationary structure are system-dependent, because the vibration natural frequencies and mode shapes will change depending on the mass, stiffness, and damping of rotor components other than those included within the pump itself. The pump assembly must be modeled in terms of at least three bodies: the pump rotor, the coupling hubs (including any spacer), and the driver rotor. If a flexible coupling (e.g., a disc coupling) is used, the coupling stiffness must be included in the analysis. Good estimates of coupling torsional stiffness, often as a function of speed and torque, are listed in the coupling catalog data. Often a range of stiffness for a given size is available, if needed to tune a torsional critical speed to a new value. If a gear box is involved, each gear must be separately accounted for in terms of both inertia and gear ratio. If the pump or driver rotor is not at least ten times as stiff torsionally than the shaft connecting the rotor to the coupling, then the individual shaft lengths and internal impellers should be included in the model. If resonances cannot be avoided, a coupling selection can be optimized based on shaft torsional stress calculations. When the coupling

is changed usually it results in different weights as well, affecting the lateral critical speed, and rotordynamic reevaluation is recommended. A heavier coupling will reduce the lateral critical speed.

- 1. Mass elastic diagrams** For the first step in calculating torsional natural frequencies, the rotor system is divided into stations, and a mass-elastic diagram is used to indicate geometric information about each station of the pump/driver rotor system, in terms of rotary inertia and torsional stiffness distribution. All of this information is used as input for the computer model of the rotary system, as illustrated in Figure 17.
- 2. Calculation techniques** Methods for calculating the first several torsional natural frequencies manually are given in Blevins for simple models.⁷ Numerical torsional analysis can be performed by using general purpose FEA programs, or specialty programs based on the Holzer method or transfer matrix method (TMM). In all of the computer-based methods, the system of differential matrix equations including damping is:

$$[J]\ddot{\theta} + [C]\dot{\theta} + [K]\theta = 0$$

where

J = diagonal rotary inertia matrix

C = rotary damping matrix

K = torsional stiffness matrix

$\ddot{\theta}$ = angular acceleration vector, rad/sec²

$\dot{\theta}$ = angular velocity vector, rad/sec

θ = angular displacement vector, rad

- 3. Penetration factors** Many rotating machinery components are not integrally machined on the shaft. Thrust collars, sleeves, disks, impellers, and coupling hubs are assembled using shrink fits on the shaft. Depending on the component length, the amount of shrink fit, and the size, such components may or may not contribute significantly to the torsional stiffness. However, API-684 applies the following principles:⁴
 - If the fit of the shrunk-on component is relieved over most of its length, then the torsional stiffening effect is negligible.
 - If a key is used to transfer torque (i.e., in a coupling hub), and has an axial length equal to $\frac{1}{3}$ of the shaft diameter, measured where the key starts, this will not be

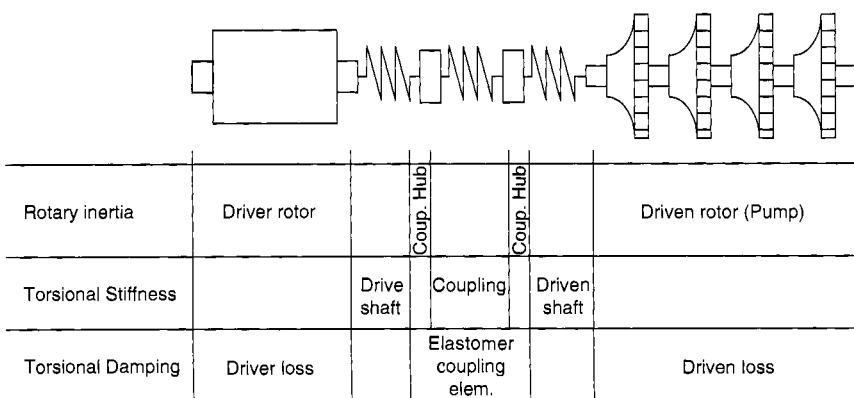


FIGURE 17 Torsional model example

considered an addition of torsional stiffness. If $\frac{1}{3}$ the shaft diameter is greater than the key length the hub is considered loose fit and does not add torsional stiffness.

- If the fit of a shrunk-on component is not relieved over a significant part of its length, made of the same material as the shaft and manufactured with a diametral shrink fit equal or greater than 1 mil/in ($\mu\text{m/mm}$) of shaft diameter, then the effective torsional stiffness should be assumed identical with the OD of the sleeve or component.
 - If the shrunk-on component has a large rotational inertia such that centrifugal loading diminishes the effect of the shrink fit to less than 1 mil/in ($\mu\text{m/mm}$) remaining press, then the torsional stiffening effect is considered negligible.
 - If a shrunk-on component has a nominal fit length with L/D greater or equal to 1, the shaft is assumed to be unrestrained by the hub over an axial length that is $\frac{1}{3}$ of the total length of shrunk surface. This $\frac{1}{3}$ penetration is typically used by coupling vendors when specifying coupling torsional stiffness from hub-to-hub.
- 4. Effects of the driver and gear train** For an electric motor, the most important torsional excitation source is the motor rotating speed times the number of poles. If a gear box is involved, each gear must be separately accounted for in terms of inertia, flexibility, backlash, and gear ratio. The gear stiffness can only be approximated, since there are several parameters varying during rotation. These include:
- Instantaneous point of contact
 - Direction of applied force
 - Number of teeth in contact

The gear mesh frequency (number of teeth times rotational speed for a given gear) is usually a strong torsional excitation frequency with a typical p.u. (per unit, i.e., oscillation amplitude peak-to-peak fraction of steady torque) factor of about 0.02.

- 5. Effects of the bearing, casing and foundation** Typically, the bearing, casing, and foundation of the pump and remainder of the drive train play a negligible role in the torsional dynamics of a pump rotor system.
- 6. VFD operation** The use of variable frequency drive (VFD) motors has resulted in significant efficiency improvement in plant operation, because the motor speed can be adjusted to optimize efficiency in the system. For pump systems that involve VFDs, however, special attention should be taken because they provide significant new excitations called "control pulse" frequencies at higher multiples of the simulated line frequency ($2 \times$ motor running speed/no. of motor poles), often at $6\times$, $12\times$, $18\times$, and so on. The manufacturer of the controller can predict these excitation frequencies and their associated per unit factors. The p.u. factors of VFDs are reduced considerably in pulse-width-modulation (PWM) based VFDs.
- 7. API-684** The API Standard 684, Rotordynamics and Balancing, includes recommendations for torsional analysis, and explains the importance of such analysis for new or modified designs in rotating systems that include untried combinations of impellers, coupling gears, shaft dimensions, and drivers. The API requirements are:⁴
- Performance of an undamped train torsional natural frequency analysis.
 - Inclusion of the complete drive train.
 - Verification of adequate separation margin between the torsional natural frequencies and significant excitation harmonics (minimum separation margin of $\pm 10\%$), where excitation frequencies include $1 \times N$, $1 \times$ and $2 \times$ electrical line frequency excitation in motors/generators, VFD control pulse frequencies at $6 \times$ multiples of electrical line frequency excitation, and gear mesh.
 - If any torsional interference can not be shifted, a stress and fatigue analysis should be performed to ensure that the interference will not adversely affect the integrity of the machine components.
 - A transient torsional vibration analysis is usually required if a synchronous motor driver is used.

Transient torsional response of the system should be calculated with respect to excitations during start-up, trip, and motor control changes. Both steady state as well as transient forced response should be in terms of the simultaneous stationary plus oscillating shear stress in the most highly stressed element of the drive train, usually the minimum shaft diameter. Judgment on the acceptability of the assembly's torsional vibration characteristics should be based on whether the forced response shaft stresses are below the fatigue limit by a sufficient factor of safety, at all operating conditions. The minimum recommended factor of safety is 2, for a carefully analyzed rotor system. To determine system stresses quantitatively, a good worst case estimate of the system damping and of the frequencies and magnitudes of all of the significant excitation forces is required. The primary torsional damping is from energy expended by the pump impellers when they operate at slightly higher and lower instantaneous rotating speeds due to the vibratory torsional motion. For acceptability of the torsional rotor system assembly, stress at any condition should be below the fatigue limit with at least safety factor of 2.0 including all stress concentration factors.

- 8. Damping and typical excitation levels (p.u. factors)** A reasonable estimate of the amount of torsional damping is expressed in the following equation:

$$C = k \frac{\tau}{N^2} f$$

where

C = damping, in $\cdot \text{lbf} \cdot \text{s/rad}$ ($\text{N} \cdot \text{m} \cdot \text{s/rad}$)

k = constant value 19.1

τ = rated torque in $\cdot \text{lbf}$ ($\text{N} \cdot \text{m}$)

N = rated speed, cycles/min

f = evaluated frequency, cycles/min

To determine the frequencies at which large values of vibratory excitation torque are expected, and the value of the torque occurring at each of these frequencies, the pump torque at any given speed and capacity can be multiplied by "per unit" factor "p.u." The p.u. factor at important frequencies can be obtained from pump, motor, and control manufacturers for a specific system, and it is typically in the range of 0.01 to 0.05 peak-to-peak for important excitations. Unsteady hydraulic torque from the pump is present at a frequency equal to the running speed times 1 as well as times the number of impeller vanes, with the higher values (0.05) being more typical of worst case off-BEP operation and/or impellers with less than four vanes. The most important torsional excitation frequencies from the motor are the number of poles times the motor slip frequency (for induction motors), rotating speed times the number of motor poles, and also the rotating speed itself. The gear mesh frequency is usually an important torsional excitation frequency with a typical p.u. factor of about 0.02.

FINITE ELEMENT ANALYSIS OF PUMP STRUCTURAL VIBRATION

The most common form of vertical pump is the vertical turbine pump, or VTP, which is very different from other pumps, because of its less stringent balancing, shaft straightness, and motor shaft alignment tolerances, and because of its long flexible casing and the casing's flexible attachment to ground, and because of the peculiar spaghetti-like line-shafting which connects the motor to the below-ground liquid-end "bowl assembly" of the pump. However, like other pumps, the bearing loads and the bearing and wear ring clearances are where problems are likely to occur.

Structural Vibration Analysis: Various Levels of Model Fidelity In general, for VTPs and vertically mounted sewage pumps, determination of the vibrations of the stationary

structure, the lineshafting, and the pump and motor rotors should be done simultaneously, using FEA. The goal of such analysis is to determine the vibration mode shapes and natural frequencies of the first roughly 10 vibration modes which involve significant motion of any of the following: any nonrotating portion of the pump, the frame of the closely coupled motor or housing of the variable speed drive (if so equipped), the pump baseplate, the pedestal, or the foundation within the geometrical limits of the model (as defined below). The components in such a model should be represented mathematically in considerable detail, as follows:

- Include foundation mass and stiffness within a radial distance (measured from the center of the pump base) at least equal to the height of the top of the motor relative to the level of attachment of the baseplate to the floor.
- Include piping details important to modal mass and stiffness, such as hangers, bulkheads, and expansion joints, and all the piping and its enclosed fluid within a spherical zone of radius (relative to the centerline) equal to twice the height of the top of the motor relative to the level of attachment of the baseplate to the floor.
- Include the effective flange-to-flange pump casing/bowl assembly stiffness, mass, and location of center-of-mass, of the close-coupled motor and variable speed drive (if so equipped), and of the discharge head or motor stand.
- Include any pedestal, discharge housing, and motor stand stiffness variations between the piping in-line and perpendicular directions, taking particularly into account the effects of coupling access or stuffing box access windows.
- Include the below-ground column piping and bowl assembly (in pump designs such as can pumps which incorporate these components), the fluid in and immediately around the column piping and bowl assembly,⁷ any column piping stiffeners or supports, and any shaft enclosure tubing.
- Include the mass of all pump impellers, and either include their nearby bowl bearings or assume they behave as if they are rigidly attached to the pump casing at their center of mass.

It should be noted that when analyzing VTPs or vertically mounted sewage pumps, the best policy is to model the entire pump, motor, and base as a system, and not only model the above-ground components. Important resonances have occurred in pumps involving below-ground activity in concert with above-ground motion. In general, there is no such thing as a strictly “above-ground mode” or “below-ground mode.” Although modes may be dominated primarily by above-ground motion, this cannot be counted on.

Combining Structural and Rotor Dynamic Preinstallation Analysis The flexibility of the VTP structure and shafting result in many closely spaced modes within the range of frequencies for which strong exciting forces are expected. An average of one mode per 100 cycles per minute (cpm) is not unusual for deepwell VTPs. VTP pumps also exhibit nonlinear shaft dynamics because of the large shaft excursions which occur in the lightly loaded long length/diameter ratio bearings, as will be explained below. Time-averaging modal methods can be used to experimentally determine the rotor nonlinear response.

An important element of VTP shaft vibrations is the strong effect of axial thrust on the impellers, causing a roughly 10% increase in shaft natural frequencies, as discussed by Kovats, and providing a restoring moment which tends to suppress lateral vibrations in a nonlinear fashion, as explained by Blevins.⁷ Another important factor is the statistical character of the support provided by any given lineshaft bearing. If the bearings behaved consistently and linearly, FEA could be used to accurately predict the lineshaft modes. However, the normally lightly loaded lineshaft bearings exhibit a rapid, nonlinear increase in bearing stiffness as the lineshaft gets close to the bearing wall. Given the flexibility of the lineshaft and the weak support provided by the pump column piping, and given the relatively large assembly tolerances and misalignments in the multiple lineshaft bearings

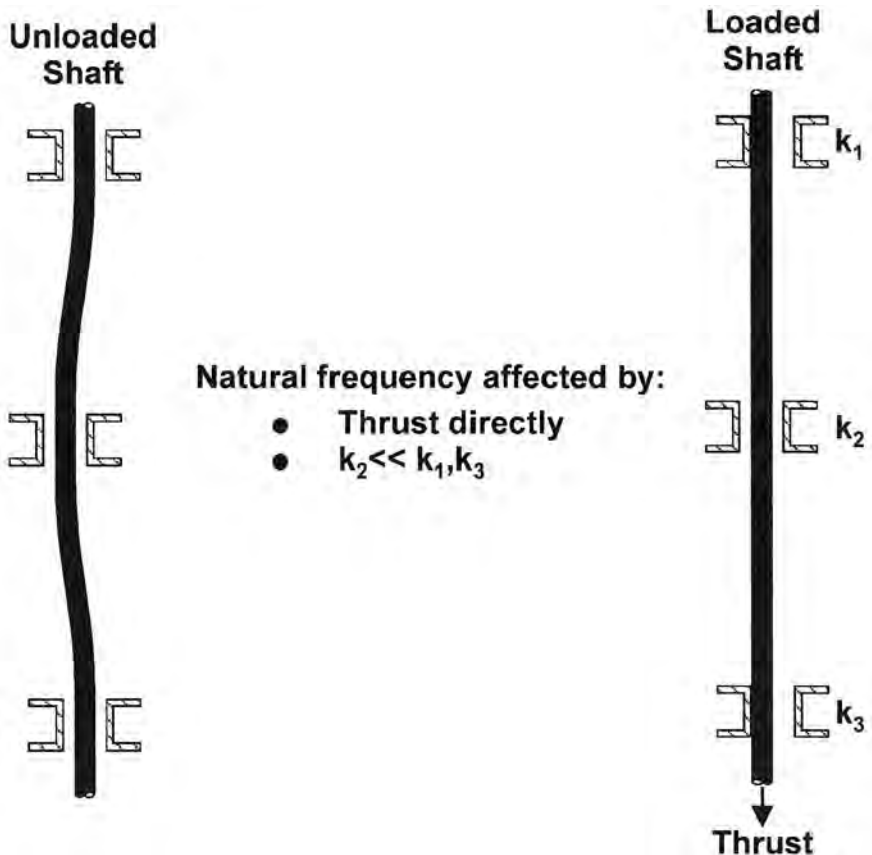


FIGURE 18 “Violin String” effects, influenced by axial thrust and bearing eccentricities (Mechanical Solutions, Inc.)

of these machines, the contribution of each bearing to the net rotordynamic stiffness is a nearly random and constantly changing situation, as explained conceptually in Figure 18. The result is that in practice there is no single value for each of the various theoretically predicted natural frequencies, but rather the natural frequencies of the lineshafting and shaft in the bowl assembly must be considered on a time-averaged and location-averaged basis.

If rotordynamic analysis is performed on VTPs, it should be performed using a validated computer program based on FEA, and the following issues need to be addressed:

- Include all rotating component masses and effective assembled flexibility for the entire pump rotor, the lineshafting, the motor coupling or drive, and the motor rotor. Also include effective added mass for fluid inside and around the impellers and lineshafting.
- Include the stiffness, both direct and cross-coupled, and damping of all bowl, lineshaft, drive, and motor bearings, and of all impeller annular seals (Lomakin effect). It is not generally possible to determine a precise value for these parameters, but a range of values can be identified, with maximum and minimum values established at representative operating points. Separate calculations for shaft natural frequencies and vibration

amplitudes should be performed for at least three situations: minimum stiffness at all bearings and seals, most probable stiffness at all bearings and seals, and maximum stiffness at all bearings and seals.

- Include the cantilevered stiffness of the motor bearing supports relative to the foundation, and of the bowl and lineshaft bearing supports as influenced primarily by the column piping flexibility.
- Include a forced response analysis and a transient stability analysis using as minimum input calculated mode shapes and natural frequencies for the first 10 rotor modes, and predictions of bearing and annular seal stiffness and damping which bracket expectations, as discussed above. In calculating forced response, include as minimum forces worst-case unbalance in each impeller and in the motor rotor and drive rotors or motor coupling halves, worst case misalignment across the drive or motor coupling, and worst case unsteady or rotating hydraulic forces on each impeller.
- Include torsional as well as lateral, axial, and mixed vibration modes in all analyses. If flexible couplings are used, a reasonable estimate must be made of the coupling torsional, lateral, and axial stiffnesses, which are usually listed as catalog data. It should be assumed that the thrust and radial bearings and annular seals provide no torsional constraint or stiffness.

If a pump is low in power, often analysis will be bypassed. This always entails some degree of risk, since small pumps can resonate just as easily as large pumps. It is the authors' experience that it is cost-effective in the long-run to analyze all pumps of 100 hp (75 kW) or greater in the manner described, prior to installation. In addition, the following should be considered "danger flags," increasing the need for detailed analysis:

- Vibration specifications requiring less than 0.20 ips peak.
- Particularly tall units (L/D greater than 4).
- Variable speed units.
- Pumps that tie into headers that look like flutes, because they may acoustically act like flutes.
- Flexibly supported pillow block bearings on U-joint drive shafting.
- All equipment mounted to a particularly flexible or loss inertia foundation (foundation mass less than $5 \times$ the weight of the total weight of the supported equipment).
- Insufficient tie rods, based on calculation of excessive deflection (more than 3 mm or 0.125 in) due to length versus cross-sectional area and worst-case applied load.
- Insufficient length of suction piping.
- Lack of pipe supports close to the pump, when piping is hard-coupled to the pump.

A useful simplified method of predicting lineshaft reliability with a worst-case model known as the "jump rope" model has been reported by Marscher.¹¹ The concept is to model the lineshaft vibratory motion and loads in the worst-case limit by the deflection and end support forces associated with a whirling jump rope, with the addition of axial thrust and bending stiffness effects. The deflection of such a jump rope may be calculated by a quasi-static analysis, based on D'Alembert's principle with the end conditions set equal to the radius of the circular path of the "hands" (bearing radial clearance) controlling the rope, and the load per unit length at each point along the rope equal to the local displacement, times the mass per unit length, times the square of the rotational frequency. The deflections predicted by this model are worst case, regardless of the value of or linearity of the bearing stiffness, if the circular orbit of the end conditions is set equal to the diametral clearance of the lineshaft bearings, and if the rotor deflection slope within each bearing is set equal to the bearing diametral clearance divided by the bearing length. The latter condition is the so-called "encastre" condition, studied by Downham, and Yamamoto.^{12, 13} It is the encastre condition which ultimately limits the shaft deflection and stresses, and the bearing loads; both by limiting the slope of the shaft, and by changing the end support

condition of a shaft length in the analysis from “simple” (i.e., knife edge) to fixed. Compared to the load caused by the whirling shaft mass in this condition, minimal bearing forces are caused by initial unbalance, misalignment, or bends in the shaft; which is why liberal tolerances on these pumps are commercially acceptable. For relatively stiff line-shafting such as in most reactor coolant pumps, the jump rope model gives answers which are too conservative to be useful, but for the majority of VTPs with shafting under 2.5 in diameter and bearing L/D ratios of 2.5 or more it gives a quick method of confirming that shaft stresses and bearing loads are acceptable.

FUNDAMENTALS OF PUMP VIBRATION TESTING

Predictive maintenance and troubleshooting of potential or suspected mechanical problems in pumps begins with the gathering of information which is used to determine the condition of the machine. The analysis of these data is commonly done through trending over time, which is a useful indicator of health. Supplementing such an approach, detailed instantaneous evaluation can diagnose specific problems through analysis of the spectral components of the vibration and, at the highest level, the creation of detailed animations which show the operating deflection shape (ODS) of the machinery during normal operation, and/or during start-up and shutdown transients. In addition to such operational testing and evaluation, experimental modal analysis is a specialized type of vibration testing in which the response of the machinery is measured in response to a known input excitation force (typically either through a broadband-exiting impact or a narrowband shaker scanned though a broad frequency range).

An experienced technician or engineer will couple knowledge of the hydraulic performance and mechanical behavior of the pump with the operational history and details of the pump design to determine probable causes of the problematic vibration. It is advisable to perform vibration testing during commissioning of critical equipment such that a record of the “newly installed” performance of the pump can be kept in the maintenance file for the pump. The vibration behavior of a pump on the manufacturer’s test stand can indicate problems with a pump, but will not necessarily be consistent with the pump’s behavior when mounted on a rigid foundation and connected to the process piping. In addition to the manufacturer’s “birth certificate” testing, it is also important to maintain detailed maintenance and operational records so that changes in vibration can be correlated to physical or operational changes in the system.

Instrumentation choices Critical machinery often has instrumentation permanently installed to measure the bearing housing and/or shaft vibrations. Typical modern instrumentation choices for pump applications are piezoelectric accelerometers (for housing vibration) and eddy current proximity probes (for shaft vibration relative to the housing). While both of these instruments measure vibration, they are quite different. Accelerometers measure the absolute acceleration of the surface on which they are mounted. Proximity probes measure the distance between the probe tip and the shaft. Changes in this distance are a result of vibration of the shaft relative to the housing, but does not include an absolute measurement of the housing on which the probe is mounted (such as with an accelerometer); and therefore from such a measurement, the absolute shaft vibration cannot be known. Older and noncritical machinery often does not have permanently installed instrumentation but is monitored periodically by predictive maintenance personnel using a portable analyzer and an accelerometer held in place by hand pressure or a magnet. The hand-held instrumentation can be effective, but the user must beware of “bouncing” or “rocking” of the accelerometer base on the machine contact surface, which is seldom flat or totally clean. A spear point on the accelerometer contact, or a thin layer of wax on a flat accelerometer base, can result in very accurate handheld readings, to frequencies of thousands of kHz.

If a problem is suspected in a pump or its driver, a more detailed investigation can be effective in determining the root cause of the problem and in assessing the risk of an

unplanned shutdown. Instrumentation which is available for such an investigation includes the following:

1. Portable or permanently installed accelerometers
2. Temporarily or permanently installed proximity probes
3. Dynamic pressure transducers
4. Microphones or their liquid-measuring counterparts, hydrophones
5. Temperature probes and/or infrared photography
6. Strain gages, sometimes monitored on a rotor, through FM telemetry

Each of these instruments can be very useful, but care must be taken to insure their proper application. Common problems which are encountered are loose attachment of accelerometers to the pump; electromagnetic interference, which is particularly problematic when unshielded cables are used; and inaccurate temperature measurement with infrared radiation detection devices due to differences in the emissivity of the surfaces. Consideration must also be given to the resonant frequency of the sensor which will produce a systematic error in frequency measurements close to it. Modern accelerometers have resonant frequencies which are 40 kHz or higher and have a usable frequency range of 10 kHz or higher.

Special instrumentation is available for measurements which must be made in extreme environments. High temperature instrumentation can be used to gather vibration data on boiler feedwater pumps and on steam turbine drivers. This instrumentation often has the signal conditioning electronics removed from the sensor such that it is not exposed to the high temperatures. It is also possible to cool the instrumentation by circulating cooling water through the sensor or its connection to the machine. Integral or intrinsically sealed cable connections allow for underwater use of sensors.

Vibrations can be measured in terms of acceleration (g's), velocity (in/s or mm/s), or displacement (mils or mm). Accelerometers measure the acceleration of the surface on which they are mounted, but it is very common for internal circuitry in either the sensor or the analyzer to convert the signal into velocity. Velocity-based vibration sensors have largely been replaced by accelerometers due to the excellent high frequency response of accelerometers, and the improvements in accelerometer robustness and low frequency performance. Exceptions are in noncontact measurements which cannot use surface mounted accelerometers. These measurements commonly employ eddy current or fiber optic proximity probes (displacement measurement) or lasers (displacement or velocity).

Past perception has been that acceleration sensors were best used to measure high frequency vibrations and displacement sensors were best used to measure low frequency vibrations, while velocity sensors were more universal than either in their application. Modern accelerometers are capable of accurately measuring vibrations at very low frequencies and have been successfully used below 1 Hz, making them universal for most pump applications. However, care must be taken if low frequency acceleration data are integrated to determine the vibration in terms of velocity or displacement. Division by frequency is required, and results in division by zero or at least a very small number at low frequencies, causing significant error. On the other side of the spectrum, proximity probes are capable of accurately measuring vibrations in the kilohertz range which is sufficient for pump applications which require noncontact measurements, and differentiation of displacement to determine velocity, for example, does not lead to significant error.

Vibration Analyzers Vibration analyzers are used to perform operations such as Fast Fourier Transforms (FFTs) on the raw data (vibration versus time) to determine the key vibration frequencies. Handheld portable analyzers are used by maintenance personnel to analyze and store vibration data from machines which do not have permanently installed and monitored instrumentation at key locations. Analyzers used for this type of vibration analysis should have at least two channels such that one channel can serve as a phase reference. Multichannel analyzers which are used by technicians or engineers for troubleshooting problems commonly have 4 to 32 channels of simultaneous data acquisition, and in principle can have hundreds of channels. Multichannel analyzers also have

digital “tape” recording capabilities, which allow the raw data to be saved for future playbacks, and alternative analysis. This feature is particularly useful for monitoring transient events such as equipment start-ups, speed changes, and load changes. These systems are more complicated and more expensive which makes them more useful to specialists in vibration analysis than to routine data acquisition for predictive maintenance. Regardless of the number of channels, the bookkeeping of measurement locations and directions of the vibrational motion is vital.

The FFT frequency spectrum analysis is critical in the understanding of the physics of most problems in that it breaks down a complex vibration waveform into its discrete frequency components. Analysis of which discrete (also called “filtered”) frequency components are dominant in the vibration spectrum provides valuable information which an experienced engineer or technician can use to identify potential root causes for the vibration problem. This information is used to determine the strong excitation frequencies and the vibration level at those frequencies versus tolerable levels, and with proper test procedures can be also used to determine any structural or rotor natural frequencies which are too close to the excitation frequencies, thereby increasing the vibration.

Analysis of vibration data through other more specialized instrumentation and analyzers, such as “shock-pulse-signal” or “stress-energy-signal” analysis, can be used for specific purposes such as identifying faults in rolling element bearings. Such analysis measures and evaluates the high frequency section of the frequency band and is not concerned with measuring narrowband vibrations. Techniques for identifying specific problems in pumps such as bearing faults and cavitation exist and/or are currently under development.

- 1. Recommended measurement locations** Locations for vibration measurements on pumps are recommended in the various specifications. The specification must be satisfied at each of these locations in order for the pump to be accepted. In general, it is best to measure the structural vibration at the bearings since these locations are close to the points at which the rotor and structure interact. It is recommended that vibrations be measured in three orthogonal directions at each bearing location, although sometimes the axial direction is bypassed at the nonthrust end of the machine. On critical-service pumps, shaft vibration measurements should be taken in two orthogonal directions at each bearing location for pumps with hydrodynamic journal bearings. Specifications such as API-610 and the HI/ANSI centrifugal pump standards, provide illustrations of typical recommended measurement locations for pumps with overhung rotors, between bearing pumps, and vertically suspended pumps (e.g., vertical turbine pumps).^{2,14}

2. Vibration acceptance testing and specifications

Specification Limits The vibrations measured on a pump and/or motor typically must remain below some specification limit. These limits have been developed by pump designers and users to set guidelines which indicate when vibrations reach a level which can cause damage to the pump. The goal of these organizations is to provide guidelines which reasonably define acceptable performance without either the manufacturer incurring unnecessary design and manufacturing costs or the end user risking avoidable maintenance costs throughout the life of the pump. The cost of achieving unnecessarily low vibration levels will be passed eventually from the manufacturer to the end user, so appropriate vibration specifications must be carefully considered when purchasing a pump.

Several organizations which have specifications that are commonly applied to pumps are the Hydraulic Institute (HI), American Petroleum Institute (API), and International Standards Organization (ISO). Each of these organizations is preferred by manufacturers and end users in various market segments and/or geographic areas. It is common for purchasers to modify these specifications to satisfy their particular application. Both purchasers and manufacturers should consider the probability of accepting a bad pump due to a vibration specification which is too lenient versus the rejection of a good pump due to a specification which is unnecessarily tight. In both cases additional time and costs are incurred by both the manufacturer and the end user.

When analyzing vibration data and comparing it to a standard, the most important thing to determine is whether the vibration level will result in premature damage to the

pump or are there indications that the vibration is trending in a way that indicates unacceptable wear in the pump. For pumps which have rolling element bearings, the oscillating forces acting on those bearings can be estimated and used to consider whether the bearings are likely to deteriorate. Rotor vibration on pumps with hydrodynamic journal bearings can lead to the shaft rubbing on the bearings which will result in premature wear. It is typical for a certain percentage of the bearing clearance (normally about 50%) to be used as the maximum allowable peak-to-peak vibration level. Other considerations are fatigue of stationary structural components, fatigue of rotor components, and coupling failure. Equally as important as simply determining whether it is safe to continue to operate is to identify the root cause of any abnormal vibration such that future failures can be eliminated or at least predicted.

The goal of specifications should be to protect all parties—manufacturers, consultants, contractors, and owners—by clearly defining and assigning responsibilities for the avoidance of unacceptable vibration in newly installed pumping systems. The effectiveness of any measures taken to minimize the likelihood of vibration problems, and to quickly solve them if they occur, will depend upon the level of technology used. The inclination or ability of low-bid equipment suppliers to provide an appropriate level of technology should not be assumed, but should be specifically required by the bid and contract.

Injury to rotating and reciprocating machinery due to excessive rotor vibration takes the form of wear or fatigue of the internal components, such as bearings, seals, and shafting. In general, wear rate and fatigue tendency for a specific machine is proportional to the oscillating displacement associated with vibration. However, testing by various sources over the last 70 years has indicated that for *classes* of machines (e.g., centrifugal pumps), operating at perhaps a range of different speeds and loads, vibration velocity rather than displacement, correlates best with likelihood of wear (tribological) or fatigue (metallurgical) failure. The reason for this is not made clear in most references concerning acceptable vibration levels and machinery vibration acceptance standards. Early references on this subject were Rathbone (insurance company data),¹⁵ Blake (the so-called “Haystack Chart”),¹⁶ Baxter and Bernhard (the so-called “TRD Chart”),¹⁷ and Hancock (petrochemical plant experience with vibration level versus failure rate for a large population of industrial pumps).¹⁸

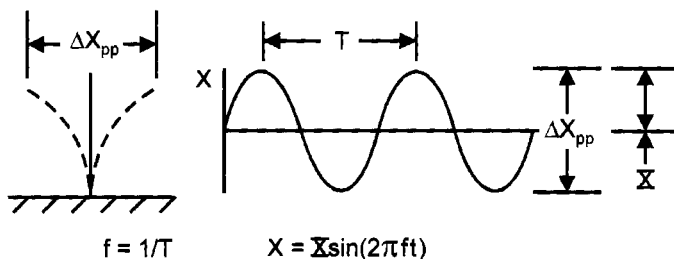
Each of the classical references concluded that for large groups of machinery of various sizes and speeds, vibration velocity (i.e., the first derivative with respect to time of the vibration displacement) correlated better than displacement with machinery deterioration rate and current state of health. However, for a given machine that vibrates primarily at a constant running speed, as the great majority do (because variable speed machines are still the exception), and when residual imbalance is the main forcing function (as it indeed is for most in-service machinery), it was accepted that either displacement or velocity level could be used for trending purposes. This is because velocity was merely displacement times speed times a units constant, as shown in Figure 19.

BALANCE AND ALIGNMENT

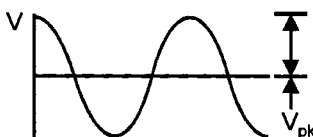
In industry, imbalance and misalignment represent about 80 to 90% of the vibration problems in rotating equipment, and these are relatively easy to fix or minimize most of the time. Balancing means a process to restore the imbalanced rotor to a balanced condition by adjusting the mass distribution of the rotor. Depending on the type of imbalance, rotors can be balanced using either a single plane or a multiplane method, as described below.

Single Plane Balancing Static imbalance is a term used when the central principal axis is displaced parallel to the rotational axis in a rigid rotor. The residual imbalance in such a case requires only a single mass located in a single plane (through the center of mass of the rotor) to balance the rotor, as shown in Figure 20.

Multiplane Balancing The two-plane balancing method is used for rotors with couple unbalance or dynamic unbalance. The couple unbalance or “dynamic imbalance” means a

1. Displacement Δx_{pp} (mils or μm)2. Velocity V_{pk} (ips or mm/sec) = $\frac{dx}{dt}$

$$V_{pk} = \text{constant} * \Delta x_{pp} * f$$

3. Acceleration a_{pk} (g's) = $\frac{dv}{dt} = \frac{d^2x}{dt^2}$

$$a_{pk} = \text{constant} * X_{pp} * f^2$$

NOTE: $V_{RMS} = 0.707 V_{pk}$ & $a_{RMS} = 0.707 a_{pk}$
at any single frequency

FIGURE 19 Relationship of vibration displacement, velocity, and acceleration

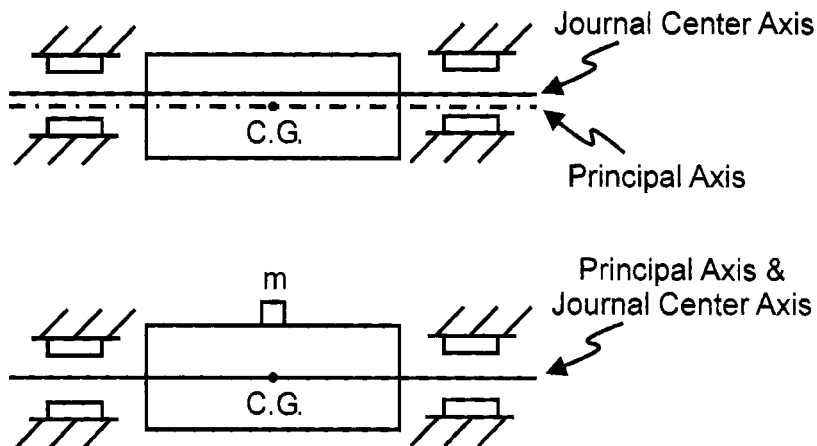


FIGURE 20 Single plane balancing

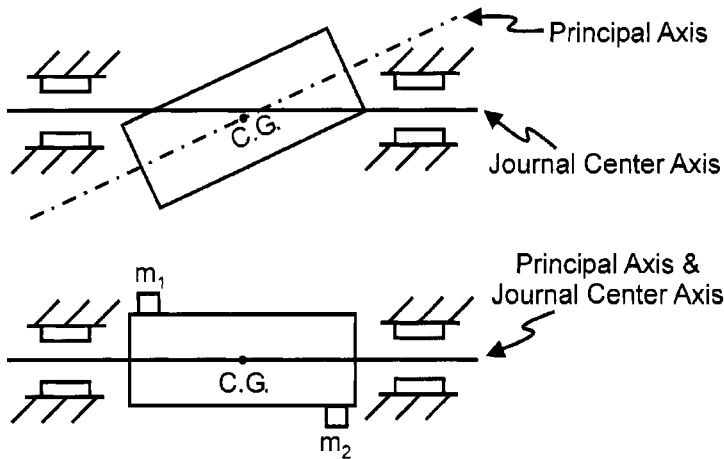


FIGURE 21 Two plane balancing

state of unbalance in a rigid rotor where the central principal axis and the rotation axis are not parallel with each other. In this case, mass is required on each end to balance the rotor. In such a case, at least two planes are required to balance the rotor, as shown in Figure 21.

Balance Specifications In the real life, is impossible to obtain a perfectly balanced rotor. Several standards have been developed over the years to specify the maximum residual unbalance allowable according to the equipment application and operating speed. The ISO 1940 imbalance tolerance standard specifies different grades of imbalance tolerance as a function of speed. Other standards include American Petroleum Institute (API) 610, which has recently been modified and adopted as an ISO standard, as has the HI/ANSI pump standard.^{2, 14, 19} For tight fitting components on pump shafts, the recommended balance limit is somewhat tighter than the "precision" ISO Grade G-2. A typical pump balance specification is:

For USCS units,
$$U = \frac{4W}{N} \quad \text{in oz-in}$$

For SI units,
$$U = \frac{6350W}{N} \quad \text{in gr-mm}$$

where

U = maximum allowable residual unbalance for each plane based on the static weight supported at each end of the rotor

W = portion of static rotor weight supported by each bearing journal (lbm for USCS units and kg for SI units)

N = maximum continuous operating speed of the rotor in revolutions per minute (rpm)

For axially short components (e.g., a thrust washer) the difference between these two can be neglected, and only single plane static balancing is required. For components greater in length than $\frac{1}{6}$ their diameter, dynamic imbalance should be assumed (and in fact is required by API-610 in this case), and at least two plane balancing is required. For rotors operating above their second critical speed (unusual for pumps), even two plane balance may not be enough, and some form of at-speed modal balancing (i.e., balancing

material removal that takes into account the closest natural frequency mode shape) may be required.

When imbalance occurs, including imbalance caused by shaft bow, it shows up with a frequency of exactly $1 \times$ running speed. This is because the heavy side of the rotor is rotating at exactly rotating speed, and so forces vibration movement at exactly this frequency. Typically, this also results in a circular shaft orbit, although the orbit may be oval if the rotor is highly loaded within a journal bearing.

Alignment Measurement Techniques and Guidelines Next to unbalance, in industry misalignment is the primary reason for high vibration in rotating equipment and can cause severe damage on coupling elements through fatigue of dry element coupling diaphragms or shim packs, lubricated gear coupling tooth wear, overheating, bearing pre-loading, and—in the case of severe misalignment—shaft fatigue.

Over the years, several techniques have been developed for measuring misalignment:

- Straightedge method
- Face and rim method
- Reverse dial indicator
- Continuous alignment monitoring, for example with “Dodd bars” or “Essinger bars”

STRAIGHTEDGE METHOD This is the least accurate method of alignment (0.10 mm or 5 mils), and typically should only be used for a quick, rough preliminary alignment by using a straight ruler between coupling hubs with the same outside diameter.

FACE AND RIM METHOD This method uses a dial indicator, and offers more accurate readings (0.010 mm or 0.5 mils). Is typically used in industry for cold readings, where the offset misalignment is measured by placing the dial indicator on one of the hubs, and then measuring the other hub in the radial direction, by rotating both shafts together and taking measurements in each quadrant (every 90 degrees). In this case, the sum of horizontal readings must be equal to the sum of the vertical readings. Usually, the pump manufacturer recommends the misalignment requirement at cold condition in order to reach the desired alignment condition during operation once the thermal growth is reached on both shafts. Offset misalignment is most of the time combined with the angular misalignment. In this case, the reading is taken in the axial direction on the coupling hub (face reading). The disadvantages of this method are that the diameter of the hub surfaces must be circular without imperfections or irregularities on this surface, as well as concentric with the face surfaces, unless the readings are taken by rotating both shafts together. The other disadvantage is that either the pump shaft or the driver should not float axially; otherwise, an axial stop should be used to reduce the face reading errors.

REVERSE DIAL INDICATOR This method is similar to rim method, but in this case, two dial indicators are used, reading on each hub (pump and driver). This method is commonly used in order to verify the alignment once the temperature in the equipment has stabilized, by bringing the equipment to a stop, and then quickly setting up and operating the reverse dial system, as a “hot alignment check.” Laser optical alignment is also used, assisted by a computer program to facilitate and reduce the time of the alignment process for the user. This method offers high precision. The principle is simple, and employs a laser beam transmitter, which is clamped to one shaft. A reflecting prism is mounted on the second shaft, with the assembly acting as in the reverse dial indicator method, but typically without shaft sag. The beam passes from transmitter to prism, the angled faces of which then reflect it directly back toward the origin but along a different path. The returning beam strikes a target that electronically senses shift from a zero position. Electronic circuitry translates that shift into the type and amount of misalignment. The necessary adjustments are displayed in the computer program graphics.

DODD BARS Dodd bars can be used to monitor the alignment between two units (driver and driven) during the transient condition until the final thermal growth is reached.²⁰

TABLE 1 Guidelines for proper alignment

Think in terms of offset in combination with parallel misalignment
Reverse Dial Indicator is most foolproof method
Check foot bolt tightness before starting, ensure gage assembly is tight and stiff
Set all gages to mid-travel
Rotate gage assembly several times to “shake-out” hysteresis
Do repeatability readings several times: Readings should repeat within 1 mil, 3:00 + 9:00 = 6:00 + 12:00
Take final readings in 4 positions: 3:00, 6:00, 9:00, 12:00
Quantify sag in indicator measurement arm, on straight rigid shaft
Subtract sag from 12:00 reading, add sag to 6:00 reading
Misalignment should be (this is a general guideline only): Offset: less than 2 mils * (3600/RPM) Parallel: less than $\frac{1}{2}$ mil $\times$ (3600/RPM) per axial inch
Remember: Alignment when machine is cold and non-pressurized will be different than when machine is hot and pressurized. May need non-zero “cold offsets” for best COS alignment, and may need compromise alignment for variable COS.
Beware of “soft foot” (e.g., “teetering” pump, or delaminated foundation)

These bars are made of titanium, with low thermal coefficient which reduces the distortion of the readings by reducing the thermal deformation of the bars. Four proximity probes are mounted on one of the bars on the driver machine bearing bracket monitoring both the horizontal and vertical directions at two axially offset locations on the other bar, which is mounted through a bracket to the driven machine bearing housing.

Sometimes a rotor must be offset when cold and not running in order to be aligned when running and hot. For example, thermal growth of the pump or its pedestal versus the driver could change alignment “hot” versus “cold.” This situation can be checked with various continuous monitoring tools, such as Dodd bars or Essinger bars, that monitor the offset and angular movement of the pump and driver bearing housings relative to either each other (Dodd bars) or relative to the foundation as a reference (Essinger bars).

Misalignment is best checked by some form of the reverse dial indicator method (or its laser alignment tool equivalent). Guidelines for a proper alignment process are provide in Table 1.

FIELD VIBRATION TROUBLESHOOTING PROCEDURES

Typically, vibration monitoring programs are designed to determine if there is a serious vibration problem, but more sophisticated analysis is required to determine the root cause of the problem and to identify suitable fixes. A good starting point for troubleshooting of vibration problems is to identify particular operating conditions (speeds, flow rates, and so on) which result in high vibrations. The next step is to measure high resolution vibration spectra at the conditions which result in high vibrations to identify the frequencies at which the highest components of vibration are present and the locations on the machinery where the highest vibration occurs. This basic information can be used to sort through potential causes of the vibration using a vibration troubleshooting list, as shown in Table 2. In some cases, additional data such as pressure pulsation measurements may be required to

TABLE 2 Pump vibration troubleshooting list

Multiples of Rotational Speed	Other Symptoms	Probable Causes
0.05 to 0.35x	Vibration response is broad in frequency	Diffuser or return channel stall
Exactly $\frac{1}{2} \times$, $\frac{1}{3} \times$, $\frac{1}{4} \times$ etc.	Vibration may increase dramatically shortly after this appears, except in vertical pumps with four or more lineshaft bearings, where it is common and not harmful	Light rub, possibly combined with low shaft or bearing support natural frequency
0.40 X (Approx)	Shaft has rolling element bearings	Bearing Cage Defect
0.42 to 0.48 X	Near rotor lateral natural frequency and the orbit has a secondary loop.	Rotordynamic instability due to fluid whirl in close clearances, e.g., bearing “oil whip”
	Onset is sudden at a speed roughly 2 times the excited natural frequency, and “locks” onto the natural frequency in spite of further speed increase	
0.65 to 0.95x	Smaller peaks at (1-f) X, and at sidebands of $\pm(1-f)$ X of the first several multiples of running speed.	Internal flow recirculation probably at suction
	Often accompanied by rumbling noise and beating.	
	Occurs at part-load capacities, but disappears at very low capacity.	
	Independently depends on speed and flow	
Less than 1x	Increased broadband vibration and noise level below running speed as NPSH decreases, especially at high flows.	Cavitation without recirculation
	Often accompanied by decreased vibration and noise above 1x.	

(Continued)

TABLE 2 Pump vibration troubleshooting list (*Continued*)

Multiples of Rotational Speed	Other Symptoms	Probable Causes
1×	Stronger on shaft than on housing.	Imbalance in Rotating Assembly or bent rotor
	Hydraulic performance and/or suction pressure normal	
	Axial vibrations within normal limits	
	Vibrations increase with roughly speed squared	
1×	Vibration highest on drive end housing	Pump coupling imbalance
1×	Vibration high on driver IB or OB housing, low on pump	Driver rotor imbalance
1×	Natural frequency near 1×	Resonance
1×	Axial vibrations are over $\frac{1}{2}$ of Horizontal or Vertical vibrations.	Pump/driver misalignment at the coupling
	Vibrations increase much slower than the square of the speed	
	Bearing oil temperature is high.	
1×	Discharge pressure pulsations are strong at 1×	Clogged or damaged impeller passage
	Discharge pressure pulsations are not strong at impeller vane pass (single volute pump)	
1×	Discharge pressure pulsations are strong at 1× Discharge pressure pulsations are strong at impeller vane pass, especially for flows far above or below BEP	Volute tongue designed too close to impeller OD (B-gap), or excessive impeller/volute eccentricity
2×	Both shaft and housing vibrations are strong	Clogged or damaged impeller passage
	Discharge pressure pulsations are strong.	
	Impeller vane pass vibrations are low (twin volute pump)	

2×	Both shaft and housing vibrations are strong	Volute vanes designed too close to impeller OD (B-gap), or clogged or damaged volute, or excessive impeller eccentricity
	Discharge pressure pulsations are strong	
	Unusually high vane pass vibrations and discharge pressure pulsations	
2×	Shaft vibrations much stronger than housing vibrations, and approach or exceed bearing clearance	Volute eccentricity, Looseness in bearing support or cracked shaft
	Decrease in shaft first bending natural frequency	
	Other multiples of running speed may be stronger than usual	
2×	Shaft vibrations stronger than housing	Torsional excitation
	Strong torque pulsations	
2×	Axial vibrations are more than $\frac{1}{2}$ of Horizontal or Vertical	Pump/driver misalignment at the coupling
	1x vibrations are also high	
	Bearing oil temperature is high	
Number of Impeller Vanes Times Running Speed (m*X)	Both shaft and housing vibrations high	Volute too close to impeller OD due to design (B-gap) or excessive rotor eccentricity
	Reasonably low discharge pressure pulsations	
m*X	Housing vibrations much higher than shaft vibrations	Piping mechanical resonance at vanepass
m*X	Discharge pressure pulsations high at vane pass frequency high	Acoustic resonance in discharge pipe
	Suction pressure pulsations reasonably low	

(Continued)

TABLE 2 Pump vibration troubleshooting list (*Continued*)

Multiples of Rotational Speed	Other Symptoms	Probable Causes
m*X	Suction pressure pulsations also high	Acoustic resonance in suction pipe
m*X	Pump rotor or casing natural frequency close to vane pass	Resonance
Several multiples of running speed (1×, 2×, etc.)	Orbit shows sharp angles or shows evidence of “ringing”	Internal rub or poorly lubricated gear coupling
	Spectrum shows evidence of exactly $\frac{1}{2}$ or $\frac{1}{3}$ × response Grinding noises and speed changes may be evident	
Several multiples of running speed (1×, 2×, etc.)	Orbit is “fuzzy” but does not pulse or “ring”	Jammed, clogged, or damaged seal
	Seal coolant flow is unexpectedly high or low, and may exhibit high temperature	
	Spectrum may also exhibit $\frac{1}{3}$ or $\frac{1}{2}$ running speed	
Several multiples of running speed (1×, 2×, etc.)	Orbit pulses, usually in one direction much more than the other	Shaft support looseness, especially in bearing insert or cap retention.
	Shaft vibrates more than housing	
Several multiples of running speed (1×, 2×, etc.)	Shaft orbit fairly steady	Looseness in pump casing, pedestal, or foundation.
	housing vibrates more than shaft	
	Often combined with vibration response over a broad range of frequencies below running speed	

Number of Bearing Rolling Elements (Nb) times about $\frac{1}{2}$ Rotor Speed N	freq $f = 0.4 N_b * N / 2$ approx	Defect Bearing Outer Race
	freq $f = 0.5 N_b * N / 2$ approx	Defect in Ball or Roller
	freq $f = 0.6 N_b * N / 2$ approx	Defect in Inner Race
2 times line frequency	Very directional	Motor Stator Eccentricity
2 times line frequency	Very directional	Shorted stator lamination (induces thermal eccentricity)
	Increases as motor operates (warms up)	
2 times line frequency	Pulsing vibration $2 \times LF$ and closest harmonic of running speed	Motor Rotor Eccentricity
	Also pole pass sidebands around running speed, pole pass frequency = slip frequency * number of poles	
2 times line frequency	Sidebands at $\frac{1}{3} \times LF$	Loose or broken phase connection
Rotor bar pass frequency	$2 \times LF$ sidebands around rotor bar pass frequency (RBPF)	Loose or Open Rotor Bars
	Often higher $2 \times RBPF$ than $1 \times RBPF$	

narrow down potential causes. This list can serve as a guide for additional measurements or in some circumstances may even lead directly to the root cause and solution.

Vibration testing to determine pump mechanical health is similar in nature to the “chest rapping” and “heart and lung listening” tests that a physician performs during a physical. Comparing to established norms the amplitude and frequency resulting from chest impacts and from your naturally occurring pulse and aspiration, the doctor diagnoses your condition. When vibration field problems occur, analogous testing can be performed, for example, signature analysis can determine the frequencies which are most strongly excited, suggesting possible root causes to someone familiar with the vibration characteristics of the pump internal components and the systems to which the pump is attached. Following signature analysis, experimental modal analysis (EMA) has proven its ability to separately determine the excitation forces and the natural frequencies of the pump system.

Troubleshooting Techniques The following measurements are suggested as a minimum for predictive maintenance or vibration troubleshooting of any style pump :

1. What the vibration level is on both bearing housings on pump, and on the pump-side (i.e., “inboard”) bearing housing on driver in the vertical, horizontal, and axial directions.
2. How hydraulic performance compares to design. In other words, for a given speed and capacity (i.e., flow rate), how close is the head of the pump to the curve supplied by the manufacturer, especially near the design or best efficiency point (BEP)? Are the head and capacity steady when the operator tries to hold the pump at a constant speed? Is the motor or steam turbine driver required to provide more power than expected?
3. What the bearing shell or lubricant exit or sump temperatures are.
4. Whether the suction pressure is steady at a given operating point, and well above NPSH requirements.
5. Whether unusual noises are present at certain operating conditions, and if so, what their main frequencies are.

For high energy multistage pumps, particularly, the following measurements are also recommended:

1. Vibration of the shaft relative to the housing near each bearing, using proximity probes permanently installed in each bearing housing to monitor vertical, horizontal, and axial displacement and vibration.
2. What the monitored leakage rates and exit temperatures are in the thrust balancing device (if any) leak-off line and seal coolant feed or lubrication lines (if any).
3. Whether any unexpected wear particles or pumpage contamination are detectable in samples taken of the lubricant on a regular basis.
4. What the pump shaft and casing natural frequencies are, and what the vibration response to a unit load near the bearings is at these frequencies, as determined by experimental modal analysis.

Recognizing and Solving Vibration Problems Probably 90% of all pump vibration problems can be solved by careful balancing of the rotor assembly, alignment of the coupling when the system is at its rated conditions (especially if it is hot), for example, per Dodd, and running the pump within the bounds of its specified head versus capacity curve.²⁰ Remaining pump vibration problems are generally due to a resonance of a system natural frequency with one of the excitation forces common to all pumps, such as residual unbalance. During resonance, the rotor vibrations can exceed clearances, or excessive bearing loads can occur, even if unbalance, misalignment, and hydraulic loads are within normally acceptable limits.

In the case of resonance, simple trial-and-error field fixes, such as addition of clamped weights or brace-stiffening of the apparently offending portions of the structure, can often

be effective. However, if such “fixes” only shift the problem to a slightly different frequency, and cannot be made effective within a reasonable period of time, it is important to stop cut-and-try and answer the following questions :

1. Are the excitation forces within normally acceptable limits?
2. What are the natural frequencies of the rotating and stationary parts of the pump and attached structures when they are assembled as a system, and have they changed?
3. What are the excitation force frequencies in the actual installation, and do they coincide with the any of the assembled system natural frequencies?
4. If there is a vibration problem due to excessive excitation force or a resonance, how can the pump or system be changed most easily to solve the problem? In testing for resonance, typical vibration test locations include the bearing housings, the casing, the pedestal or baseplate near the pump attachment bolts, the baseplate near the foundation bolts, the nearby foundation and piping, and any already instrumented or exposed portion of the pump shaft including the driver coupling.

In performing vibration troubleshooting, generalized charts matching symptoms to possible causes can be useful for many typical or simple problems. However, do not rely too heavily on such lists, especially if their initial application does not lead to immediate resolution of the problem. Persistent pump vibration problems are usually due to an unexpected combination of factors, some of which are specific to the particular pumping system, like mechanical or acoustical piping resonances, or hot running misalignment of the pump/driver due to thermal distortions of the piping or baseplate. With this warning, a list of some of the more common pump vibration symptoms and their causes is given in Table 2. The list is *not* meant to be all inclusive, and is in the order of the frequency value observed, not in order of likelihood or importance to reliability. Definitions useful in reading this list are:

Runout: False vibrations picked up by a proximity probe, actually reflecting shaft scratches, gouges, undulations, and so on, or static shaft bends or misalignment (“mechanical runout”), or eddy current sensitivity variations along the shaft surface or instrumentation noise (“electrical runout”). This can be determined by observing the apparent shaft orbit when the shaft is slowly turned.

Inboard: The coupling side of the pump or driver.

Outboard: The “nondriven” end of the pump or driver opposite to the coupling.

Narrowband: Vibration response which is at a single frequency.

Broadband: Vibration response which covers a band of frequencies on a spectrum plot.

Puzzle Approach It is often not possible to measure the behavior of the entire pump from measurements on the accessible portions of the pump. In such cases, knowledge of the pump and system can be used in conjunction with the available vibration data to “puzzle through” likely causes of the vibration problem. For instance, it is difficult to measure the rotor deflections at the many locations along the span of the rotor in a multistage pump. However, the deflected shape of the rotor at key frequencies can be deduced from measurements at the bearing locations (to identify natural frequencies of lateral rotor modes) when combined with rotordynamic calculations for the rotor system.

A time-proven technique for identifying and solving many problems is to determine the offending excitation sources and the natural frequencies of the structure and rotor. Knowledge of both helps an engineer or technician to determine if the vibration problem is a result of an abnormally high excitation force as would be expected from rotor imbalance, or a result of a normal excitation force which is being amplified by a nearby natural frequency. The symptom is the same in either case, but the means of fixing the problem (not just alleviating the symptom and allowing it to return later) is significantly different. In the case of an abnormally high $1\times$ excitation force, correcting the balance of the rotor may be a suitable solution. In the case of amplification from a natural frequency, balance of the rotor will be a short-lived solution which will permit the vibrations

to increase to unacceptable levels as the rotor wears or deposits accumulate and the imbalance returns.

Complementary Measurements Many problems manifest themselves as mechanical vibration in pumps, but effectively troubleshooting them often requires measurements in addition to the external structural vibration. Measurements which help to define the cause of the vibration are dynamic pressure measurements in the suction and discharge piping, acoustic measurements, thermography, static and dynamic torque measurement, oil or grease analysis, and electric current spectral analysis. These complementary measurements can identify abnormally high pressure pulsations in the suction or discharge, acoustic resonance in the piping, high torque pulsations resulting from electrical problems, and other unexpected difficulties.

Oil Monitoring Analysis A new technology has reached maturity recently known as oil monitoring analysis.²¹ This involves monitoring the condition of the bearing lubricant, both to determine whether the lubricant needs replacing, as well as to use the lubricant as a "telltale" of whether there are problems in the pump or its driver. Typical methods include chemical analysis and the observance of the amount of particulates in the flow. Chemical analysis can consist of determining the oil's total acid number TAN or total base number TBN (determines if the antioxidation capability of the oil has degraded, implying excessive temperature in the bearing), running a spectroscopic analysis in which the atomic spectra of the small particles and other constituents in the oil are examined (determines if bearing metal is flaking off, or if dirt or contaminants are entering the oil), or running a water-detection test such as a Karl Fischer test. Particulate analysis can take the form of observing the degree to which light is blocked by a lubricant sample (implies how many particles are in the oil), or can consist of "ferrography" in which metal particles are collected from the flowing oil by a magnetic strip, and examined under a microscope for shape, size, and color. Other specialty tests are possible, from the intuitive (smell and color) to the very sophisticated (electron microscope examination of wear particles).

Oil analysis is better than vibration for picking up the early stages of lubrication or wear problems in rolling element bearings or gears, or for detecting severe misalignment where each rotor is "pinned" against its bearing housing by the misalignment forces.

Thermography Thermography involves mapping out the surface temperatures on a machine or system component, looking for lack of symmetry where symmetry should exist, for example, around the periphery of a bearing housing.²² It can be a very sensitive method of discovering particular problems. In the case of the bearing housing, for example, if a heavy misalignment were present, a hot spot would be observed on one side of the pump bearing housing, and a hot spot would be also observed on the opposite side of the driver bearing housing. Thermography is most easily implemented using thermally sensitive video equipment, but infrared cameras are still very expensive, particularly if temperatures are shown in color rather than as shades of gray. Although it takes a little more time, exactly the same information is available for a much smaller budget by plotting on paper the readings from a surface thermometer, surface thermocouple, or an inexpensive infrared gun. In the case of either a \$40,000 infrared camera or a \$200 infrared gun, be careful that what is being detected is emitted radiation, which depends not only on the true temperature, but also on the emissivity of the surface, typically about 1.0, but which can decrease up to about a factor of 2 depending upon amount dirt, roughness of the surface, and whether the surface has been freshly painted. If readings are taken off a piece of material (e.g., fresh duct tape) firmly pasted on the surface rather than from the surface itself, problems with this issue can be avoided.

Electric Current Spectrum Monitoring Oak Ridge National Laboratory (ORNL) and the U.S. Navy pioneered this method. It involves observing output from an oscilloscope or FFT analyzer at the output from an inductive "clamp-on" electric current probe, that is attached in an encircling manner around a motor phase lead.²² The evaluation is like

interpreting a vibration spectrum, such that whatever frequencies show up strongly in the current are assessed relative to whether they are too high in current, and if so what would physically cause this. The most common problem frequencies are induction motor $2\times$ slip frequencies (including $1\times$ speed $\pm 2\times$ slip “sidebands”), of course the line frequency, twice the line frequency, the rotor bar passing frequency, and the stator slot passing frequency. Broken rotor bars, for example, lead to increased slip sidebands and stronger levels of slotpass. Theoretically, faults in the pump that affect the motor torque and therefore current should also be evident in the spectrum, but generally vibration is more sensitive for pump problem detection.

Operating Deflection Shapes A very useful modern troubleshooting method is called operating deflection shape (ODS) analysis. In this method, vibration is detected by probe signals fed into an FFT analyzer, where the data are taken in three orthogonal directions at all representative locations on a pump and its relevant system (baseplate, nearby piping, driver, exposed shafting). The data are taken such that, as probes are “roved” over the pump and system during different portions of the test, one reference probe is always left at the same location and direction, to allow “phasing” (timing) of when the motion peaks in one group of test points versus another. The resulting database is plotted as an animation that exaggerates the scale of the motion, so that it is easy to make physical interpretations of causes for high vibration at a certain frequency. For example, a high pump versus motor out-of-phase oval motion may be observed at a frequency of $2\times$ running speed, indicating misalignment.

Experimental Modal Analysis or “Bump” Testing It is possible that, in spite of competent preinstallation vibration analysis, complicated vibration problems may still occur in the final installation due to, for instance, idiosyncrasies in foundation material properties or in bolted joint behavior, inaccurate mass information supplied by motor or drive manufacturers, or an accidental defect in materials or workmanship. If the standard field service troubleshooting procedures of the pump manufacturer are not effective within a reasonable period of time, then modal analysis testing can be used to understand such problems quickly, particularly when combined with ODS. In addition, some form of modal analysis, using known exciting forces, such as from a calibrated impact hammer, should be used at the time that the pump and piping installation is completed to verify that the frequency versus excitation force frequency margins of structural vibration modes closest to $1\times$, $2\times$, and vane pass frequencies meet acceptance standards. If problems exist, modal analysis software can be used to extend the experimentally based model with “what-if” proposed bracing or other modifications (called structural dynamics modification) should be used to determine practical solutions. However, exercise care:

- Modal analysis testing requires skill and experience to be applied usefully to pumps and pumping systems. Proof of such capability should be supplied by the agent who will perform the test.
- The modal analysis test should include at least 50 locations (usually with three orthogonal directions at each location) of the response probes, including the corners of the baseplates, the foundation floor and pedestal, key segments of the pump stationary structure (e.g., vertical pump discharge head flanges, horizontal pump bearing housings and casing ends, driver housing ends), and the nearby attached piping and other important structural details.
- The test should include a means for discriminating between apparent modal response caused by undocumented pump operational forces and environmental background vibrations, and the actual modal response due to the documented artificial excitation forces of the modal test.

In experimental modal analysis (EMA), a known force (constant at all frequencies within the test range) is put into a pump, and the pump’s vibration response exclusively due to this force is observed and analyzed. EMA can determine the vibration characteristics

of pumps either on the test stand or in the field. The actual natural frequencies of combined casing, piping, and supporting structure can be obtained, and if time-averaging data collection procedures are used, EMA can also determine the rotor natural frequencies at the pump operating conditions. Separately, the frequencies of strong excitation forces within the pump can be estimated by comparing the vibration versus frequency spectrum of the pump's EMA artificial force response to the signature analysis spectrum of the pump's response to the naturally occurring forces from within the pump and from its attached system and environment. Figure 22 shows how the broadband "terrain" or peaks and valleys from an EMA impact test, when amplitude is plotted logarithmically to accentuate low level response and put it on the same footing as the largest response frequencies, can be used to better understand whether the high vibrations are caused by the amplifying effect of a natural frequency, or are just due to an excessive oscillating force. Knowing this allows a troubleshooter to focus on the root cause, and provide a fundamental fix.

The main tools required to do EMA are a multichannel FFT frequency analyzer, a computer with specialty software, a set of vibration response probes such as accelerometers

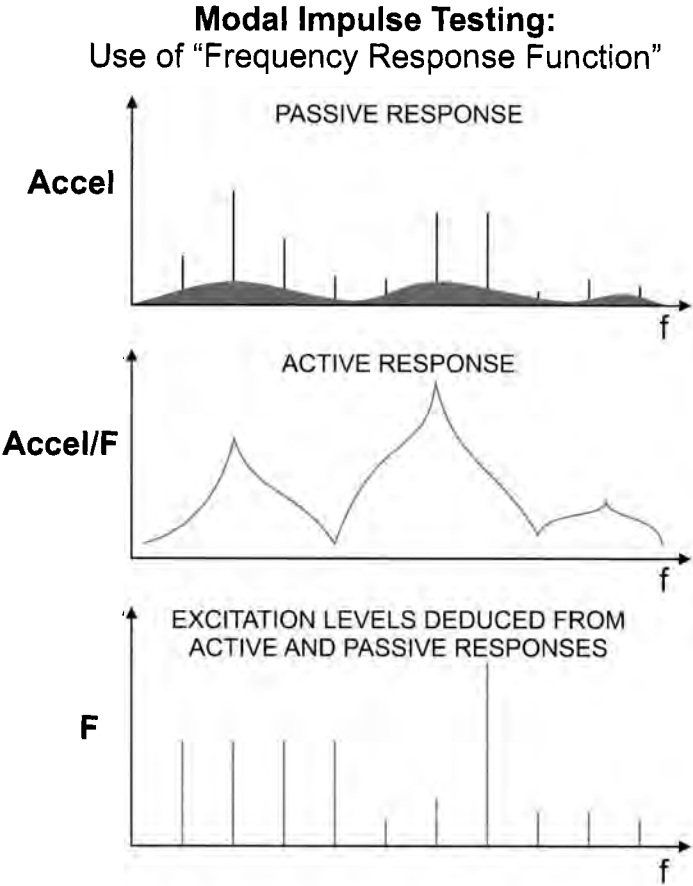


FIGURE 22 Vibration frequency spectrum dependence of natural frequencies and forces F (Mechanical Solutions, Inc.)

or proximity probes, and a load-cell-instrumented impact hammer designed to spread its force over a frequency range which covers the test range, as if the results of a number of shaker tests were combined.

The impact hammer should have an accelerometer load cell in its head which is calibrated to indicate the force being applied. During an EMA test, the signal from the hammer input force accelerometer is sent to one channel of the spectrum analyzer, and the signals from the vibration response probes are sent to the other channels. Dividing, at each frequency, the response channels by the impulse force channel gives the "frequency response function" (FRF) of the pump and its attached system. The peaks of the FRF are the noncritically damped natural frequencies, and the width and height of the peaks indicate the damping of each natural frequency, and how sensitive vibration at the test location is to forces which occur in the vicinity of the hammer impact at frequencies near a given natural frequency. The details of classical modal analysis and its application are given by Ewins.²³

A variation of EMA has been developed by Marscher to achieve useful results within the time and operational constraints of actual field tests. This technique is known as time averaged pulse (TAPTM). The method statistically reorganizes the data obtained from modal analysis in order to reliably determine structural natural frequencies and mode shapes, resonating force locations and frequencies, and rotor critical speeds, with all testing performed while the pump remains operating at the problem running condition. The method then uses classical modal analysis processing techniques to produce an animated model of each natural frequency's vibration pattern, and to make predictions about the effectiveness of proposed design changes, such as stiffer bearings, new piping supports, or a thicker baseplate. The TAPTM method may be applied to machines at any operating speed and load.^{24, 25, 26}

Determining the critical speeds of a pump rotor by impacting while the pump is operating at the condition of interest provides an alternative to the current practice of basing critical speeds on the peaks of "waterfall plots" of spectra obtained during pump start-ups or shutdowns. Waterfall plots can poorly represent pump rotor dynamic performance during steady state operation due to the strong sensitivity of annular seal stiffness and damping to operating conditions. The primary reason that pump natural frequencies are sensitive to operating condition is the Lomakin effect, which can significantly change the rotor support stiffness and damping, depending upon the pump's speed and load. The strong dependence on speed of a large fraction of the rotor support stiffness leads to differences between the frequencies of peaks in the transient waterfall plot and actual natural frequencies at steady state operating conditions of interest. This error is greater in pumps than in compressors and gas turbines, where circumferential variation in the seal static pressure does not get as high and therefore waterfall plotting has enjoyed success.

The analytical counterpart of the waterfall plot is the Campbell diagram, which plots vibration excitation frequency versus running speed. Since the strongest vibration excitations in pumps usually occur at integer multiples of the running speed, these are plotted on the diagram as diagonal lines emanating from the plot origin. Likewise, roughly horizontal lines are plotted for the first several calculated rotor natural frequencies. Intersections of excitation and natural frequency curves are marked by circles of radius equal to 10% of the frequency at which the intersection occurs. If any part of any circle lies between two vertical lines which represent the minimum and maximum running speeds, then a resonance is predicted, requiring steps to shift the problem natural frequency, increase its damping until it is critically damped, or eliminate the exciting force.

PROGNOSIS: DETERMINING PROBLEM SEVERITY

Physical Relationship between Vibration and Damage Figures 23 and 24 show typical failure modes and associated frequencies in a representative pump style. The point of these figures is not to be all inclusive of pump problems, but to show major issues, and how the problem relates to the resulting vibration.

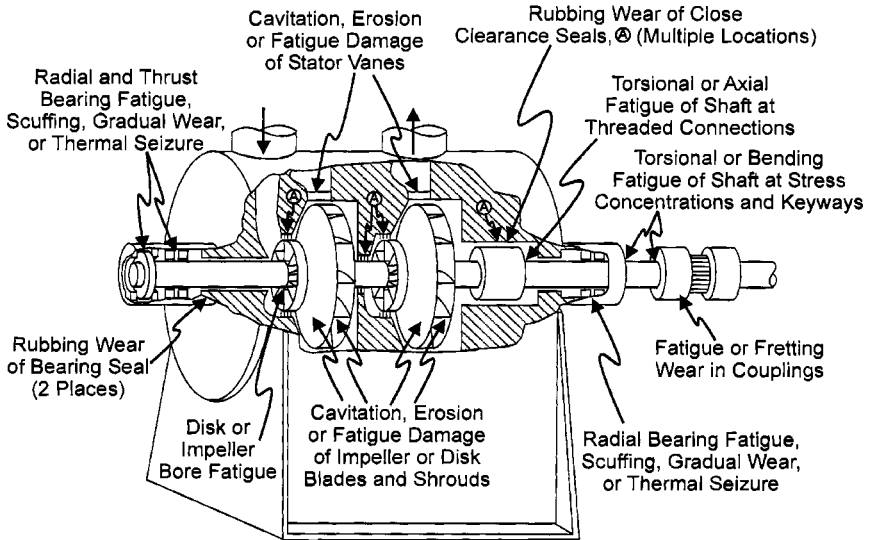


FIGURE 23 Horizontal pump symptoms of typical hydraulic or mechanical problems (Mechanical Solutions, Inc.)

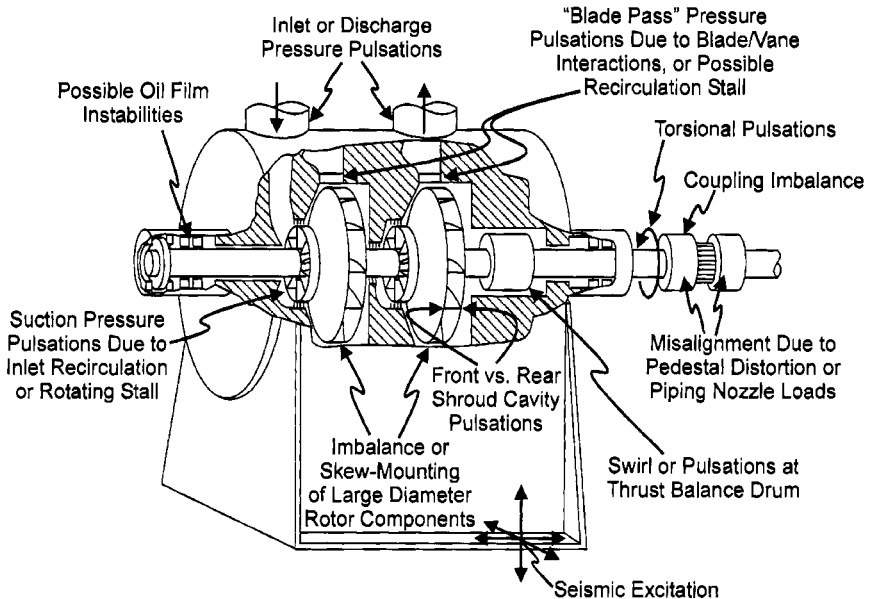


FIGURE 24 Typical frequencies associated with pump failure modes (Mechanical Solutions, Inc.)

CONCLUSION

Machinery issues such as the effects of nozzle loads and procedures for checking acceptability of vibration can seem deceptively simple. In reality, it takes education and experience to reach the correct conclusion relative to the many interrelated issues associated with choosing and operating a centrifugal pump.

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SECTION 2.2

CENTRIFUGAL PUMP CONSTRUCTION

2.2.1 CENTRIFUGAL PUMPS: MAJOR COMPONENTS

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CLASSIFICATION AND NOMENCLATURE

A centrifugal pump, stripped of all its refinements, has two main parts: (1) a rotating element, including an impeller and a shaft, and (2) a stationary element made up of a casing, casing cover, and bearings.

In a centrifugal pump, the liquid is forced by atmospheric or other pressure into a set of rotating vanes. These vanes constitute an impeller that discharges the liquid at its periphery at a higher velocity. This velocity is converted to pressure energy by means of a volute (see Figure 1) or by a set of stationary diffusion vanes (see Figure 2) surrounding the impeller periphery. Pumps with volute casings are generally called *volute pumps*, while those with diffusion vanes are called *diffuser pumps*. Diffuser pumps were once quite commonly called *turbine pumps*, but this term has become more selectively applied to the vertical deep-well centrifugal diffuser pumps usually referred to as *vertical turbine pumps*. Figure 1 shows the path of the liquid passing through an end-suction volute pump operating at rated capacity (the capacity at which best efficiency is obtained).

Impellers are classified according to the major direction of flow in reference to the axis of rotation. Thus, centrifugal pumps may have the following:

- Radial-flow impellers (see Figures 22, 30, 31, 32, and 33)
- Axial-flow impellers (see Figure 26)
- Mixed-flow impellers, which combine radial- and axial-flow principles (see Figures 24 and 25)

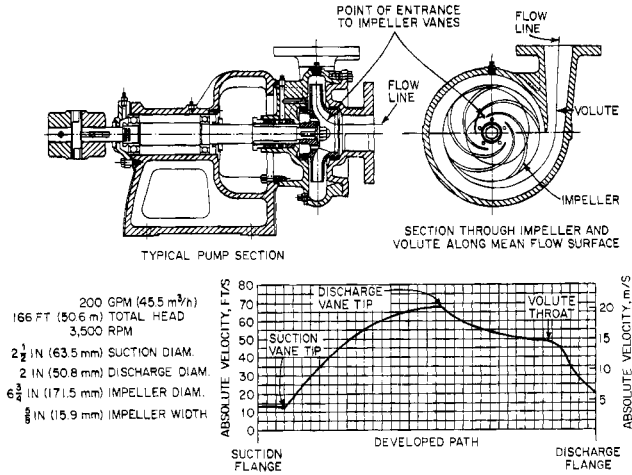


FIGURE 1 A typical single-stage, end-suction volute pump (Flowserve Corporation)

Impellers are further classified in one of two categories:

- Single-suction, with a single inlet on one side (see Figures 22 and 33)
- Double-suction, with liquid flowing to the impeller symmetrically from both sides (see Figures 23, 24, and 34)

The mechanical construction of the impellers gives a still further subdivision into

- Enclosed, with shrouds or side walls enclosing the waterways (see Figures 22, 23, and 24)
- Open, with no shrouds (see Figures 26 and 30)
- Semiopen or semienclosed (see Figure 32)

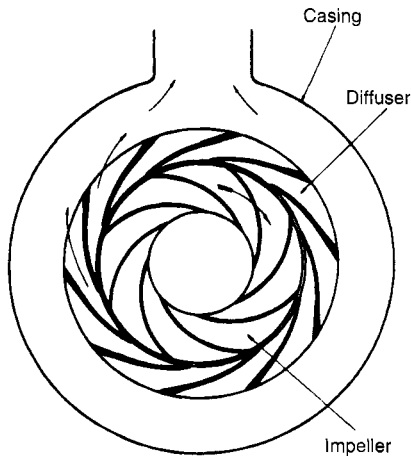


FIGURE 2 A typical diffuser-type pump

A pump in which the head is developed by a single impeller is called a *single-stage pump*. Often the total head to be developed requires the use of two or more impellers operating in a series, each taking its suction from the discharge of the preceding impeller. For this purpose, two or more single-stage pumps can be connected in a series or all the impellers can be incorporated in a single casing. The unit is then called a *multistage pump*.

The mechanical design of the casing provides the added pump classification of *axially split* or *radially split*, and the axis of rotation determines whether the pump is a horizontal or vertical unit.

Horizontal-shaft centrifugal pumps are classified still further according to the location of the suction nozzle:

- End-suction (see Figures 1, 8, 11, and 12)
- Side-suction (see Figures 7 and 10)
- Bottom-suction (see Figure 14)
- Top-suction (see Figure 21)

Some pumps operate in air with the liquid coming to and being conducted away from the pumps by piping. Other pumps, most often vertical types, are submerged in their suction supply. Vertical-shaft pumps are therefore called either *dry-pit* or *wet-pit* types. If the wet-pit pumps are axial-flow, mixed-flow, or vertical-turbine types, the liquid is discharged up through the supporting drop or column pipe to a discharge point above or below the supporting floor. These pumps are consequently designated as *aboveground discharge* or *belowground discharge units*.

Figures 3, 4, and 8 show typical constructions of a horizontal double-suction volute pump, the bowl section of a single-stage axial-flow propeller pump, and a vertical dry-pit single-suction volute pump, respectively. The names recommended by the Hydraulic Institute for various pump parts are given in Table 1.

CASINGS AND DIFFUSERS

The Volute Casing Pump This pump (refer to Figure 1) derives its name from the spiral-shaped casing surrounding the impeller. This casing section collects the liquid discharged by the impeller and converts velocity energy to pressure energy.

A centrifugal pump volute increases in area from its initial point until it encompasses the full 360° around the impeller and then flares out to the final discharge opening. The

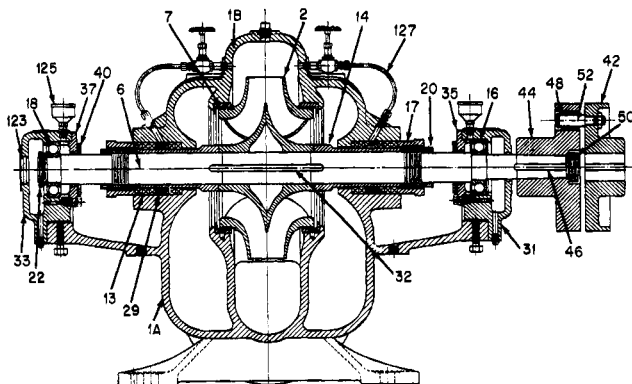


FIGURE 3 Horizontal, single-stage, double-suction volute pump (the numbers refer to parts listed in Table 1) (Flowserve Corporation)

TABLE 1 Recommended names of centrifugal pump parts^a

Item no.	Name of part	Item no.	Name of part
1	Gasing	33	Bearing housing (outboard)
1A	Gasing (lower half)	35	Bearing cover (inboard)
1B	Gasing (upper half)	36	Propeller key
2	Impeller	37	Bearing cover (outboard)
4	Propeller	39	Bearing bushing
6	Pump shaft	40	Deflector
7	Gasing ring	42	Coupling (driver half)
8	Impeller ring	44	Coupling (pump half)
9	Suction cover	46	Coupling key
11	Stuffing box cover	48	Coupling bushing
13	Packing	50	Coupling lock nut
14	Shaft sleeve	52	Coupling pin
15	Discharge bowl	59	Handhole cover
16	Bearing (inboard)	68	Shaft collar
17	Gland	72	Thrust collar
18	Bearing (outboard)	78	Bearing spacer
19	Frame	85	Shaft enclosing tube
20	Shaft sleeve nut	89	Seal
22	Bearing lock nut	91	Suction bowl
24	Impeller nut	101	Column pipe
25	Suction head ring	103	Connector bearing
27	Stuffing box cover ring	123	Bearing end cover
29	Lantern ring (seal cage)	125	Grease (oil) cup
31	Bearing housing (inboard)	127	Seal pipe (tubing)
32	Impeller key		

^aThese parts are called out in Figures 3, 4, and 8.

wall dividing the initial section and the discharge nozzle portion of the casing is called the tongue of the volute or the "cutwater." The diffusion vanes and concentric casing of a diffuser pump fulfill the same function as the volute casing in energy conversion.

In propeller and other pumps in which axial-flow impellers are used, it is not practical to use a volute casing. Instead, the impeller is enclosed in a pipe-like casing. Generally, diffuser vanes are used following the impeller proper, but in certain extremely low-head units these vanes may be omitted.

A diffuser is seldom applied to a single-stage, radial-flow pump, except in special instances where volute passages become so small that machined or precision-cast volute or diffuser-like pieces are utilized for precise flow control. Conventional diffusers are often applied to multistage pump designs in conjunction with guide vanes to direct the flow efficiently from one impeller (stage) to another in a minimum radial and axial space. Diffuser vanes are used as the primary construction method for vertical turbine pumps and single-stage, low-head propeller pumps (see Figure 4).

Radial Thrust In a *single-volute pump casing* design (see Figure 5), uniform or near uniform pressures act on the impeller when the pump operates at design capacity (which coincides with the best efficiency). At other capacities, the pressures around the impeller are not uniform (see Figure 6) and there is a resultant radial reaction (F). A detailed discussion of the radial thrust and of its magnitude is presented in Section 2.1.3. Note that the unbalanced radial thrust increases as capacity decreases from that at the design flow.

For any percentage of capacity, this radial reaction is a function of total head and of the width and diameter of the impeller. Thus, a high-head pump with a large impeller diameter will have a much greater radial reaction force at partial capacities than a low-head

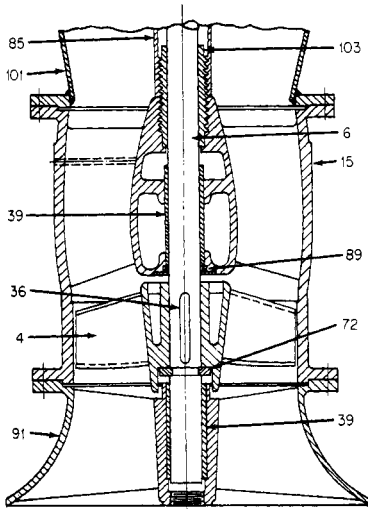


FIGURE 4 Vertical wet-pit diffuser pump bowl (the number refers to parts listed in Table 1) (Flowserve Corporation)

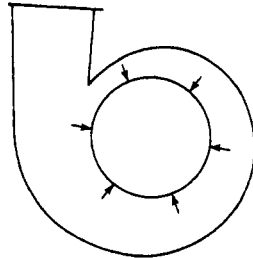


FIGURE 5 Uniform casing pressures exist at design capacity, resulting in zero radial reaction

pump with a small impeller diameter. A zero radial reaction is not often realized; the minimum reaction occurs at design capacity.

Although the same tendency for unbalance exists in the diffuser-type pump, the reaction is limited to a small arc repeated all around the impeller. As a result, the individual reactions cancel each other out as long as flow is constantly removed from around the periphery of the diffuser discharge. If flow is not removed uniformly around its periphery, a pressure imbalance may occur around the diffuser discharge that will be transmitted back through the diffuser to the impeller, resulting in a radial reaction on the shaft and bearing system.

In a centrifugal pump design, shaft diameter as well as bearing size can be affected by the allowable deflection as determined by the shaft span, impeller weight, radial reaction forces, and torque to be transmitted.

Formerly, standard designs compensated for radial reaction forces encountered at capacities in excess of 50% of the design capacity for the maximum-diameter impeller of the pump. For sustained operations at lower capacities, the pump manufacturer, if properly advised, would supply a heavier shaft, usually at a much higher cost. Sustained operations at extremely low flows without the manufacturer being informed at the time of purchase are a much more common practice today. This can result in broken shafts, especially older designs, on high-head units.

Because of the increasing application of pumps that must operate at reduced capacities, it has become desirable to design standard units to accommodate such conditions. One solution is to use heavier shafts and bearings. Except for low-head pumps in which only a small additional load is involved, this solution is not economical. The only practical answer is a casing design that develops a much smaller radial reaction force at partial capacities. One of these is the *double-volute* casing design, also called the *twin-volute* or *dual-volute* design.

The application of the double-volute design principle to neutralize radial reaction forces at reduced capacity is illustrated in Figure 7. Basically, this design consists of two 180° volutes, and a passage external to the second joins the two into a common discharge. Although a pressure unbalance exists at partial capacity through each 180° arc, the two

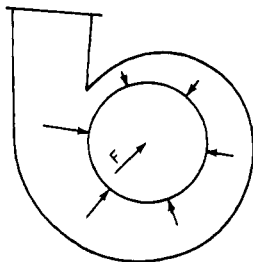


FIGURE 6 At reduced capacities, uniform pressures do not exist in a single-volute casing, resulting in a radial reaction F

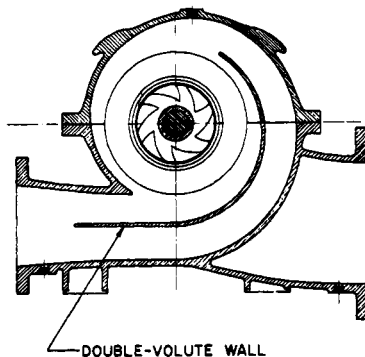


FIGURE 7 Transverse view of a double-volute casing pump

forces are approximately equal and opposite. Thus, little if any radial force acts on the shaft and bearings. Section 2.1.3 also covers this topic.

The double-volute design has many hidden advantages. For example, in large-capacity medium- and high-head single-stage vertical pumps, the rib forming the second volute that separates it from the discharge waterway of the first volute strengthens the casing (see Figure 8).

The individual stages of a multistage pump can be made double volute, as illustrated in Figure 9. The kinetic energy of the pumped liquid discharged from the impeller must be transformed into pressure energy and then must be turned 180° to enter the impeller of the next stage. The double volute therefore also acts as a return channel. The back view in Figure 9 shows this as well as the guide vanes used to straighten the flow into the next stage. An alternative to the volute design for multistage pumps is the diffuser and its return vanes that channel the flow from the discharge of the diffuser vanes back into the impeller of the next stage.

Solid and Split Casings *Solid casing* implies a design in which the discharge waterways leading to the discharge nozzle are all contained in one casting or fabricated piece. The casing must have one side open so that the impeller can be introduced into it. Because the sidewalls surrounding the impeller are actually part of the casing, a solid casing, strictly speaking, cannot be used, and designs normally called solid casing are really radially split (refer to Figure 1 and see Figures 11 and 12).

A *split casing* is made of two or more parts fastened together. The term *horizontally split* had regularly been used to describe pumps with a casing divided by a horizontal plane through the shaft centerline or axis (see Figure 10). The term *axially split* is now preferred. Because both the suction and discharge nozzles are usually in the same half of the casing, the other half may be removed for inspection of the interior without disturbing the bearings or the piping. Like its counterpart horizontally split, the term *vertically split* is poor terminology. It refers to a casing split in a plane perpendicular to the axis of rotation. The term *radially split* is now preferred.

End-Suction Pumps Most end-suction, single-stage pumps are made of one-piece solid casings. At least one side of the casing must be open so that the impeller can be assembled in the pump. Thus, a cover is required for that side. If the cover is on the suction side, it becomes the casing sidewall and contains the suction opening (refer to Figure 1). This is called the *suction cover* or *casing suction head*. Other designs are made with casing covers (see Figure 11) and still others have both casing suction covers and casing covers (refer to Figure 8 and see Figure 12).

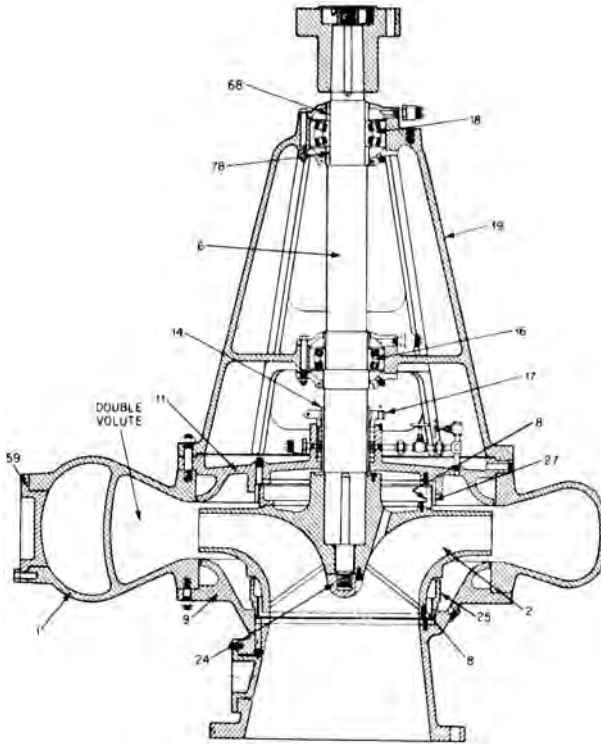


FIGURE 8 The sectional view of a vertical-shaft, end-suction pump with a double-volute casing (the numbers refer to parts listed in Table 1) (Flowserve Corporation)

For general service, the end-suction, single-stage pump design is extensively used for small pumps with a 4- or 6-in (102- or 152-mm) discharge size for both motor-mounted and coupled types. In these pumps, the small size makes it feasible to cast the volute and one side integrally. Whether or not the seal chamber side or the suction side is made integrally with the casing is usually determined by the most economical pump design.

For larger pumps, especially those for special services such as sewage handling, there is a demand for pumps of both rotations. A design with separate suction and seal chamber heads permits the use of the same casing for either rotation if the flanges on the two sides



FIGURE 9 The double volute of a multistage pump, front view (left) and back view (right) (Flowserve Corporation)



FIGURE 10 An axially split casing, horizontal-shaft, double-suction volute pump (Flowserve Corporation)

are made identical. There is also a demand for vertical pumps that can be disassembled by removing the rotor and bearing assembly from the top of the casing. Many horizontal applications of the pumps of the same line, however, require partial dismantling from the suction side. Such lines are most adaptable when they have separate suction and casing covers.

Casing Construction for Open- and Semiopen-Impeller Pumps In the open- or semiopen impeller pump, the impeller rotates within close clearance of the pump casing or suction cover (refer to Figure 11). If the intended service is abrasive, a side plate is mounted within the casing to provide a renewable close-clearance guide to the liquid



FIGURE 11 End-suction pump with semiopen impeller (Flowserve Corporation)

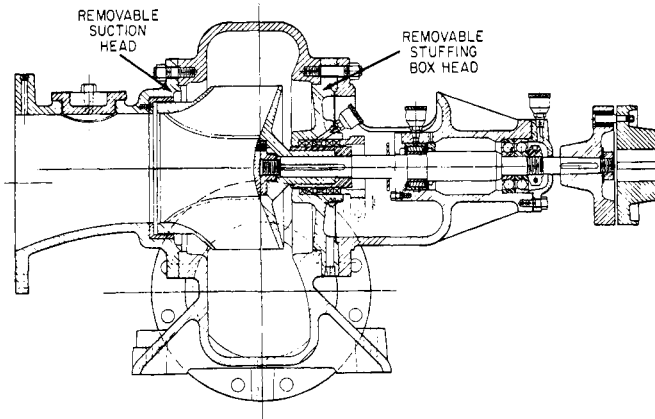


FIGURE 12 End-suction pump with removable suction and stuffing box heads (Flowserve Corporation)

flowing through the impeller. One of the advantages of using side plates is that abrasion-resistant material, such as stainless steel, can be used for the impeller and side plate, while the casing itself may be of a less costly material. Although double-suction, semiopen-impeller pumps are seldom used today, they were common in the past and were generally made with side plates.

In order to maintain pump efficiency, a close running clearance is required between the front unshrouded face of the open or semiopen impeller and the casing, suction cover, or side plate. Pump designs provide either jackscrews or shims to adjust the position of the thrust bearing housing (and, as a result, the axial position of the shaft and impeller) relative to the bearing frame.

Pre-rotation and Stop Pieces Improper entrance conditions and inadequate suction approach shapes may cause the liquid column in the suction pipe to spiral for some distance ahead of the impeller entrance. This phenomenon is called *pre-rotation*, and it is attributed to various operational and design factors in both vertical and horizontal pumps.

Pre-rotation is usually harmful to pump operation because the liquid enters between the impeller vanes at an angle other than that allowed in the design. This frequently lowers the net effective suction head and the pump efficiency. Various means are used to avoid pre-rotation both in the construction of the pump and in the design of the suction approaches.

Practically all horizontal, single-stage, double-suction pumps and most multistage pumps have a suction volute that guides the liquid in a streamline flow to the impeller eye. The flow comes to the eye at right angles to the shaft and separates unequally on the two sides of the shaft. Moving from the suction nozzle to the impeller eye, the suction waterways are reduced in area, meeting in a projecting section of the sidewall dividing the two sections. This dividing projection is called a *stop piece*. To minimize pre-rotation in end-suction pumps, a radial-fin stop piece projecting toward the center is sometimes cast into the suction nozzle wall.

Nozzle Locations The discharge nozzle of end-suction, single-stage horizontal pumps is usually in a top-vertical position (refer to Figures 1 and 11). However, other nozzle positions can be obtained, such as top-horizontal, bottom-horizontal, or bottom-vertical. Figure 13 illustrates the flexibility available in discharge nozzle locations. Sometimes the pump frame, bearing bracket, or baseplate may interfere with the discharge flange, prohibiting a bottom-horizontal or bottom-vertical discharge nozzle position. In other instances, solid casings cannot be rotated for various nozzle positions because the seal chamber connection would become inaccessible.

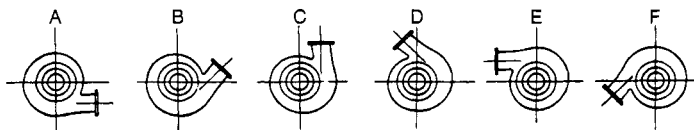


FIGURE 13 Possible positions of discharge nozzles for a specific design of an end-suction, solid-casing, horizontal-shaft pump. The rotation illustrated is counterclockwise from suction end.

Practically all double-suction, axially split casing pumps have a side discharge nozzle and either a side- or a bottom-suction nozzle. If the suction nozzle is placed on the side of the pump casing with its axial centerline (refer to Figure 10), the pump is classified as a *side-suction pump*. If its suction nozzle points vertically downward (see Figure 14), the pump is called a *bottom-suction pump*. Single-stage, bottom-suction pumps are rarely made in sizes below a 10-in (254-mm) discharge nozzle diameter.

Special nozzle positions can sometimes be provided for double-suction, axially split casing pumps to meet special piping arrangements, such as a radically split casing with bottom suction and top discharge in the same half of the casing. Such special designs are generally costly and should be avoided.

Centrifugal Pump Rotation Because suction and discharge nozzle locations are affected by pump rotation, it is important to understand how the direction of rotation is defined. According to Hydraulic Institute standards, rotation is defined as clockwise or counterclockwise by looking at the driven end of a horizontal pump or looking down on a vertical pump. To avoid misunderstanding, clockwise or counterclockwise rotation should always be qualified by including the direction from which one looks at the pump.

The terms *inboard end* or *drive end* (the end of the pump closest to the driver) and *out-board end* or *nondrive end* (the end of the pump farthest from the driver) are used only with horizontal pumps. The terms lose their significance with dual-driven pumps. Any centrifugal pump casing pattern can be arranged for either clockwise or counterclockwise rotation, except for end-suction pumps, which have integral heads on one side. These require separate directional patterns.



FIGURE 14 A bottom-suction, axially split casing single-stage pump (Flowserve Corporation)

Casing Handholes Casing handholes are furnished primarily on pumps handling sewage and stringy materials that may become lodged on the impeller suction vane edges or on the tongue of the volute. The holes permit removal of this material without completely dismantling the pump. End-suction pumps used primarily for liquids of this type are provided with handholes or access to the suction side of the impellers. These access points are located on the suction head or in the suction elbow. Handholes are also provided in drainage, irrigation, circulating, and supply pumps if foreign matter may become lodged in the waterways. On very large pumps, manholes provide access to the interior for both cleaning and inspection.

Mechanical Features of Casings Most single-stage centrifugal pumps are intended for service at moderate pressures and temperatures. As a result, pump manufacturers usually design a special line of pumps for high operating pressures and temperatures rather than make their standard line unduly expensive by making it suitable for too wide a range of operating conditions.

If axially split casings are subject to high pressure, they tend to “breathe” at the split joint, leading to misalignment of the rotor and, even worse, leakage. For such conditions, internal and external ribbing is applied to casings at the points subject to the greatest stress. In addition, whereas most pumps are supported by feet at the bottom of the casing, high temperatures require centerline support so that, as the pump becomes heated, expansion will not cause misalignment.

Series Units For large-capacity medium-high-head conditions, two single-stage, double-suction pumps can be connected in a series on one baseplate with a single driver. Such an arrangement was at one time very common in waterworks applications for heads of 250 to 400 ft (76 to 122 m). One series arrangement uses a double-extended shaft motor in the middle, driving two pumps connected in a series by external piping. In a second type, a standard motor is used with one pump having a double-extended shaft. This latter arrangement may have limited applications because the shaft of the pump next to the motor must be strong enough to transmit the total pumping horsepower. If the total pressure generated by such a series unit is relatively high, the casing of the second pump could require ribbing. Higher heads per stage are becoming more and more common, and series units are generally used in only very high ranges of total head.

MULTISTAGE PUMP CASINGS

Although the majority of single-stage pumps are of the volute-casing type, both volute and diffuser casings are used in multistage pump construction. Because a volute casing gives rise to radial thrust, axially split multistage casings generally have staggered volutes so that the resultant of the individual radial thrusts is balanced out (see Figures 15 and 16). Both axially and radially split casings are used for multistage pumps. The choice between the two designs is dictated by the design pressure, with 2000 lb/in² (138 bar)^{*} being typical for 3600-rpm axially split casing pumps.

Axially Split Casings Regardless of the arrangement of the stages in the casing, it is necessary to connect the successive stages of a multistage pump. In the low and medium pressure and capacity ranges, these interstage passages are cast integrally with the casing proper (see Figures 17 and 18). As the pressures and capacities increase, the desire to maintain as small a casing diameter as possible, coupled with the necessity of avoiding sudden changes in the velocity or the direction of the flow, leads to the use of external interstage passages cast separately from the pump casing. They are formed in the shape of a loop, bolted or welded to the casing proper (see Figure 19).

Interstage Construction for Axially Split Casing Pumps A multistage pump inherently has adjoining chambers subjected to different pressures, and means must be made

^{*}1 bar = 10⁵ Pa

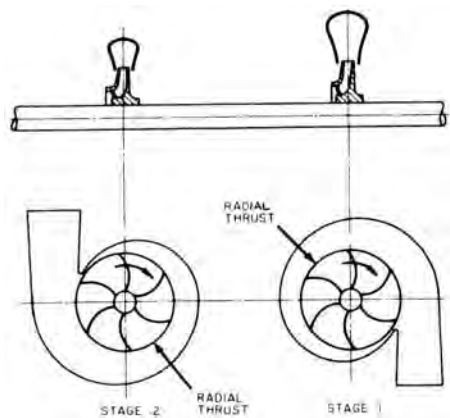


FIGURE 15 An arrangement of a multistage volute pump for radial-thrust balance

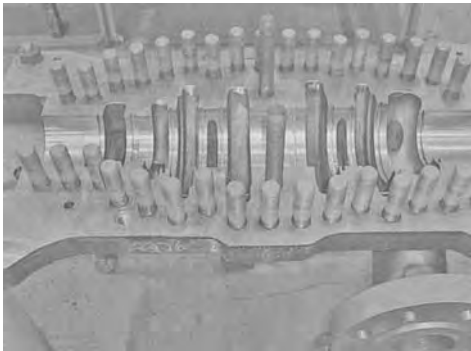


FIGURE 16 The horizontal flange of an axially split casing pump (Flowserve Corporation)

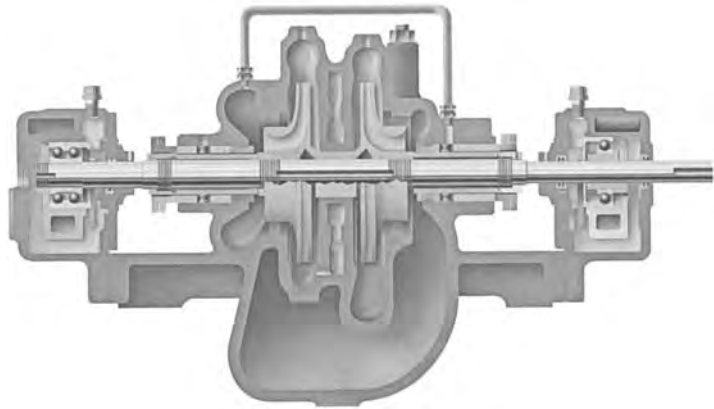


FIGURE 17 Two-stage axially split casing volute pump for small capacities and pressures up to 450 lb/in² (31 bar) (Flowserve Corporation)



FIGURE 18 Two-stage axially split casing volute pump for pressures up to 500 lb/in² (34 bar) (Flowserve Corporation)

available to isolate these chambers from one another so that the leakage from high to low pressure will take place only at the clearance joints formed between the stationary and rotating elements of the pump. Thus, the leakage will be kept to a minimum. The isolating wall used to separate two adjacent chambers of a multistage pump is called a *stage piece*, *diaphragm*, or *interstage diaphragm*. The stage piece may be formed of a single piece, or it may be fitted with a renewable stage piece bushing at the clearance joint between the stationary stage piece and the part of the rotor immediately inside the former. The stage pieces, which are usually solid, are assembled on the rotor along with impellers, sleeves, bearings, and similar components. To prevent the stage pieces from rotating, a locked tongue-and-groove joint is provided in the lower half of the casing. Clamping the upper casing half to the lower half securely holds the stage piece and prevents rotation.

The problem of seating a solid stage piece against an axially split casing is one that has given designers much trouble. First, there is a three-way joint and, second, this seating must make the joint tight and leakproof under a pressure differential without resorting to bolting the stage piece directly to the casing.

To overcome this problem, it is wise to make a pump that has a small-diameter casing so that when the casing bolting is pulled tight, there is a seal fitting of the two casing halves adjacent to the stage piece. The small diameter likewise helps to eliminate the possibility of a stage piece cocking and thereby leaving a clearance on the upper-half casing when it is pulled down. No matter how rigidly the stage piece is located in the lower-half casing, there must be a sliding fit between the seat face of the stage piece and that in the upper-half casing so that the upper-half casing can be pulled down. Each stage piece,



FIGURE 19 A two-stage double-suction, axially split volute pump with an external interstage passage bolted to the casing (Flowserve Corporation)

furthermore, must be arranged so that the pressure differential developed by the pump will tend to seat the piece tightly against the casing rather than open up the joint.

We have said that axially split casing pumps are typically used for working pressures of up to 2000 lb/in² (138 bar). High-pressure piping systems, of which these pumps form a part, are inevitably made of steel because this material has the valuable property of yielding without breaking. Considerable piping strains are unavoidable, and these strains, or at least a part thereof, are transmitted to the pump casing. The latter consists essentially of a barrel that is split axially, flanged at the split, and fitted with two necks that serve as inlet and discharge openings. When piping stresses exist, these necks, being the weakest part of the casing, are in danger of breaking off if they cannot yield. Steel is therefore the safest material for pump casings whenever the working pressures in the pump are in excess of 1000 lb/in² (70 bar).

This brings up an important feature in the design of the suction and discharge flanges. Although raised-face flanges are perfectly satisfactory for steel-casing pumps, their use is extremely dangerous with cast-iron pumps. This danger arises from the lack of elasticity in cast iron, which leads to flange breakage when the bolts are being tightened, the fulcrum of the bending moment being located inward from the bolt circle. As a result, it is essential to avoid raised-face flanges with cast-iron casings as well as the use of a raised-face flange pipe directly against a flat-face cast-iron pump flange. Suction flanges should obviously be suitable for whatever hydrostatic test pressure is applied to the pump casing.

The location of the pump casing feet is not critical in smaller pumps operating at discharge pressures below 275 lb/in² (19 bar) and at moderate temperatures of up to 300°F (150°C). Since the unit is relatively small, very little distortion is likely to occur. However, for larger units operating at higher pressures and temperatures, it is important that the casing be supported at the horizontal centerline or immediately below the bearings (refer to Figure 19).

Radially Split Double-Casing Pumps The oldest form of radially split casing multi-stage pump is commonly called the *ring-section*, *ring-casing*, or the *doughnut type*. When more than one stage was found necessary to generate higher pressures, two or more single-stage units of the prevalent radially split casing type were assembled and bolted together.

In later designs, the individual stage sections and separate suction and discharge heads were held together with large throughbolts. These pumps, still an assembly of bolted-up sections, can present serious dismantling and reassembly problems because suction and discharge connections have to be broken each time the pump is serviced. The double-casing pump retains the advantages of the radially split casing design and minimizes the dismantling problem.

The basic principle consists of enclosing the working parts of a multistage centrifugal pump in an inner casing and building a second casing around this inner casing. The space between the two casings is maintained at the discharge pressure of the last pump stage. The construction of the inner casing follows one of two basic principles: (1) axial splitting (see Figure 20) or (2) radial splitting (see Figure 21).

The double-casing pump with radially split inner casing is an evolution of the ring-casing pump with added provisions to ease dismantling. The inner unit is generally constructed exactly as a ring-casing pump. After assembly, it is inserted and bolted inside a cylindrical casing that supports it and leaves it free to expand under temperature changes. The suction and discharge nozzles form an integral part of the outer casing, and the internal assembly of the pump can be withdrawn without disturbing the piping connections.

Hydrostatic Pressure Tests It is standard practice for the manufacturer to conduct hydrostatic tests for the parts of a pump that contain fluid under pressure. This means that the pump casing and, where applicable, parts like suction or casing covers are assembled with the internal parts, removed, and are then subjected to a hydrostatic test,

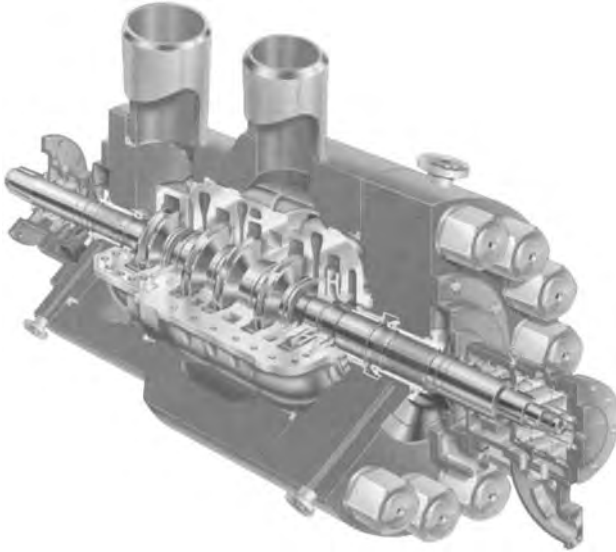


FIGURE 20 Double-casing multistage pump with axially split inner casing (Flowserve Corporation)

generally for a minimum of 30 min. Such a test demonstrates that the casing containment is sound and that there is no leakage of fluid to the exterior.

Additional tests may be conducted by the pump manufacturer to determine the soundness of internal partitions separating areas of the pump operating under different pressures.

The definition of the applicable hydrostatic pressure for these tests varies. The most generally accepted definition is that given by the Hydraulic Institute Standards. Each part

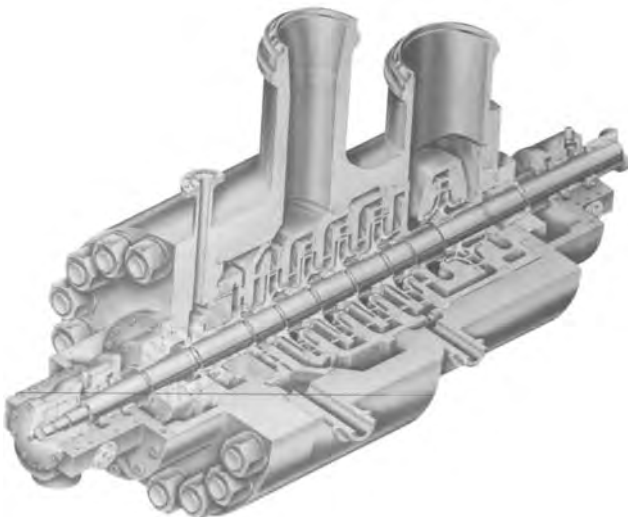


FIGURE 21 Double-casing multistage pump with radially split inner casing (Flowserve Corporation)

of the pump that contains fluid under pressure shall be capable of withstanding a hydrostatic test at not less than the greatest of the following:

- 150% of the pressure that will occur in that part when the pump is operated at rated conditions for the given application of the pump, except thermoset parts
- 125% of the pressure that would occur in that part when the pump is operating at rated speed for a given application, but with the pump discharge valve closed

IMPELLERS

In a single-suction impeller, the liquid enters the suction eye on one side only. A double-suction impeller is, in effect, two single-suction impellers arranged back to back in a single casing. The liquid enters the impeller simultaneously from both sides, while the two casing suction passageways are connected to a common suction passage and a single suction nozzle.

For the general service single-stage, axially split casing design, a double-suction impeller is favored because it is theoretically in an axial hydraulic balance and because the greater suction area of a double-suction impeller permits the pump to operate with less net absolute suction head. For small units, the single-suction impeller is more practical for manufacturing reasons, as the waterways are not divided into two very narrow passages. It is also sometimes preferred for structural reasons. End-suction pumps with single-suction overhung impellers have both first-cost and maintenance advantages unobtainable with double-suction impellers. Most radially split casing pumps therefore use single-suction impellers. Because an overhung impeller does not require the extension of a shaft into the impeller suction eye, single-suction impellers are preferred for pumps handling suspended matter, such as sewage. In multistage pumps, single-suction impellers are almost universally used because of the design and first-cost complexity that double-suction staging introduces.

Impellers are called radial vane or radial flow when the liquid pumped is made to discharge radially to the periphery. Impellers of this type usually have a specific speed below 4200 (2600) if single-suction and below 6000 (3700) if double-suction. Specific speed is discussed in detail in Section 2.1.3. The units for specific speed used here are in USCS rpm, gallons per minute, and feet. In SI, they are rpm, liters per second, and meters. Impellers can also be classified by the shape and form of their vanes:

- The straight-vane impeller (see Figures 22, 30, 31, 32, and 33)
- The Francis-vane or screw-vane impeller (see Figures 23 and 24)
- The mixed-flow impeller (see Figure 25)
- The propeller or axial-flow impeller (see Figure 26)

In a *straight-vane radial impeller*, the vane surfaces are generated by straight lines parallel to the axis of rotation. These are also called *single-curvature vanes*. The vane surfaces of a *Francis-vane radial impeller* have a double curvature. An impeller design that has both a radial-flow and an axial-flow component is called a *mixed-flow impeller*. It is generally restricted to single-suction designs with a specific speed above 4200 (2600). Types with lower specific speeds are called *Francis-vane impellers*. Mixed-flow impellers with a small radial-flow component are usually referred to as propellers. In a true propeller, or *axial-flow impeller*, the flow strictly parallels the axis of rotation. In other words, it moves only axially.

An inducer is a low-head, axial-flow impeller with few blades that is placed in front of a conventional impeller. The hydraulic characteristics of an inducer are such that it requires considerably less NPSH than a conventional impeller. Both inducer and impeller are mounted on the same shaft and rotate at the same speed (see Figure 27). The main purpose of the inducer is not to generate an appreciable portion of the total pump head, but to increase the suction pressure to a conventional impeller. Inducers are therefore used



FIGURE 22 Straight-vane, radial, single-suction closed impeller (Flowserve Corporation)



FIGURE 23 Francis vane, radial, closed impeller (Flowserve Corporation)



FIGURE 24 Medium specific speed, Francis vane radial impeller (Flowserve Corporation)



FIGURE 25 Open mixed-flow impeller (Flowserve Corporation)



FIGURE 26 Axial-flow impeller (Flowserve Corporation)



FIGURE 27 A centrifugal pump with a conventional impeller preceded by an inducer (Flowserve Corporation)

to reduce the NPSH requirements of a given pump or to permit the pump to operate at higher speeds with a given available NPSH. For a further discussion of inducers, see Section 2.1 and Section 2.1.3.

The relation of single-suction impeller profiles to specific speed is shown in Figure 28. The classification of impellers according to their vane shape is naturally arbitrary inasmuch as there is much overlapping in the types of impellers used in the different types of pumps. For example, impellers in single- and double-suction pumps of low specific speeds have vanes extending across the suction eye. This provides a mixed flow at the impeller entrance for low pickup losses at high rotative speeds but enables the discharge portion of the impeller to use the straight-vane principle. In pumps of higher specific speed operating against low heads, impellers have double-curvature vanes extending over the full vane surface. They are therefore full Francis-type impellers. The mixed-flow impeller, usually a single-suction type, is essentially one-half of a double-suction, high-specific-speed, Francis-vane impeller.

In addition, many impellers are designed for specific applications. For instance, the conventional impeller design with sharp vane edges and restricted areas is not suitable for handling liquids containing rags, stringy materials, and solids like sewage because it will become clogged. Special nonclogging impellers with blunt edges and large waterways have been developed for such services (see Figure 29). For pumps up to the 12- to 16-in (305- to 406-mm) discharge size, these impellers have only two vanes. Larger pumps normally use three or four vanes.

Impeller Mechanical Types Mechanical design also determines impeller classification. Accordingly, impellers may be completely open, semiopen, or closed.

Strictly speaking, an *open impeller* (see Figure 28) consists of nothing but vanes attached to a central hub for mounting on the shaft without any form of sidewall or shroud. The disadvantage of this impeller is structural weakness. If the vanes are long, they must be strengthened by ribs or a partial shroud. Generally, open impellers are used in small, low-energy pumps. One advantage of open impellers is that they are better suited for handling liquids containing stringy materials. It is also sometimes claimed that they are better suited for handling liquids containing suspended matter because the solids in such matter are more likely to be clogged in the space between the rotating shrouds of a closed impeller and the stationary casing walls. It has been demonstrated, however, that closed impellers do not clog easily, thus disproving the claim for the superiority of the open-impeller design. In addition, the open impeller is much more sensitive to wear than the closed impeller and therefore its efficiency may deteriorate rather rapidly.

The open impeller rotates between two side plates, between the casing walls of the volute, or between the casing cover and the suction head. The clearance between the

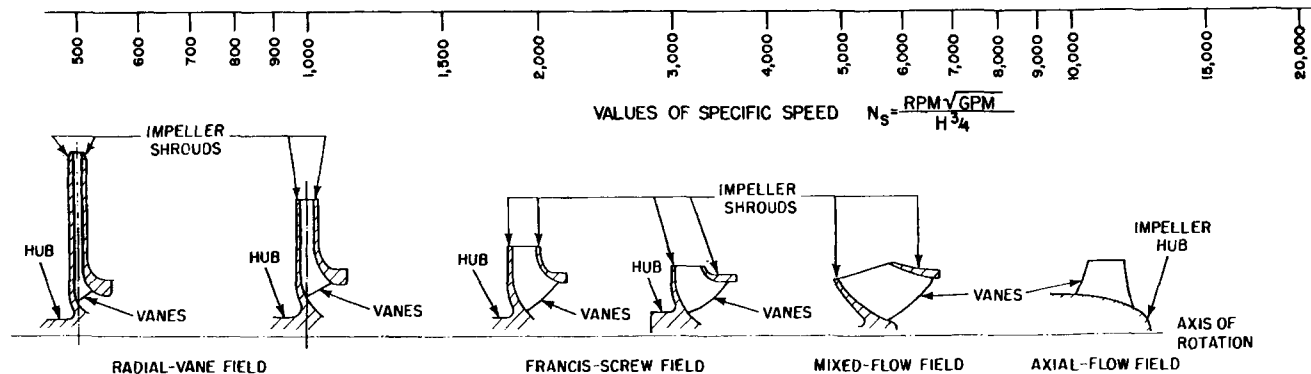


FIGURE 28 Variations in impeller profiles with specific speeds and approximate ranges of specific speeds for the various types [Universal specific speed $\Omega_s = N_s / 2733$. N_q (in rpm, m³/s, m) = $N_s / 51.65$]



FIGURE 29 Phantom view of a radial-vane nonclogging impeller (Flowserve Corporation)



FIGURE 30 Barske-type radial multiple-vane open impeller with multiple balance holes to reduce axial pressure unbalance (axial thrust) (Flowserve Corporation)

impeller vanes and the sidewalls enables a certain amount of water slippage. This slippage increases as wear increases. To restore the original efficiency, both the impeller and the side plate(s) must be replaced. This, incidentally, involves a much larger expense than would be entailed in closed impeller pumps where simple rings form the leakage joint.

The *semiopen impeller* (see Figures 30 through 33) incorporates a single shroud, usually at the back of the impeller. This shroud may or may not have *pump-out vanes*, which are vanes located at the back of the impeller shroud. This function reduces the pressure at the back hub of the impeller and prevents foreign matter from lodging in back of the impeller that would interfere with the proper operation of the pump and the seal chamber. The back shrouds may also be scalloped or relieved with balance holes through them (see Figure 33) to reduce axial unbalance.

The *closed impeller* (refer to Figures 22 through 24), which is almost universally used in centrifugal pumps handling clear liquids, incorporates shrouds or sidewalls that totally enclose the impeller waterways from the suction eye to the periphery. Although this design prevents the liquid slippage that occurs between an open or semiopen impeller and its side plates, a running joint must be provided between the impeller and the casing to separate the discharge and suction chambers of the pump. This running joint is usually formed by a relatively short cylindrical surface on the impeller shroud that rotates within a slightly larger stationary cylindrical surface. If one or both surfaces are made renewable, the leakage joint can be repaired when wear causes excessive leakage.

If the pump shaft terminates at the impeller so that the latter is supported by bearings on one side, the impeller is called an *overhung impeller*. This type of construction is the best for end-suction pumps with single-suction impellers.

Impeller Nomenclature The inlet of an impeller just before the section where the vanes begin is called the suction eye (see Figure 34). In a closed-impeller pump, the suction eye



FIGURE 31 Radial vane open impeller (Flowserve Corporation)



FIGURE 32 Open impeller with scalloped back shroud walls to reduce axial pressure unbalance (Flowserve Corporation)

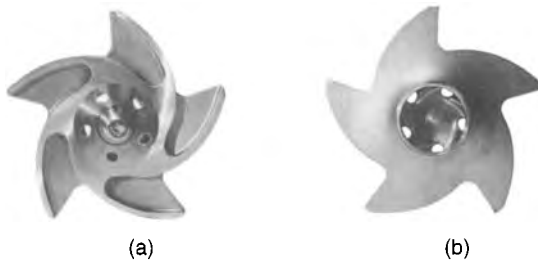


FIGURE 33 Semiopen reversed vane impeller with partially scalloped shroud - (a) back side has close running clearance to wear plate; (b) front side, showing entrance to back side, vane inlet edges and balance holes (Flowserve Corporation)

diameter is taken as the smallest inside diameter of the shroud. In determining the area of the suction eye, the area occupied by the impeller shaft hub is deducted.

The *hub* is the central part of the impeller that is bored out to receive the pump shaft. The term, however, is also frequently used for the part of the impeller that rotates in the casing fit or in the casing wearing ring. It is then referred to as the *outer impeller hub* or the *wearing-ring hub* of the shroud.

WEARING RINGS

Wearing rings provide an easily and economically renewable leakage joint between the impeller and the casing. A leakage joint without renewable parts is illustrated in Figure 39. To restore the original clearances of such a joint after wear occurs, the user must either (1) build up the worn surfaces by welding or metal spraying, or (2) buy new parts.

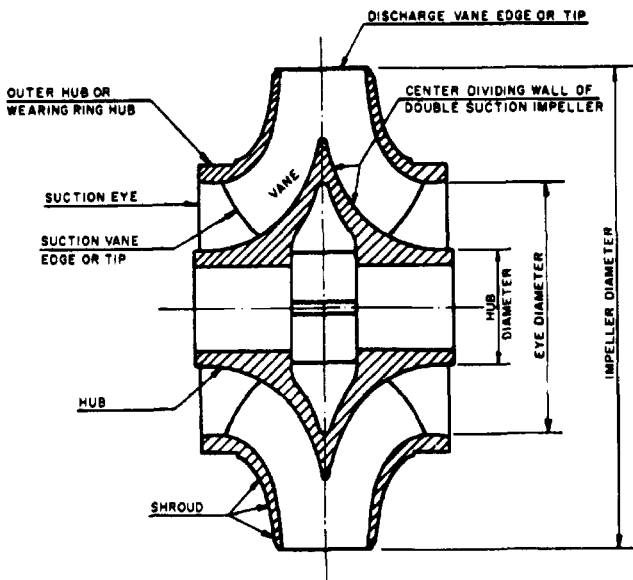


FIGURE 34 Parts of a double-suction impeller

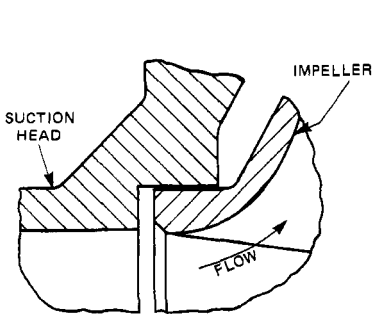


FIGURE 35 A plain flat leakage joint with no rings

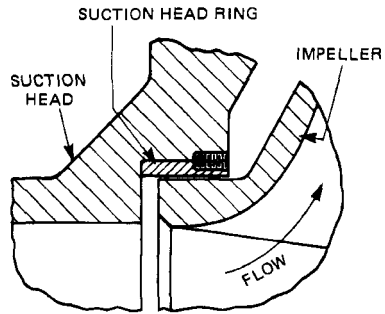


FIGURE 36 A single flat casing ring construction

The new parts are not very costly in small pumps, especially if the stationary casing element is a simple suction cover. This is not true for larger pumps or where the stationary element of the leakage joint is part of a complicated casting. If the first cost of a pump is of prime importance, it is more economical to provide for remachining both the stationary parts and the impeller. Renewable casing and impeller rings can then be installed (see Figures 36, 37, and 38). The nomenclature for the casing or stationary part forming the leakage joint surface is as follows: (1) *casing ring* (if mounted in the casing), (2) *casing ring* or *suction head ring* (if mounted in a suction cover or head), and (3) *casing cover ring* or *head ring* (if mounted in the casing cover or head). Some engineers like to identify the part further by adding the word *wearing*, such as *casing wearing ring*. A renewable part for the impeller wearing surface is called the *impeller ring*. Pumps with both stationary and rotating rings are said to have *double-ring construction*.

Wearing Ring Types Various types of wearing-ring designs, and the selection of the most desirable type depends on the liquid being handled, the pressure differential across the leakage joint, the rubbing speed, and the particular pump design. In general, centrifugal pump designers use the ring construction that they have found to be most suitable for each particular pump service.

The most common ring constructions are the flat type (see Figures 36 and 37) and the L type. The leakage joint in the former is a straight, annular clearance. In the L-type ring (see Figure 39), the axial clearance between the impeller and the casing ring is large, so the velocity of the liquid flowing into the stream entering the suction eye of the impeller is low. The L-type casing rings shown in Figures 39 and 40 have the additional function of guiding the liquid into the impeller eye; they are called *nozzle rings*. Impeller rings of the L type shown in Figure 40 also furnish protection for the face of the impeller wearing-ring hub.

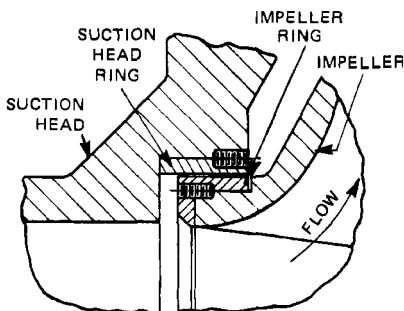


FIGURE 37 A double flat ring construction

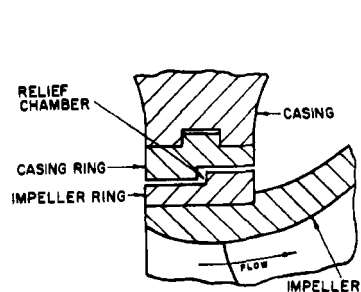


FIGURE 38 A step-type leakage joint with double rings

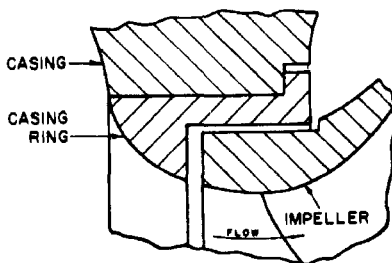


FIGURE 39 An L-type nozzle casing ring

Some designers favor labyrinth-type rings (see Figures 41 and 42) that have two or more annular leakage joints connected by relief chambers. In leakage joints involving a single unbroken path, the flow is a function of both the area and the length of the joint as well as of the pressure differential across the joint. If the path is broken by relief chambers (Figures 38, 41, and 42), the velocity energy in the jet is dissipated in each relief chamber, increasing the resistance. As a result, with several relief chambers and several leakage joints for the same actual flow through the joint, the area and hence the clearance between the rings can be greater than for an unbroken, shorter leakage joint.

The single labyrinth ring with only one relief chamber (refer to Figure 41) is often called an *intermeshing ring*. The *step-ring type* (refer to Figure 38) utilizes two flat-ring elements of slightly different diameters over the total leakage joint width with a relief chamber between the two elements. Other ring designs also use some form of relief chamber. For example, one commonly used in small pumps has a flat joint similar to that in Figure 40, but with one surface broken by a number of grooves. These act as relief chambers to dissipate the jet velocity head, thereby increasing the resistance through the joint and decreasing the leakage.

For raw water pumps in waterworks service and for larger pumps in sewage services in which the liquid contains sand and grit, *water-flushed rings* have been used (see Figure 43). Clear water under a pressure greater than that on the discharge side of the rings is piped to the inlet and distributed by the cored passage, the holes through the stationary ring, and the groove to the leakage joint. Ideally, the clear water should fill the leakage joint with some flow to the suction and discharge sides to prevent any sand or grit from getting into the clearance space. Wearing-ring flush is also employed in some process pumps when pumping solids or abrasives to minimize ring wear by injecting a compatible clean liquid between the rings.

In large pumps (with roughly a 36-in [900-mm] or larger discharge size), particularly vertical, end-suction, single-stage volute pumps, size alone permits some refinements not found in smaller pumps. One example is the inclusion of inspection ports for measuring ring clearance (see Figure 44). These ports can be used to check the impeller centering after the original installation as well as to observe ring wear without dismantling the pump.

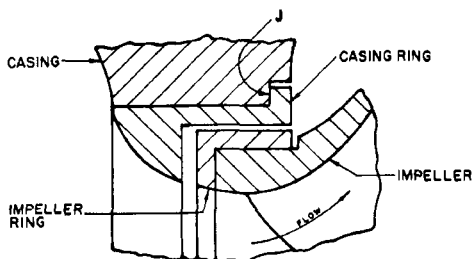


FIGURE 40 Double rings, both of L type

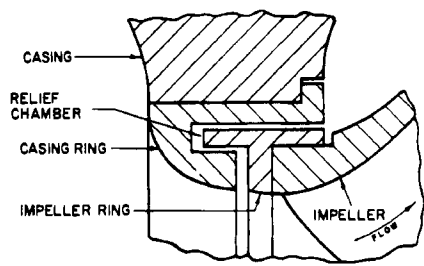


FIGURE 41 A single-labyrinth, intermeshing type. It is a double-ring construction with a nozzle-type casing ring.

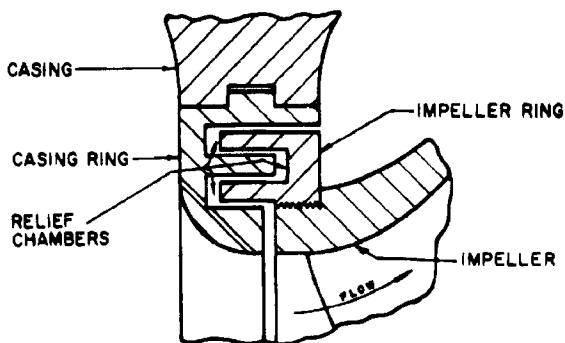


FIGURE 42 Labyrinth-type rings in a double-ring construction

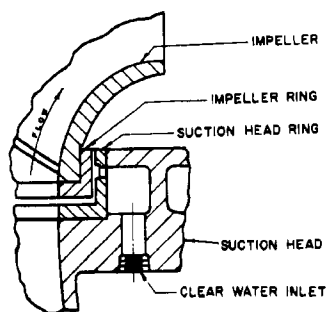


FIGURE 43 Water-flushed wearing ring

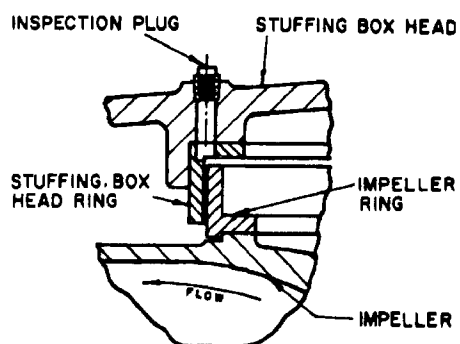


FIGURE 44 Wearing ring with an inspection port for checking clearance

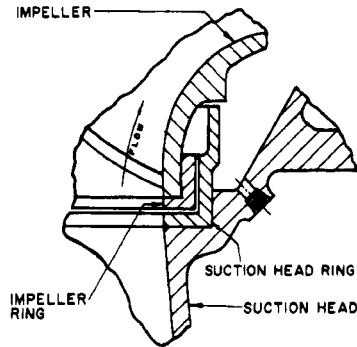


FIGURE 45 Dam-type ring construction

The lower rings of large vertical pumps handling liquids containing sand and grit in intermittent services are highly susceptible to wear. During shutdown periods, the grit and sand settle out and naturally accumulate in the region where these rings are installed, as it is the lowest point on the discharge side of the pump. When the pump is started again, this foreign matter is washed into the joint all at once and causes wear. To prevent this action in medium and large pumps, a dam-type ring is often used (see Figure 49). Periodically, the pocket on the discharge side of the dam can be flushed out.

One problem with the simple water-flushed ring is the failure to get uniform pressure in the stationary ring groove. If pump size and design permit, two sets of wearing rings arranged in tandem and separated by a large water space (see Figure 46) provide the best solution. The large water space enables a uniform distribution of the flushing water to the full 360° of each leakage joint. Because ring 2 is shorter and because a greater clearance is used there than at ring 1, equal flow can take place to the discharge pressure side and to the suction pressure side. This design also makes it easier to harden or coat the surfaces.

For pumps handling gritty or sandy water, the ring construction should provide an apron on which the stream leaving the leakage joint can impinge, as sand or grit in the jet will erode any surface it hits. Thus, a form of L-type casing ring similar to that shown in Figure 45 should be used.

Wearing-Ring Location In some designs, leakage is controlled by an axial clearance (see Figure 47). Generally, this design requires a means of adjusting the shaft position for

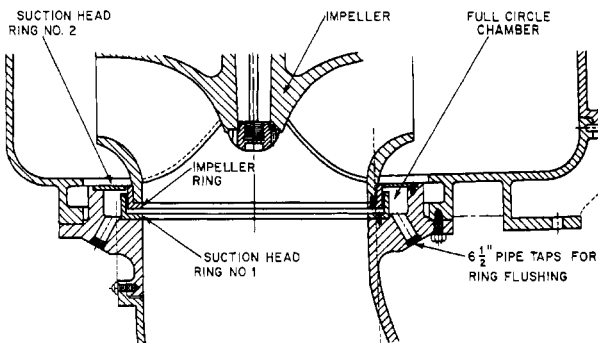


FIGURE 46 Two sets of rings with space between for flushing water (Flowserve Corporation)

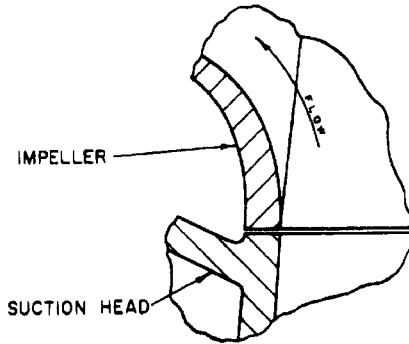


FIGURE 47 A leakage joint with axial clearance

proper clearance. Then, if uniform wear occurs over the two surfaces, the original clearance can be restored by adjusting the position of the impeller. A limit exists to the amount of wear that can be compensated, because the impeller must be nearly central in the casing waterways.

Leakage joints with axial clearance are not popular for double-suction pumps because a very close tolerance is required in machining the fit of the rings in reference to the centerline of the volute waterways. Joints with radial clearances, however, enable some shifting of the impeller for centering. The only adverse effect is a slight inequality in the lengths of the leakage paths on the two sides.

So far, this discussion has treated only those leakage joints located adjacent to the impeller eye or at the smallest outside shroud diameter. Designs have been made where the leakage joint is at the periphery of the impeller. In a vertical pump, this design is advantageous because the space between the joint and the suction waterways is open and so sand or grit cannot collect. Because of rubbing speed and because the impeller diameters used in the same casing vary over a wide range, the design is impractical in regular pump lines.

Mounting Stationary Wearing Rings In small single-suction pumps with suction heads, a stationary wearing ring is usually pressed into a bore in the head and may or may not be further locked by several set screws located half in the head and half in the ring (refer to Figure 37). Larger pumps often use an L-type ring with the flange held against a face on the head. In axially split casing pumps, the cylindrical casing bore (in which the casing ring will be mounted) should be slightly larger than the outside diameter of the ring. Unless some clearance is provided, distortion of the ring may occur when the two casing halves are assembled. However, the joint between the casing ring and the casing must be tight enough to prevent leakage. This is usually provided by a radial metal-to-metal joint (refer to Figure 39) arranged so that the discharge pressure presses the ring against the casing surface.

As it is not desirable for the casing ring of an axially split casing pump to be pinched by the casing, the ring will not be held tightly enough to prevent its rotation unless special provisions are made to keep it in place. One means of accomplishing this is to place a pin in the casing that will project into a hole bored in the ring or, conversely, to provide a pin in the ring that will fit into a hole bored in the casing or into a recess at the casing split joint.

Another method is to have a tongue on the casing ring that extends around 180° and engages a corresponding groove in one-half of the casing. This method can be used with casting rings having a central flange by making the diameter of the flange larger for 180° and cutting a deeper groove in that half of the casing.

Many methods are used for holding impeller rings on the impeller. Probably the simplest is to rely on a press fit of the ring on the impeller or, if the ring is of proper material,

on a shrink fit. Designers do not usually feel that a press fit is sufficient and often add several machine screws or set screws located half in the ring and half in the impeller, as in Figure 37.

An alternative to axial machine screws being located half in the ring and half in the impeller is the radial pin, which goes through the center of the ring into the impeller (or from the inside of the impeller eye outward into the ring). This pinning method avoids having to drill and tap holes half in a hardened wearing ring and half in a softer impeller hub. For higher speeds, installing the radial pins from the inside of the impeller eye outward into the wearing ring captures the pins and protects against pin loss, especially at higher operating speeds. Generally, some additional locking method is used rather than relying solely on friction between the ring and the impeller.

In the design of impeller rings, consideration has to be given to the stretch of the ring caused by centrifugal force, especially if the pump is of a high-speed design for the capacity involved. For example, some pumps operate at speeds that would cause the rings to become loose if only a press fit is used. For such pumps, shrink fits should be used or, preferably, impeller rings should be eliminated.

Wearing-Ring Clearances Typical clearance and tolerance standards for nongalling wearing-joint metals in general service pumps are shown in Figure 48. They apply to the following combinations: (1) bronze with a dissimilar bronze, (2) cast-iron with bronze, (3) steel with bronze, (4) Monel metal with bronze, and (5) cast-iron with cast iron. If the metals gall easily (like the chrome steels), the values given should be increased by 0.002 to 0.004 in.

The tolerance indicated is positive for the casing ring and negative for the impeller hub or impeller ring.

In a single-stage pump with a joint of nongalling components, the correct machining dimension for a casing-ring diameter of 9.000 in would be 9.000 plus 0.003 and minus 0.000 in. For the impeller hub or ring, the values would be 9.000 minus 0.018 (or 8.982) plus 0.000 and minus 0.003 in. Diametral clearances would be between 0.018 and 0.024 in. Obviously, clearances and tolerances are not translated into SI units by merely using conversion multipliers; they are rounded off as they are in the USCS system. The SI values for the previous USCS examples given are outlined here.

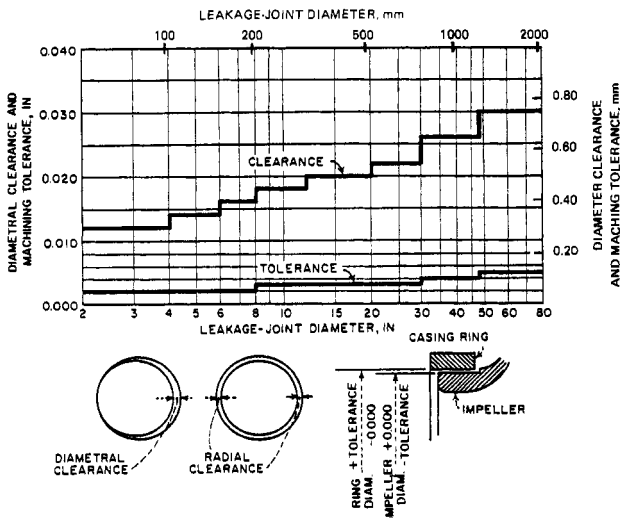


FIGURE 48 Wearing-ring clearances for single-stage pumps using nongalling materials

For a single-stage pump with a casing ring diameter of 230 mm, the machining dimensions would be as follows:

Casing ring	230 plus 0.08 and minus 0.00 mm
Impeller hub	230 minus 0.50 (229.5) plus 0.00 and minus 0.08 mm
Diametral clearances	Between 0.50 and 0.66 mm

Naturally, the manufacturer's recommendation for ring clearances and tolerances should be followed.

AXIAL THRUST

Axial Thrust in Single-Stage Pumps with Closed Impellers The pressures generated by a centrifugal pump exert forces on both stationary and rotating parts. The design of these parts balances some of these forces, but separate means may be required to counterbalance others.

An axial hydraulic thrust on an impeller is the sum of the unbalanced forces acting in the axial direction. Because reliable, large-capacity thrust bearings are readily available, an axial thrust in single-stage pumps remains a problem only in larger, higher speed units. Theoretically, a double-suction impeller is in hydraulic axial balance, with the pressures on one side equal to and counterbalancing the pressures on the other (see Figure 49). In practice, this balance may not be achieved for the following reasons:

1. The suction passages to the two suction eyes may not provide equal or uniform flows to the two sides.
2. External conditions, such as an elbow located too close to the pump suction nozzle, may cause unequal flow to the two suction eyes.
3. The two sides of the discharge casing waterways may not be symmetrical, or the impeller may be located off-center. These conditions will alter the flow characteristics between the impeller shrouds and the casing, causing unequal pressures on the shrouds.
4. Unequal leakage through the two leakage joints can upset the balance.

Combined, these factors can create an axial unbalance. To compensate for this, all centrifugal pumps, even those with double-suction impellers, incorporate thrust bearings.

The ordinary single-suction, closed, radial-flow impeller with the shaft passing through the impeller eye (refer to Figure 49) is subject to an axial thrust because a portion of the front wall is exposed to suction pressure and thus relatively more backwall

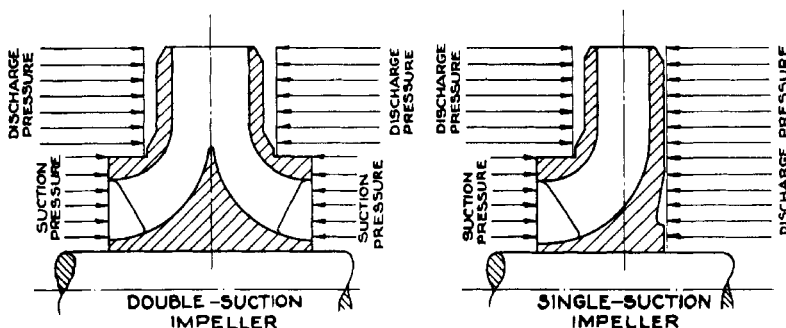


FIGURE 49 The origin of pressures acting on impeller shrouds to produce an axial thrust

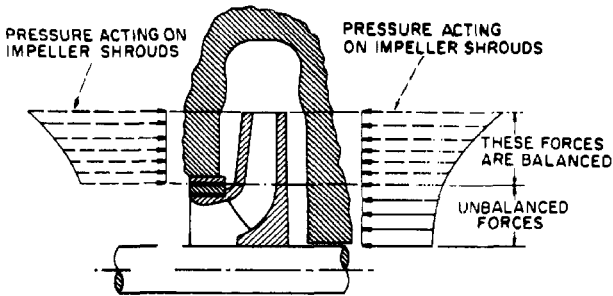


FIGURE 50 Pressure distribution on the front and back shrouds of the single-suction impeller with a shaft through the impeller eye

surface is exposed to discharge pressure. If the discharge chamber pressure is uniform over the entire impeller surface, the axial force acting toward the suction would be equal to the product of the net pressure generated by the impeller and the unbalanced annular area.

Actually, pressure on the two single-suction, closed impeller walls is not uniform. The liquid trapped between the impeller shrouds and casing walls is in rotation, and the pressure at the impeller periphery is appreciably higher than the impeller hub. Although we need not be concerned with the theoretical calculations for this pressure variation, Figure 50 describes it qualitatively. Generally speaking, the axial thrust toward the impeller suction may be about 20 to 30% less than the product of the net pressure and the unbalanced area.

To eliminate the axial thrust of a single-suction impeller, a pump can be provided with both front and back wearing rings. To equalize the thrust areas, the inner diameter of both rings is made the same (see Figure 51). Pressure approximately equal to the suction pressure is maintained in a chamber located on the impeller side of the back wearing ring by drilling balancing holes through the impeller. Leakage past the back wearing ring is returned to the suction area through these holes. However, with large single-stage, single-suction pumps, balancing holes are considered undesirable because leakage back to the impeller suction opposes the main flow, creating disturbances. In such pumps, a piped connection to the pump suction replaces the balancing holes.

Another way to eliminate or reduce an axial thrust in single-suction impellers is to use pump-out vanes on the back shroud. The effect of these vanes is to reduce the pressure

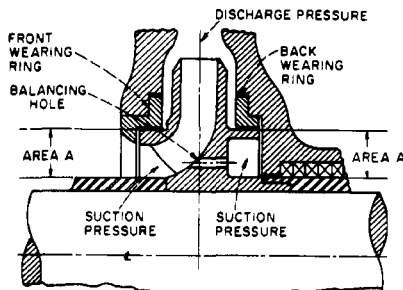


FIGURE 51 Balancing the axial thrust of a single-suction impeller by means of wearing rings on the back side and balancing holes

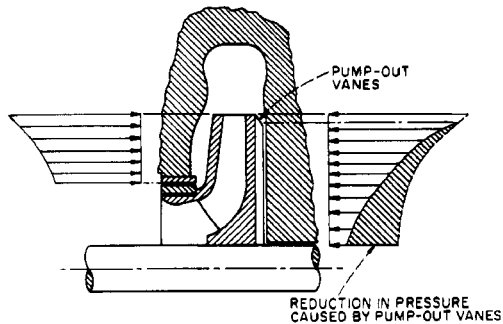


FIGURE 52 Pump-out vanes used in a single-suction impeller to reduce axial thrust

acting on the back shroud of the impeller (see Figure 52). This design, however, is generally used only in pumps handling dirty liquids where it keeps the clearance space between the impeller back shroud and the casing free of foreign matter.

So far, our discussion of axial thrust has been limited to single-suction, closed impellers with a shaft passing through the impeller eye that are located in pumps with two seal chambers, one on either side of the impeller. In these pumps, suction-pressure magnitude does not affect the resulting axial thrust.

On the other hand, axial forces acting on an overhung impeller with a single seal chamber (see Figure 53) are definitely affected by suction pressure. In addition to the unbalanced force found in a single-suction, two-seal chamber design (refer to Figure 50), there is an axial force equivalent to the product of the shaft area through the seal chamber and the difference between suction and atmospheric pressure. This force acts toward the impeller suction when the suction pressure is less than atmospheric or in the opposite direction when it is higher than atmospheric.

When an overhung impeller pump handles a suction lift, the additional axial force is very low. For example, if the shaft diameter through the stuffing box is 2 in (50.8 mm) (area = 3.14 in² or 20.26 cm²), and if the suction lift is 20 ft (6.1 m) of water equivalent to an absolute pressure of 6.06 lb/in² (0.42 bar abs), the axial force caused by the overhung impeller and acting toward the suction will be only 27 lb (121 N). On the other hand, if the suction pressure is 100 lb/in² (6.89 bar), the force will be 314 lb (1405 N) and will act in the opposite direction. Therefore, because the same pump may be used under many conditions

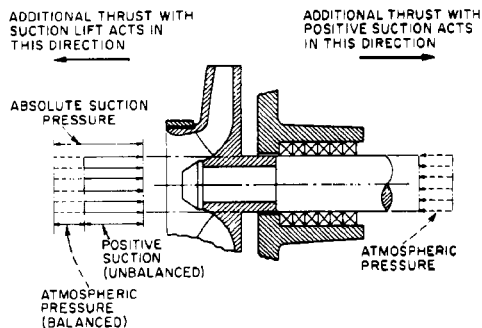


FIGURE 53 An axial thrust problem with a single-suction, overhung impeller and a single stuffing box or seal chamber

over a wide range of suction pressures, the thrust bearing of pumps with single-suction, overhung impellers must be arranged to take the thrust in either direction. They must also be selected with a sufficient thrust capacity to counteract forces set up under the maximum suction pressure established for that particular pump.

This extra thrust capacity can become quite significant in certain special cases, such as with boiler circulating pumps. These are usually of the single-suction, single-stage, overhung impeller type and may be exposed to suction pressures as high as 2800 lb/in² gage (193 bar). This is approximately the vapor pressure of 685°F (363°C) boiler water. If such a pump has a shaft diameter of 6 in (15.24 cm), the unbalanced thrust would be as much as 77,500 lb (346,770 N), and the thrust bearing has to be capable of counteracting this.

Axial Thrust of Single-Suction Semiopen Radial-Flow Impellers The axial thrust generated in semiopen impellers is higher than that in closed impellers. This is illustrated in Figure 58, which shows that the pressure on the open side of the impeller varies from essentially the discharge pressure at the periphery (at diameter D_2) to the suction pressure at the impeller eye (at diameter D_1). The pressure distribution on the back shroud is essentially the same as that illustrated in Figure 50, varying from discharge pressure at the periphery to some portion of this pressure at the impeller hub. This latter pressure is, of course, substantially higher than the suction pressure. The unbalanced portion of the axial thrust on the impeller is represented by the cross-hatched area in Figure 54.

One of the means available for partially balancing this increased axial thrust is to provide the back shroud with pump-out vanes, as in Figure 52.

Fully open impellers or semiopen impellers with a portion of the back shroud removed produce an axial thrust somewhat higher than closed impellers and somewhat lower than semiopen impellers.

Axial Thrust of Mixed-Flow and Axial-Flow Impellers The axial thrust in radial impellers is produced by the static pressures on the impeller shrouds. Axial-flow impellers have no shrouds, and the axial thrust is created strictly by the difference in pressure on the two faces of the impeller vanes. In addition, a difference may exist in pressure acting on the two shaft hub ends, one generally subject to discharge pressure and the other to suction pressure.

With mixed-flow impellers, axial thrust is a combination of forces caused by the action of the vanes on the liquid and those arising from the difference in the pressures acting on the various surfaces. Wearing rings are often provided on the back of mixed-flow impellers, with either balancing holes through the impeller hub or an external balancing pipe leading back to the suction.

Except for very large units and in certain special applications, the axial thrust developed by mixed-flow and axial-flow impellers is carried by thrust bearings with the necessary load capacity.

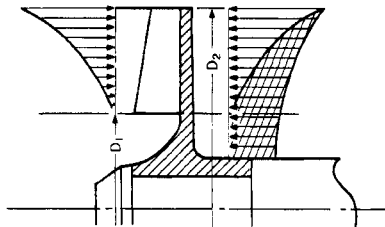


FIGURE 54 Axial thrust in a semiopen single-suction impeller

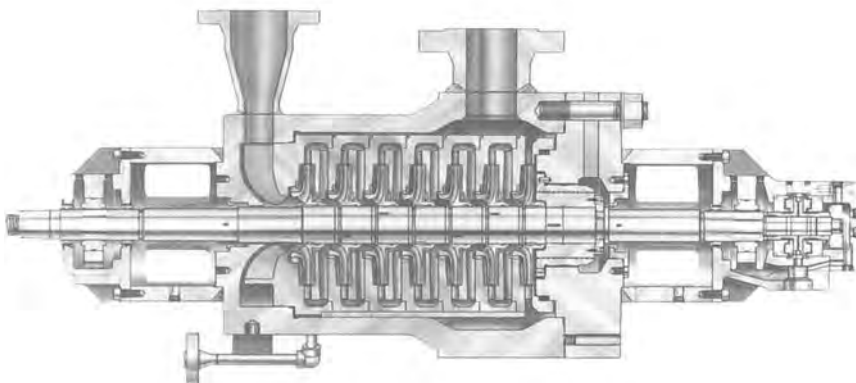


FIGURE 55 Multistage pump with single-suction impellers facing in one direction and hydraulic balancing device (Flowsolve Corporation)

Axial-Thrust in Multistage Pumps Most multistage pumps are built with single-suction impellers in order to simplify the design of the interstage connections. Two obvious arrangements are possible for the single-suction impellers:

1. Several single-suction impellers can be mounted on one shaft, each having its suction inlet facing in the same direction and its stages following one another in ascending order of pressure (see Figure 55). The axial thrust is then balanced by a hydraulic balancing device.
2. An even number of single-suction impellers can be used, one-half facing in one direction and the other half facing in the opposite direction. With this arrangement, an axial thrust on the first half is compensated by the thrust in the opposite direction on the other half (see Figure 56). This mounting of single-suction impellers back to back is frequently called *opposed impellers*.

An uneven number of single-suction impellers can be used with this arrangement, provided the correct shaft and interstage bushing diameters are used to give the effect of a hydraulic balancing device that will compensate for the hydraulic thrust on one of the stages.

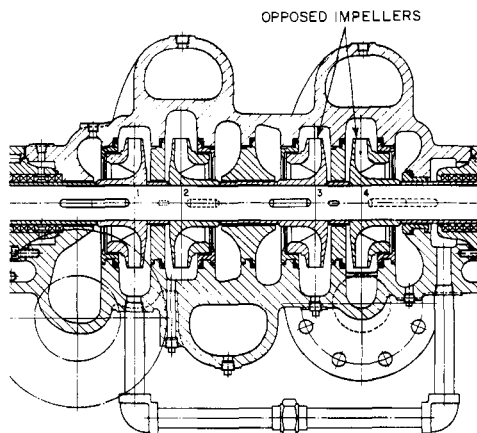


FIGURE 56 A four-stage pump with opposed impellers (Flowsolve Corporation)

It is important to note that the opposed impeller arrangement completely balances an axial thrust only under the following conditions:

- The pump must be provided with two seal chambers.
- The shaft must have a constant diameter.
- The impeller hubs must not extend through the interstage portion of the casing separating adjacent stages.

Except for some special pumps that have an internal and enclosed bearing at one end, and therefore only one seal chamber, most multistage pumps fulfill the first condition. Because of structural requirements, however, the last two conditions are not practical. A slight residual thrust is usually present in multistage opposed impeller pumps and is carried on the thrust bearing.

HYDRAULIC BALANCING DEVICES

If all the single-suction impellers of a multistage pump face the same direction, the total theoretical hydraulic axial thrust acting toward the suction end of the pump will be the sum of the individual impeller thrusts. The thrust magnitude will be approximately equal to the product of the net pump pressure and the annular unbalanced area. Actually, the axial thrust turns out to be about 70 to 80% of this theoretical value.

Some form of hydraulic balancing device must be used to balance this axial thrust and to reduce the pressure on the seal chamber adjacent to the last-stage impeller. This hydraulic balancing device may be a balancing drum, a balancing disk, or a combination of the two.

Balancing Drums The balancing drum is illustrated in Figure 57. The balancing chamber at the back of the last-stage impeller is separated from the pump interior by a drum that is usually keyed to the shaft and rotates with it. The drum is separated by a small radial clearance from the stationary portion of the balancing device, called the *balancing-drum head*, or balancing sleeve, which is fixed to the pump casing.

The balancing chamber is connected either to the pump suction or to the vessel from which the pump takes its suction. Thus, the back pressure in the balancing chamber is only slightly higher than the suction pressure, the difference between the two being equal to the friction losses between this chamber and the point of return. The leakage between

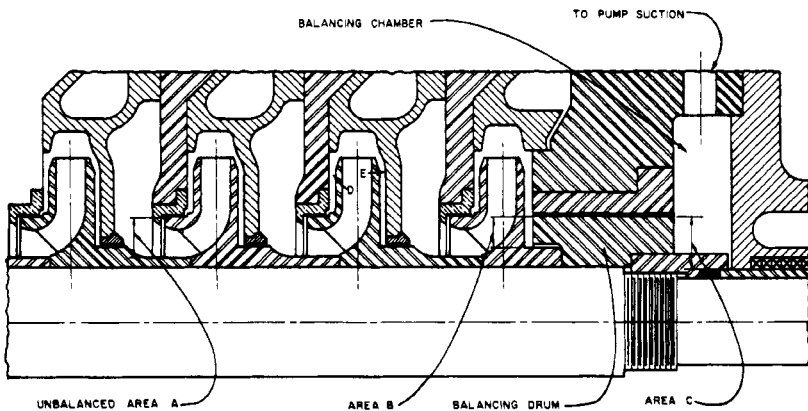


FIGURE 57 Balancing drum

the drum and the drum head is, of course, a function of the differential pressure across the drum and of the clearance area.

The forces acting on the balancing drum in Figure 57 are the following:

- *Toward the discharge end:* the discharge pressure multiplied by the front balancing area (area *B*) of the drum
- *Toward the suction end:* the back pressure in the balancing chamber multiplied by the back balancing area (area *C*) of the drum

The first force is greater than the second, thereby counterbalancing the axial thrust exerted upon the single-suction impellers. The drum diameter can be selected to balance the axial thrust completely or within 90 to 95%, depending on the desirability of carrying any thrust-bearing loads.

It has been assumed in the preceding simplified description that the pressure acting on the impeller walls is constant over their entire surface and that the axial thrust is equal to the product of the total net pressure generated and the unbalanced area. Actually, this pressure varies somewhat in the radial direction because of the centrifugal force exerted upon the liquid by the outer impeller shroud (refer to Figure 50). Furthermore, the pressures at two corresponding points on the opposite impeller faces (*D* and *E* in Figure 57) may not be equal because of a variation in clearance between the impeller wall and the casing section separating successive stages. Finally, a pressure distribution over the impeller wall surface may vary with head and capacity operating conditions.

This pressure distribution and design data can be determined quite accurately for any one fixed operating condition, and an effective balancing drum could be designed on the basis of the forces resulting from this pressure distribution. Unfortunately, varying head and capacity conditions change the pressure distribution, and as the area of the balancing drum is necessarily fixed, the equilibrium of the axial forces can be destroyed.

The objection to this is not primarily the amount of the thrust, but rather that the direction of the thrust cannot be predetermined because of the uncertainty about internal pressures. Still it is advisable to predetermine normal thrust direction, as this can influence external mechanical thrust-bearing design. Because 100% balance is unattainable in practice and because the slight but predictable unbalance can be carried on a thrust bearing, the balancing drum is often designed to balance only 90 to 95% of the total impeller thrust.

The balancing drum satisfactorily balances the axial thrust of single-suction impellers and reduces pressure on the discharge-side stuffing box. It lacks, however, the virtue of automatic compensation for any changes in axial thrust caused by varying impeller reaction characteristics. In effect, if the axial thrust and balancing drum forces become unequal, the rotating element will tend to move in the direction of the greater force. The thrust bearing must then prevent excessive movement of the rotating element. The balancing drum performs no restoring function until such time as the drum force again equals the axial thrust. This automatic compensation is the major feature that differentiates the balancing disk from the balancing drum.

Balancing Disks The operation of the simple balancing disk is illustrated in Figure 58. The disk is fixed to and rotates with the shaft. It is separated by a small axial clearance from the balancing disk head, or balancing sleeve, which is fixed to the casing. The leakage through this clearance flows into the balancing chamber and from there either to the pump suction or to the vessel from which the pump takes its suction. The back of the balancing disk is subject to the balancing chamber back pressure, whereas the disk face experiences a range of pressures. These vary from discharge pressure at its smallest diameter to back pressure at its periphery. The inner and outer disk diameters are chosen so that the difference between the total force acting on the disk face and that acting on its back will balance the impeller axial thrust.

If the axial thrust of the impellers should exceed the thrust acting on the disk during operation, the latter is moved toward the disk head, reducing the axial clearance between

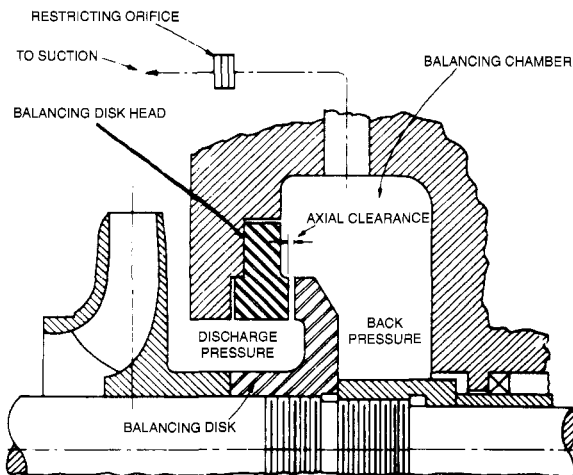


FIGURE 58 A simple balancing disk

the disk and the disk head. The amount of leakage through the clearance is reduced so that the friction losses in the leakage return line are also reduced, lowering the back pressure in the balancing chamber. This lowering of pressure automatically increases the pressure difference acting on the disk and moves it away from the disk head, increasing the clearance. Now the pressure builds up in the balancing chamber, and the disk is again moved toward the disk head until an equilibrium is reached.

To assure proper balancing in disk operation, the change in back pressure in the balancing chamber must be of an appreciable magnitude. Thus, with the balancing disk wide open with respect to the disk head, the back pressure must be substantially higher than the suction pressure to give a resultant force that restores the normal disk position. This can be accomplished by introducing a restricting orifice in the leakage return line that increases back pressure when leakage past the disk increases beyond normal. The disadvantage of this arrangement is that the pressure on the seal chamber is variable, a condition that may be injurious to the life of the seal and therefore should be avoided.

Combination Balancing Disk and Drum For the reasons just described, the simple balancing disk is seldom used. The combination balancing disk and drum (see Figure 59) was developed to obviate the shortcomings of the disk while retaining the advantage of automatic compensation for axial thrust changes.

The rotating portion of this balancing device consists of a long cylindrical body that turns within a drum portion of the disk head. This rotating part incorporates a disk similar to the one previously described. In this design, radial clearance remains constant regardless of disk position, whereas the axial clearance varies with the pump rotor position. The following forces act on this device:

- *Toward the discharge end:* the sum of the discharge pressure multiplied by area A , plus the average intermediate pressure multiplied by area B
- *Toward the suction end:* the back pressure multiplied by area C

Whereas the position-restoring feature of the simple balancing disk required an undesirably wide variation of the back pressure, it is now possible to depend upon a variation of the intermediate pressure to achieve the same effect. Here is how it works: When the pump rotor moves toward the suction end (to the left in Figure 59) because of increased axial thrust, the axial clearance is reduced and pressure builds up in the intermediate

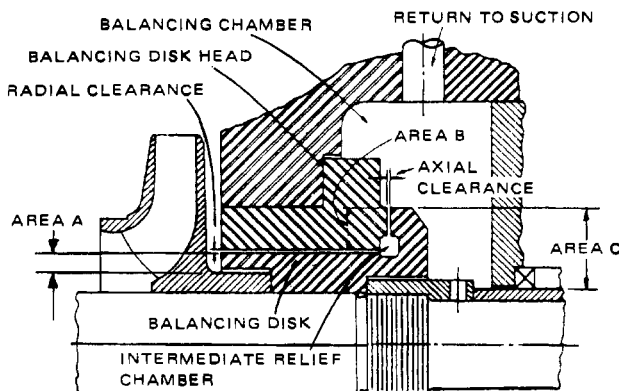


FIGURE 59 A combination balancing disk and drum

relief chamber, increasing the average value of the intermediate pressure acting on area *B*. In other words, with reduced leakage, the pressure drop across the radial clearance decreases, increasing the pressure drop across the axial clearance. The increase in intermediate pressure forces the balancing disk toward the discharge end until equilibrium is reached. Movement of the pump rotor toward the discharge end would have the opposite effect, increasing the axial clearance and the leakage and decreasing the intermediate pressure acting on area *B*.

Now numerous hydraulic balancing device modifications are in use. One typical design separates the drum portion of a combination device into two halves, one preceding and the second following the disk (see Figure 60). The virtue of this arrangement is a definite cushioning effect at the intermediate relief chamber, thus avoiding too positive a restoring action, which might result in the contacting and scoring of the disk faces.

SHAFTS AND SHAFT SLEEVES

The basic function of a centrifugal pump shaft is to transmit the torques encountered when starting and during operation while supporting the impeller and other rotating parts. It must do this job with a deflection less than the minimum clearance between rotating and stationary parts. The loads involved are (1) the torques, (2) the weight of the parts, and (3) both radial and axial hydraulic forces. In designing a shaft, the maximum allowable deflection, the span or overhang, and the location of the loads all have to be considered, as does the critical speed of the resulting design.

Shafts are usually proportioned to withstand the stress set up when a pump is started quickly, such as when the driving motor is energized directly across the line. If the pump handles hot liquids, the shaft is designed to withstand the stress set up when the unit is started cold without any preliminary warm-up.

Critical Speeds Any object made of an elastic material has a natural period of vibration. When a pump rotor or shaft rotates at any speed corresponding to its natural frequency, minor unbalances will be magnified. These speeds are called the *critical speeds*.

In conventional pump designs, the rotating assembly is theoretically uniform around the shaft axis and the center of mass should coincide with the axis of rotation. This theory does not hold for two reasons. First, minor machining or casting irregularities always occur. Second, variations exist in the metal density of each part. Thus, even in vertical-shaft machines having no radial deflection caused by the weight of the parts, this eccentricity of the center of mass produces a centrifugal force and therefore a deflection when

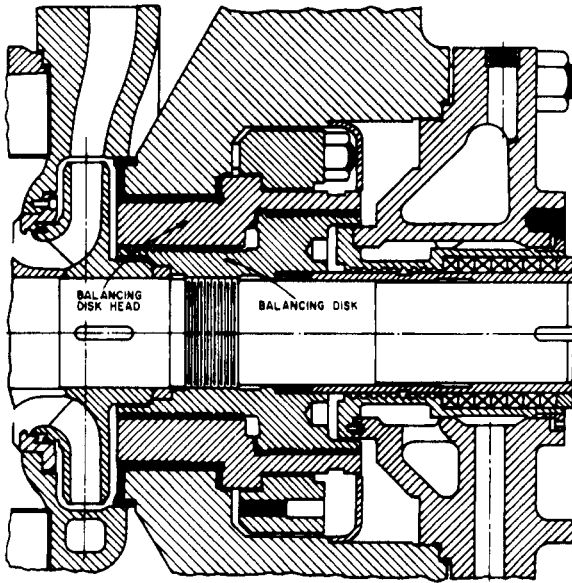


FIGURE 60 A combination balancing disk and drum with a disk located in the center portion of the drum
(Flowserve Corporation)

the assembly rotates. At the speed where the centrifugal force exceeds the elastic restoring force, the rotor will vibrate as though it were seriously unbalanced. If it is run at that speed without restraining forces, the deflection will increase until the shaft fails.

Rigid and Flexible Shaft Designs The lowest critical speed is called the first critical speed, the next highest is called the second, and so forth. In centrifugal pump nomenclature, a *rigid shaft* means one with an operating speed lower than its first critical speed. A *flexible shaft* is one with an operating speed higher than its first critical speed. Once an operating speed has been selected, relative shaft dimensions must still be determined. In other words, it must be decided whether the pump will operate above or below the first critical speed.

Actually, the shaft critical speed can be reached and passed without danger because frictional forces tend to restrain the deflection. These forces are exerted by the surrounding liquid, and the various internal leakage joints acting as internal liquid-lubricated bearings. Once the critical speed is passed, the pump will run smoothly again up to the second speed corresponding to the natural rotor frequency, and so on to the third, fourth, and all higher critical speeds.

Designs rated for 1800 rpm (or lower) are usually of the rigid-shaft type. On the other band, high-head 3600 rpm (or higher) multistage pumps, are frequently of the flexible-shaft type. It is possible to operate centrifugal pumps above their critical speeds for the following two reasons: (1) very little time is required to attain full speed from rest (the time required to pass through the critical speed must therefore be extremely short) and (2) the pumped liquid in the internal leakage joints acts as a restraining force on the vibration.

Shaft Sizing Shaft diameters are usually larger than what is actually needed to transmit the torque. Two factors that assure this conservative design are a) shaft deflection limit requirements at the impellers and mechanical seal faces, and b) requirements for ease of rotor assembly.

For between bearings pumps (single-stage double-suction, multistage), the maximum shaft diameter is at the impeller mounting surface. From there, the shaft is stepped down in diameter toward the bearing mounting surfaces and the coupling mounting surface. The shaft size at the coupling must consider torsional strength; the remainder of the shaft size is based on other considerations, including the steps in diameter required for shaft nuts or other axial location devices for the impeller and sealing members (mechanical seal components or shaft sleeves for packing). When these considerations have been decided, the rotor must be analyzed for conformance to the dynamic requirements or deflection and critical speed. These topics are covered in Section 2.3.

Shaft Sleeves Pump shafts are usually protected from erosion, corrosion, and wear at seal chambers, leakage joints, internal bearings, and in the waterways by renewable sleeves. Years ago, the most common shaft sleeve function was to protect the pump shaft from wear at stuffing box packing. However, most pumps today use mechanical seals, not packing. Furthermore, most mechanical seals today are cartridge mechanical seals, and the sleeve is part of the seal package. Packing and mechanical seal topics are covered in Section 5.

Shaft sleeves serving other functions are given specific names to indicate their purpose. For example, a shaft sleeve used between two multistage pump impellers in conjunction with the interstage bushing to form an interstage leakage joint is called an *interstage* or *distance sleeve*.

For between bearings pumps, a favored shaft sleeve construction for pumps equipped with packing uses an external shaft nut to hold the sleeve in an axial position against the impeller hub. Sleeve rotation is prevented by a key, usually an extension of the impeller key (see Figure 61). The axial thrust of the impeller(s) is transmitted through the sleeve to the external shaft nut.

For pumps using cartridge mechanical seals, the preferred design is shown in Figure 63. An internal impeller nut locates the impeller axially on both sides, leaving a large open seal chamber for installing the seal cartridges and locking them to both the seal chamber and the shaft. When shaft nuts are used to retain the sleeves and impellers axially, they are usually manufactured with right- and left-hand threads. The friction of the pumpage and inadvertent contact with stationary parts or bushings will tend to tighten the nuts against the sleeve and impeller hub (rather than loosen them). Usually, the shaft sleeves utilize extended impeller keys to prevent rotation.

In pumps with overhung impellers, various types of sleeves are used. Most pumps use mechanical seals, and the shaft sleeve is usually a part of the mechanical seal package supplied by the seal manufacturer. Many mechanical seals are of the cartridge design, which is set and may be bench-tested for leakage prior to installation in the pump. (For a further discussion of mechanical seals, see Section 5.2.)

For overhung impeller pumps that utilize packing for sealing, the packing sleeves generally extend from the impeller hub through the seal chambers (or stuffing boxes) to protect

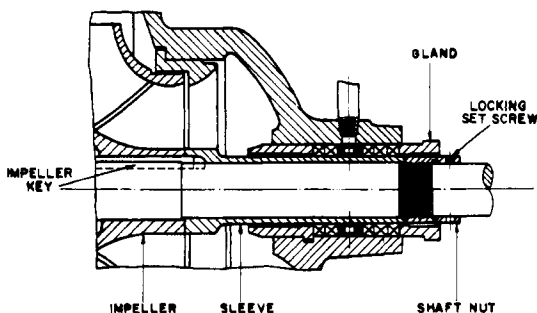


FIGURE 61 A sleeve with external locknut and impeller key extending into the sleeve to prevent rotation

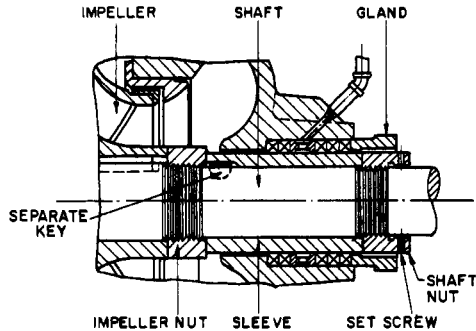


FIGURE 62 A sleeve with an internal impeller nut, external shaft-sleeve nut, and a separate key for the sleeve

the pump shaft from wear (see Figure 63). The sleeves are usually keyed to the shaft to prevent rotation. If a hook-type sleeve is used, the hook part of the sleeve is clamped between the impeller and a shaft shoulder to maintain the axial position of the sleeve. A hook-type sleeve used to be popular for overhung impeller pumps that operate at high temperatures because it is clamped at the impeller end and the rest of the sleeve is free to expand axially with temperature changes. But with the increased use of cartridge-type seals, the use of hook-type sleeves is minimal today.

In designs with a metal-to-metal joint between the sleeve and the impeller hub or shaft nut, a sealing device is required between the sleeve and the shaft to prevent leakage. Pumped liquid can leak into the clearance between the shaft and the sleeve when operating under a positive suction head and air can leak into the pump when operating under a negative suction head. This seal can be accomplished by means of an *O*-ring, as shown in Figure 64, or a flat gasket. For high temperature services, the sealing device must be either acceptable for the temperature to which it will be exposed, or it must be located outside the high temperature liquid environment. An alternative design used for some high-temperature process pumps is shown in Figure 65. In this arrangement, the contact surface of the hook-type sleeve and the shaft is ground at a 45° angle to form a metal-to-metal seal. That end of the sleeve is locked, but the other is free to expand with temperature changes.

When *O*-rings are used, any sealing surfaces must be properly finished to ensure a positive seal is achieved. All bores and changes in diameter over which *O*-rings must be passed should be properly radiused and chamfered to protect against damage during assembly. Guidelines for assembly dimensions and surface finish criteria are listed in *O*-ring manufacturers' catalogs.

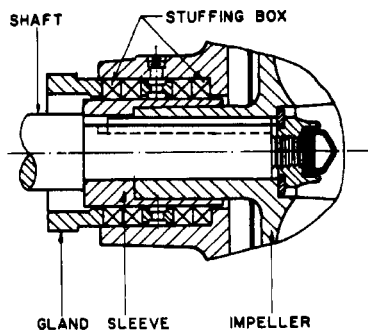


FIGURE 63 A sleeve for pumps with an overhung impeller

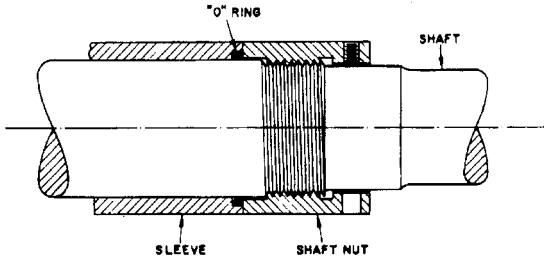


FIGURE 64 A seal arrangement for the shaft sleeve to prevent leakage along the shaft

Material for Packing Sleeves Packing sleeves are surrounded in the stuffing box by packing. The sleeve must be smooth so that it can turn without generating too much friction and heat. Thus, the sleeve materials must be capable of taking a very fine finish, preferably a polish. Cast-iron is therefore not suitable. A hard bronze is generally used for pumps handling clear water, but chrome or other stainless steels are sometimes preferred. For pumps subject to abrasives, hardened chrome or other stainless steels give good results. In most applications, a hardened chromium steel sleeve will be technically adequate, and the most economical choice. For severe or unusual conditions, coated sleeves are used. Ceramic coatings, applied using a plasma spray process, have also been used. Chromium oxide and aluminum oxide are the most common ceramic coatings. Both are extremely hard and resist abrasive wear well.

Ceramic coatings have been replaced in some applications by tungsten carbide coatings applied using a *high-velocity oxyfuel* (HVOF) process. The superior impact resistance and bond strength of these coatings is well documented. Another coating that is widely used on pump sleeves is a nickel-chromium-silicon-boron self-fluxing coating. This coating has a good resistance to galling and moderate resistance to abrasive wear.

Sleeves intended for coating should be machined with an undercut, so that the coating does not extend to the edge of the sleeve. This will prevent chipping at the edge, especially with the more brittle ceramic coatings.

SEAL CHAMBERS AND STUFFING BOXES

Seal chambers have the primary function of protecting the pump against leakage at the point where the shaft passes out through the pump pressure casing. If the pump handles a suction lift and the pressure at the bottom of the seal chamber (the point closest to the

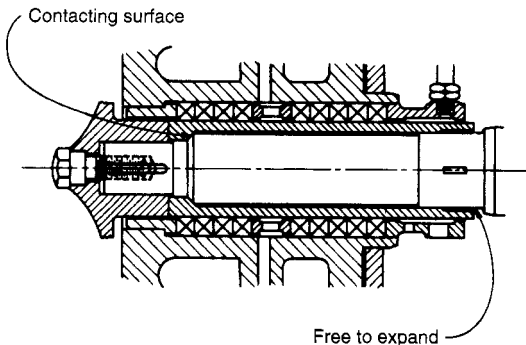


FIGURE 65 A sleeve with a 45° bevel contacting surface

inside of the pump) is below atmospheric, the seal chamber function is to prevent air leakage into the pump. If this pressure is above atmospheric, the function is to prevent liquid leakage out of the pump.

When sealing is accomplished by means of a mechanical seal, the seals are installed in a *seal chamber*. When sealing is accomplished by means of packing, the seal chamber is commonly referred to as a *stuffing box*. Packing and mechanical seal topics are covered in Section 5.

Sealing Liquid Arrangements When a pump handles clean, cool water, sealing liquid connections are usually to the pump discharge or, in multistage pumps, to an intermediate stage. An independent supply of sealing water should be provided if any of the following conditions exist:

- A suction lift in excess of 15 ft (4.5 m)
- A discharge pressure under 10 lb/in² (0.7 bar)
- Hot water (over 250°F or 120°C) being handled without adequate cooling (except for boiler-feed pumps, in which lantern rings are not used)
- Muddy, sandy, or gritty water being handled
- The pump is a hot-well pump.
- The liquid being handled is other than water, such as acid, juice, molasses, or sticky liquids, without special provision in the stuffing box design for the nature of the liquid

If the suction lift exceeds 15 ft (4.5 m), excessive air infiltration through the stuffing boxes may make priming difficult unless an independent seal is provided. A discharge pressure under 10 lb/in² (0.7 bar) may not provide sufficient sealing pressure. Hot-well (or condensate) pumps operate with as much as a 28-in Hg (710 mm Hg) vacuum, and air infiltration would take place when the pumps are standing idle in standby service.

When sealing water is taken from the pump discharge, an external connection may be made through small-diameter piping (see Figure 66) or internal passages. In some pumps, these connections are arranged so that a sealing liquid can be introduced into the packing or mechanical seal space through an internal drilled passage either from the pump casing or from an external source. When the liquid pumped is used for sealing, the external connection is plugged. If an external sealing liquid source is required, it is connected to the external pipe tap with a socket-head pipe plug inserted at the internal pipe tap.

For packing, it is sometimes desirable to locate the lantern ring with more packing on one side than on the other. For example, in gritty-water services, a lantern ring location closer to the inner portion of the pump would divert a greater proportion of sealing liquid into the pump, thereby keeping grit from working into the box. An arrangement with most of the packing rings between the lantern ring and the inner end of the stuffing box would be applied to reduce dilution of the pumped liquid.

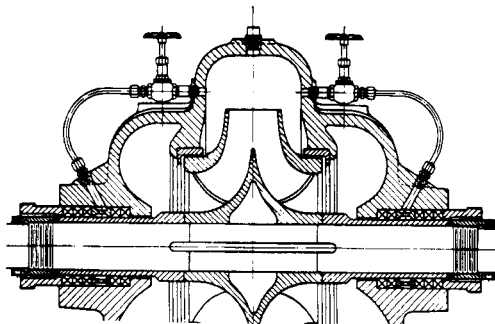


FIGURE 66 Piping connections from the pump discharge to seal cages

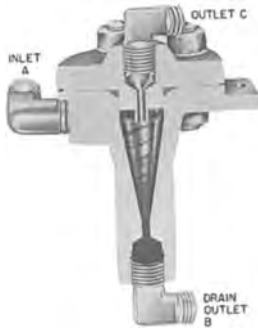


FIGURE 67 An illustration of the principle of cyclone separators (Flowserve Corp.)

Some pumps handle water in which there are small, even microscopic, solids. Using water of this kind as a sealing liquid introduces the solids into the leakage path, shortening the life of the packing and sleeves. It is sometimes possible to remove these solids by installing small pressure filters in the sealing water piping from the casing to the stuffing box.

Filters ultimately get clogged, though, unless they are frequently backwashed or otherwise cleaned out. This disadvantage can be overcome by using a cyclone (or centrifugal) separator. The operating principle of the cyclone separator is based on the fact that if a liquid under pressure is introduced tangentially into a vortexing chamber, a centrifugal force will make it rotate in the chamber, creating a vortex. Particles heavier than the liquid in which they are carried will hug the outside wall of the vortexing chamber and the liquid in the center of the chamber will be relatively free of foreign matter. The action of such a separator is illustrated in Figure 67. Liquid piped from the pump discharge or from an intermediate stage of a multistage pump is piped to inlet tap A, which is drilled tangentially to the cyclone bore. The liquid containing solids is directed downward to the apex of the cone at outlet tap B and is piped to the suction or to a low-pressure point in the system. The cleaned liquid is taken off at the center of the cyclone at outlet tap C and is piped to the stuffing box.

Sand that will pass through a No. 40 sieve will be completely eliminated in a cyclone separator with supply pressures as low as 20 lb/in² (1.4 bar). With 100 lb/in² (7-bar) supply pressure, 95% of the particles of 5-micron size will be eliminated.

Most city ordinances require that some form of backflow preventer be interposed between city water supply lines and connections to equipment where backflow or siphoning could contaminate a drinking water supply.

This is the case, for instance, with an independent sealing supply used for stuffing boxes of sewage pumps. Quite a variety of backflow preventers are available. In most cases, the device consists of two spring-loaded check valves in a series and a spring-loaded, diaphragm-actuated, differential-pressure relief valve located in the zone between the check valves (see Figure 68).

Stuffing Box Packing See Section 5.1.

MECHANICAL SEALS VERSUS PACKING

Sealing with a packed stuffing box is impractical for many conditions of service. In an ordinary packed stuffing box, the sealing between the rotating shaft or shaft sleeve and the stationary portion of the stuffing box (or seal chamber) is accomplished by means of rings of packing forced between the two surfaces and held tightly in place by a packing gland. The leakage around the shaft is controlled by merely tightening or loosening the packing

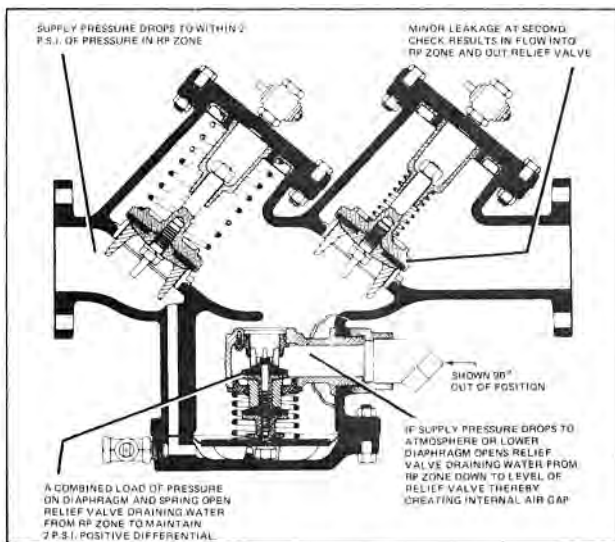


FIGURE 68 Backflow preventer (Hersey Products)

gland nuts (or bolts). The actual sealing surfaces consist of the axial rotating surface of the shaft or shaft sleeve and the stationary packing. Attempts to reduce or eliminate all leakage from a conventional packed stuffing box increase the compressive load of the packing gland on the packing. The packing, being semiplastic, forms more closely to the shaft or shaft sleeve and tends to reduce the leakage. After a certain point, however, the leakage between the packing and the rotating shaft or shaft sleeve becomes inadequate to carry away the heat generated by the packing rubbing on the rotating surface, and the packing fails to function. This failure can result in burned packing, packing “blow out,” and severely damaged shaft or shaft sleeve surfaces. If the sealing surface is coated, this coating may be destroyed. Even before this condition is reached, the shaft or shaft sleeve may be severely worn and scored by the packing, so that it becomes impossible to pack the stuffing box satisfactorily.

These undesirable characteristics prohibit the use of packing as a sealing method if some leakage of the pumpage to the atmosphere is not acceptable. Packing is limited in its application pressure and temperature range (see Section 5.1), and it is usually not acceptable for any flammable or hazardous pumping services. To address these limitations, the mechanical seal was developed (see Section 5.2). The mechanical seal has found general acceptance in nearly all pumping applications. Packing is still used in certain low-pressure, low-temperature applications where leakage of the pumpage is not a problem and a history of satisfactory, economical service exists.

Mechanical seals are not always the solution to every sealing situation. Seals are still subject to failure, and their failure may be more rapid and abrupt than that of packing. If packing fails, the pump can many times be kept running by temporary adjustments until it is convenient to shut it down. If a mechanical seal fails, most often the pump must be shut down immediately. As both packed stuffing boxes and conventional mechanical face seals are subject to wear, both are subject to failure. Whether one or the other should be used depends on the specific application and the experience of the user. In some cases, both give good service and the choice becomes a matter of personal preference or cost. Table 2 adds other comments and summarizes some advantages and disadvantages of packing and mechanical seals.

Centrifugal Pump Mechanical Seals See Section 5.2.

TABLE 2 Comparison of packing and mechanical seals

Advantages	Disadvantages
Packing	
1. Lower initial cost	1. Relatively high leakage
2. Easily installed as rings and glands are split	2. Requires regular maintenance
3. Good reliability to medium pressures and shaft speeds	3. Wear of shaft of shaft sleeve can be relatively high
4. Can handle large axial movements (thermal expansion of stuffing box versus shaft)	4. Power losses may be high
5. Can be used in rotating or reciprocating applications	
6. Leakage tends to increase gradually, giving warning of impending breakdown	
Mechanical seals	
1. Very low leakage/no leakage	1. Higher initial cost
2. Require no maintenance	2. Easily installed but may require some disassembly of pump (couplings and so on)
3. Eliminate sleeve wear/shaft wear	
4. Very good reliability	
5. Can handle higher pressures and speeds	
6. Easily applied to carcinogenic, toxic, flammable, or radioactive liquids	
7. Low power loss	

Source: John Crane Inc.

Injection-Type Shaft Seals See Section 5.3.

BEARINGS

See Section 6.

Oil Film or Sleeve Bearings See Section 6.1.

COUPLINGS

Centrifugal pumps are connected to their drivers through couplings of one sort or another, except for close-coupled units, in which the impeller is mounted on an extension of the shaft of the driver.

Because couplings can be used with both centrifugal and positive displacement pumps, they are discussed separately in Section 9.3.

BEDPLATE AND OTHER PUMP SUPPORTS

For very obvious reasons, it is desirable that pumps and their drivers be removable from their mountings. Consequently, they are usually bolted and doweled to machined surfaces

that in turn are firmly connected to a foundation. To simplify the installation of horizontal-shaft units, these machined surfaces are usually part of a common bedplate on which either the pump or the pump and its driver have been prealigned.

Bedplates The primary function of a pump bedplate is to furnish mounting surfaces for the pump feet that can be rigidly attached to the foundation. Mounting surfaces are also necessary for the feet of the pump driver or drivers and for the feet of any independently mounted power transmission device. Although such surfaces could be provided by separate bedplates or by individually planned surfaces, it would be necessary to align these separate surfaces and fasten them to the foundation with the utmost care. Usually, this method requires in-place mounting in the field as well as drilling and tapping for the bolts after all the parts have been aligned. To minimize such field work, coupled horizontal-shaft pumps are usually purchased with a continuous base extending under the pump and its driver. Ordinarily, both these units are mounted and aligned at the place of manufacture.

Although such bases are designed to be quite rigid, they deflect if improperly supported. It is therefore necessary to support them on a foundation that can supply the required rigidity. Furthermore, as the base can be sprung out of shape by improper handling during transit, it is imperative that the alignment be carefully rechecked during erection and prior to starting the unit.

As the unit size increases, so does the size, weight, and cost of the base required. The cost of a prealigned base for most large units exceeds the cost of the field work necessary to align individual bedplates or soleplates and to mount the component parts. Such bases are therefore used only if appearances require them or if their function as a drip collector justifies the additional cost. Even in fairly small units, the height at which the feet of the pump and the other elements are located may differ considerably. A more rigid and nice-looking installation can frequently be obtained by using individual bases or soleplates and building up the foundation to various heights under the separate portions of equipment.

Baseplates are usually provided with a raised edge or raised lip around the base to prevent dripping or draining onto the floor (see Figure 69). The base itself is sloped toward one end to collect the drainage for further disposal. A drain pocket is provided near the bottom of the slope, sometimes with a mesh screen. A tapped connection in the pocket permits piping the drainage to a convenient point.

Bedplates are usually fabricated from steel plate and structural steel shapes (see Figures 70 and 71). Even though most of these fabrications have a drain capability, because of the popular use of mechanical seals and the containment of stuffing box leakage for pumps that continue to use shaft packing (leakage is usually collected in the bearing

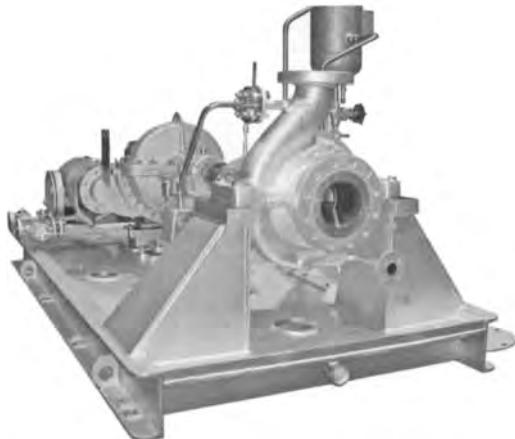


FIGURE 69 Horizontal shaft overhung pump and driver on a structural steel bedplate with a raised edge around the base and a tapped drainage connection (Flowserve Corporation)

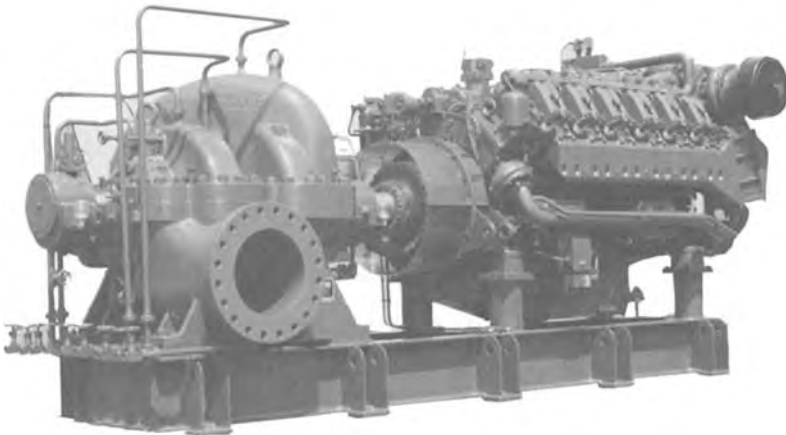


FIGURE 70 Horizontal shaft centrifugal pump and internal combustion engine mounted on a structural steel bedplate (Flowsolve Corporation)

bracket and piped to a common collecting point), the bedplate surfaces actually are seldom used to collect leakage from the pumping equipment during operation. Bedplate drain surfaces are usually employed to contain the leakage of pumpage and other liquids during pump maintenance and removal or in the event of a seal or packing failure.

Soleplates Soleplates are cast-iron or steel pads located under the feet of the pump or its driver and are embedded in the foundation. The pump or its driver is doweled and bolted to them. Soleplates are customarily used for vertical dry-pit pumps and also for some of the larger horizontal units to save the cost of the large bedplates otherwise required.

Centerline Support For operation at high temperatures, the pump casing must be supported as near to its horizontal centerline as possible in order to prevent excessive strains caused by temperature differences. Such strains might seriously disturb the alignment of the unit and eventually damage it. Centerline construction is usually employed in boiler-feed, refinery, and hot-water circulating pumps (see Figure 72). The exact temperature at which centerline support construction becomes mandatory varies from 250°F to 350°F (121°C to 177°C).



FIGURE 71 Horizontal shaft centrifugal pump and driver on a structural steel bedplate made of a simple channel shape (Flowsolve Corporation)



FIGURE 72 Single-stage, double-suction pump with centerline support (Flowsolve Corporation)

Horizontal Units Using Flexible Pipe Connections The previous discussion of bedplates and supports for horizontal-shaft units assumed their application would be to pumps with piping setups that do not impose hydraulic thrusts on the pumps. If flexible pipe connections or expansion joints are desirable in the suction or discharge piping of a pump (or in both), the pump manufacturer should be so advised for several reasons. First, the pump casing will be required to withstand various stresses caused by the resultant hydraulic thrust load. Although this is rarely a limiting or dangerous factor, it is best that the manufacturer have the opportunity to check the strength of the pump casing. Second, the resulting hydraulic thrust has to be transmitted from the pump casing through the casing feet to the bedplate or soleplate and then to the foundation. Usually, horizontal-shaft pumps are merely bolted to their bases or soleplates, and so any tendency to displacement is resisted only by the frictional grip of the casing feet on the base and by relatively small dowels. If flexible pipe joints are used, this attachment may not be sufficient to withstand the hydraulic thrust. If high hydraulic thrust loads are to be encountered, therefore the pump feet must be keyed to the base or supports. Similarly, the bedplate or supporting soleplates must be of a design that will permit transmission of the load to the foundation. Each of these design elements must be checked to confirm their capability to withstand hydraulic thrust loads from flexible pipe connections. Preferably expansion joints will be restrained to avoid transmitting loads to the pump nozzles.

VERTICAL PUMPS

Vertical-shaft pumps fall into two classifications: dry-pit and wet-pit. Dry-pit pumps are surrounded by air, and the wet-pit types are either fully or partially submerged in the liquid handled.

Vertical Dry-Pit Pumps Dry-pit pumps with external bearings include most small, medium, and large vertical sewage pumps, most medium and large drainage and irrigation pumps for medium and high heads, many large condenser circulating and water supply pumps, and many marine pumps. Sometimes the vertical design is preferred (especially for marine pumps) because it saves floor space. At other times, it is desirable to mount a pump at a low elevation because of suction conditions, and it is then also preferable or necessary to have the pump driver at a high elevation. The vertical pump is normally used for large capacity applications because it is more economical than the horizontal type, all factors considered.

Many vertical dry-pit pumps are basically horizontal designs with minor modifications (usually in the bearings) to adapt them for vertical-shaft drive. This is not true of small- and medium-sized sewage pumps, however. In these units, a purely vertical design is the most popular. Most of these sewage pumps have elbow suction nozzles (see Figures 73 and 74) because their suction supply is usually taken from a wet well adjacent to the pit in which the pump is installed. The suction elbow usually contains a handhole with a removable cover to provide easy access to the impeller.

To dismantle one of these pumps, the stuffing box head must be unbolted from the casing after the intermediate shaft or the motor and motor stand have been removed. The rotor assembly is drawn out upward, complete with the stuffing box head, the bearing housing, and the like. This rotor assembly can then be completely dismantled at a convenient location.

Vertical-shaft installations of single-suction pumps with a suction elbow are commonly furnished with either a pedestal or a base elbow (refer to Figure 73), both of which can be bolted to soleplates or even grouted in. The grouting arrangement is not desirable unless there is full assurance that the pedestal or elbow will never be disturbed or that the grouted space is reasonably regular and the grout will separate from the pump without excessive difficulty.

Vertical single-suction pumps with bottom suction are commonly used for larger sewage, water supply, or condenser circulating applications. Such pumps are provided with wing feet that are bolted to soleplates grouted in concrete pedestals or piers (see Figure 75).

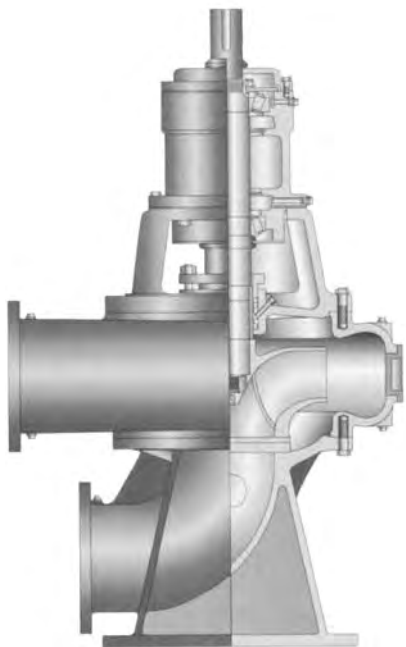


FIGURE 73 Section of a vertical sewage pump with end-suction (elbow) and side discharge (Flowserve Corporation)

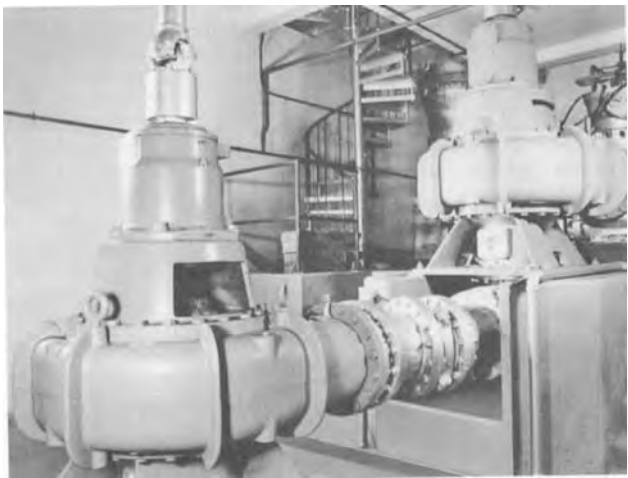


FIGURE 74 An installed vertical sewage pump similar to that shown in Figure 102 (Flowserve Corporation)

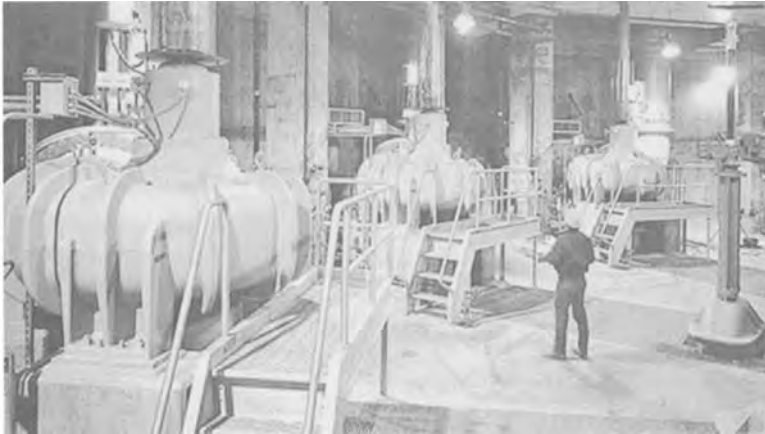


FIGURE 75 Vertical bottom-suction volute pumps installed in a sewage pumping station (Flowserve Corporation)

Sometimes the wing feet may be grouted right in the pedestals. These must be suitably arranged to provide proper access to any handholes in the pump and to allow clearance for the elbow suction nozzles if these are used.

If a vertical pump is applied to a condensate service or some other service for which the eye of the impeller must be vented to prevent vapor binding, a pump with a bottom single-inlet impeller is not desirable because it does not permit effective venting. Neither does a vertical pump employing a double-suction impeller (see Figure 76). The most suitable design for such applications incorporates a top single-inlet impeller (see Figure 77).



FIGURE 76 A vertical double-suction volute pump with a direct-mounted motor (Flowserve Corporation)

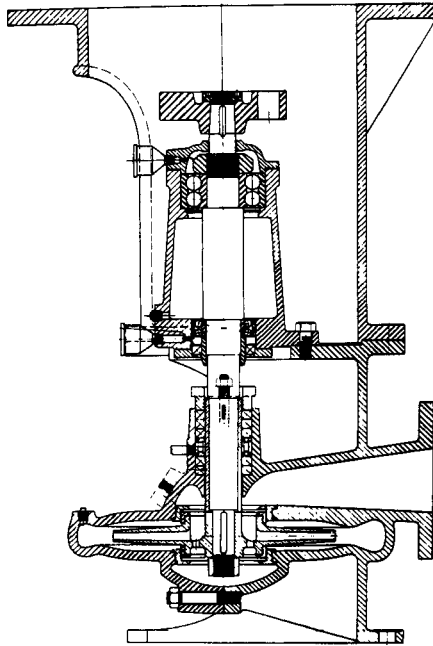


FIGURE 77 A section of a vertical pump with a top-suction inlet impeller (Flowserve Corporation)

If the driver of a vertical dry-pit pump can be located immediately above the pump, it is often supported on the pump itself. The shafts of the pump and driver may be connected by a flexible coupling, which requires that each have its own thrust bearing. If the pump shaft is rigidly coupled to the driver shaft or is an extension of the driver shaft, a common thrust bearing is used, normally in the driver.

Although the driving motors are frequently mounted on top of the pump casing, one important reason for the use of the vertical shaft design is the possibility of locating the motors at an elevation sufficiently above the pumps to prevent the accidental flooding of the motors. The pump and its driver may be separated by an appreciable length of shafting, which may require steady bearings between the two units. Section 9.3.1 discusses the construction and arrangement of the shafting used to connect vertical pumps to drivers located some distance above the pump elevation.

Bearings for vertical dry-pit pumps and for intermediate guide bearings are usually antifriction grease-lubricated types to simplify the problem of retaining a lubricant in a housing with a shaft projecting vertically through it. Larger units, for which antifriction bearings are not available or desirable, use self-oiling, babbit steady bearings with spiral oil grooves (see Figures 78 and 79). Figure 79 illustrates a vertical dry-pit pump design with a single-sleeve line bearing. The pump is connected by a rigid coupling to its motor (not shown), which is provided with a line and a thrust bearing.

Vertical dry-pit centrifugal pumps are structurally similar to horizontal-shaft pumps. It is to be noted, however, that many of the large, vertical, single-stage, single-suction (usually bottom) volute pumps that are preferred for large storm water pumpage, drainage, irrigation, sewage, and water supply projects have no comparable counterpart among horizontal-shaft units. The basic U-section casing of these pumps, which is structurally weak, often requires the use of heavy ribbing to provide sufficient rigidity. In the comparable water turbine practice, a set of vanes (called a *speed ring*) is employed between the casing and the runner to act as a strut. Although the speed ring does not adversely affect the operation of a water turbine, it would function basically as a diffuser in a pump because of the inherent hydraulic

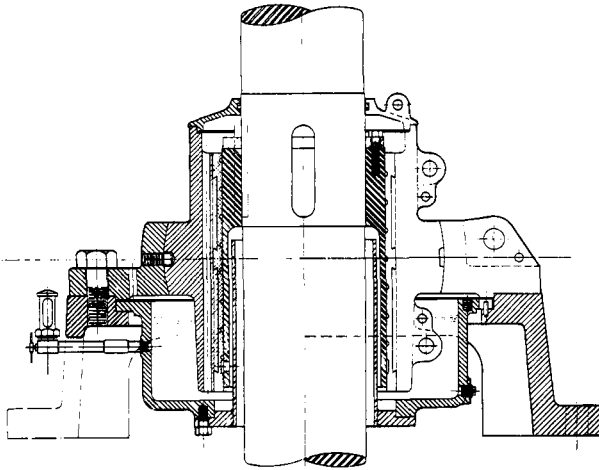


FIGURE 78 A self-oiling, babbitt steady bearing for large vertical shafting (Flowserve Corporation)

limitations of that construction. Some high-head pumps of this type have been made in the twin-volute design. The wall separating the two volutes acts as a strengthening rib for the casing, thus making it easier to design a casting strong enough for the pressure involved.

Bases and Supports for Vertical Pumping Equipment Vertical-shaft pumps, like horizontal-shaft units, must be firmly supported. Depending upon the installation, the unit can be supported at one or several elevations. Vertical units are seldom supported from walls, but even that type of support is sometimes encountered.

Occasionally, a nominal horizontal-shaft pump design is arranged with a vertical shaft and a wall used as the supporting foundation. Regular horizontal-shaft units can be used for this purpose without modification, except that the bedplate is attached to a wall. Careful

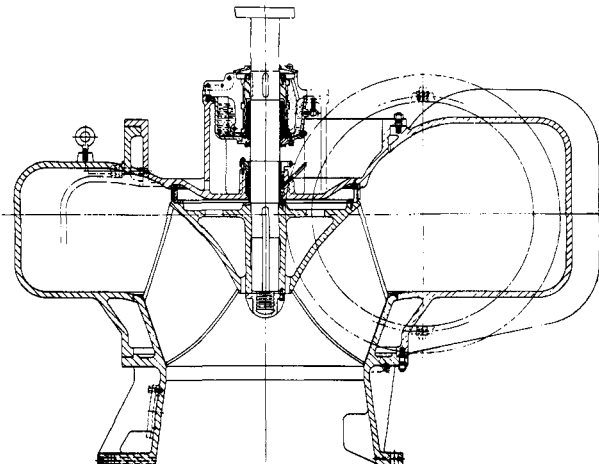


FIGURE 79 A section of a large, vertical, bottom-suction volute pump with a single sleeve bearing (Flowserve Corporation)

attention must be given to the arrangement of the pump bearings to prevent the escape of the lubricant. Installations of double-suction, single-stage pumps with the shaft in the vertical position are relatively rare, except in some marine or navy applications. Hence, manufacturers have few standard pumps of this kind arranged so that a portion of the casing forms the support (to be mounted on soleplates). Figure 76 shows such a pump, which also has a casing extension to support the driving motor.

Vertical Wet-Pit Pumps Vertical pumps intended for submerged operations are manufactured in a great number of designs, depending mainly upon the service for which they are intended. Small pumps of this type are often referred to as *sump pumps*. Wet-pit centrifugal pumps can be classified in the following manner:

- Vertical turbine pumps
- Propeller or modified propeller pumps
- Volute pumps

Vertical Turbine Pumps Vertical turbine pumps were originally developed for pumping water from wells and have been called *deep-well pumps*, *turbine well pumps*, and *bore-hole pumps*. As their application to other fields has increased, the name *vertical turbine pumps* has been generally adopted by manufacturers. This is not too specific a designation because the term *turbine pump* has been applied in the past to any pump employing a diffuser. There is now a tendency to designate pumps using diffusion vanes as *diffuser pumps* to distinguish them from *volute pumps*. As that designation becomes more universal, applying the term *vertical turbine pumps* to the construction formerly called *turbine well pumps* will become more specific.

The largest fields of application for the vertical turbine pump are pumping from wells for irrigation and other agricultural purposes, for a municipal water supply, and for industrial water supplies, as well as for processing, circulating, refrigerating, and air conditioning. This type of pump has also been utilized for brine pumping, mine dewatering, oil field repressuring, and other purposes.

These pumps have been made for capacities as low as 10 or 15 gpm (2 or 3 m³/h) and as high as 25,000 gpm (5700 m³/h) or more and for heads up to 1000 feet (300 m). Most applications naturally involve the smaller capacities. The capacity of the pumps used for bored wells is naturally limited by the size of the well as well as by the rate at which water can be drawn without lowering its level to a point of insufficient pump submergence.

Vertical turbine pumps should be designed with a shaft that can be readily raised or lowered from the top to permit proper positioning of the impeller in the bowl. An adequate thrust bearing is also necessary to support the vertical shafting, the impeller, and the hydraulic thrust developed when the pump is in service. As the driving mechanism must also have a thrust bearing to support its vertical shaft, it is usually provided with one large enough to carry the pump parts as well. For these two reasons, the hollow-shaft motor or gear is most commonly used for vertical turbine pump drives. In addition, these pumps are sometimes made with their own thrust bearings to allow for a belt drive or for a drive through a flexible coupling by a solid-shaft motor, gear, or turbine. Dual-driven pumps usually employ an angle gear with a vertical motor mounted on its top.

The design of vertical pumps illustrates how a centrifugal pump can be specialized to meet a specific application. Figure 80 illustrates a turbine design with closed impellers and enclosed-line shafting and another turbine design with closed impellers and open-line shafting.

The bowl assembly, or section, consists of the suction case (also called *suction head* or *inlet vane*), the impeller or impellers, the discharge bowl, the intermediate bowl or bowls (if more than one stage is involved), the discharge case, the various bearings, the shaft, and the miscellaneous parts, such as keys, impeller-locking devices, and the like. The column pipe assembly consists of the column pipe, the shafting above the bowl assembly, the shaft bearings, and the cover pipe or bearing retainers. The pump is suspended from the driving head, which consists of the discharge elbow (for above ground discharge), the motor or driver and support, and either the stuffing box (in an open-shaft construction) or the assembly

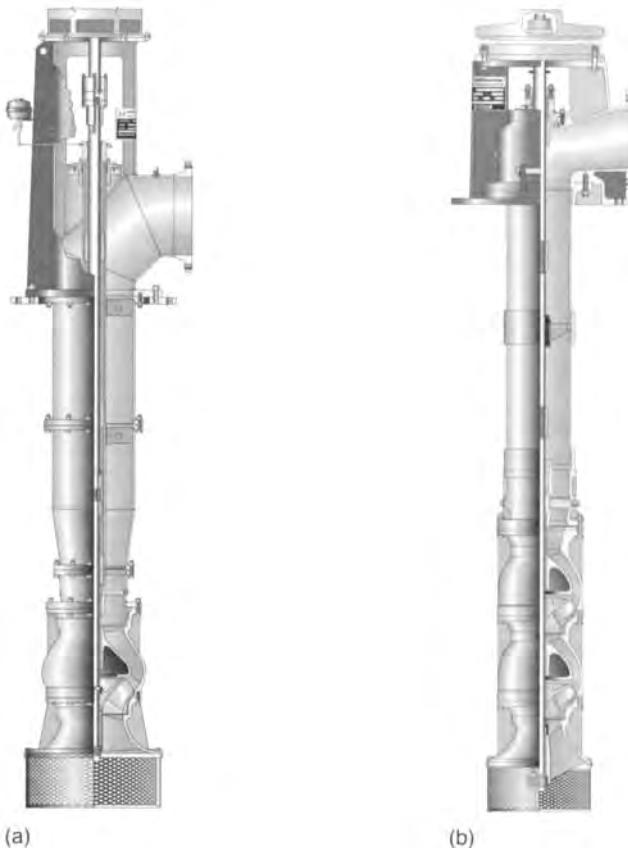


FIGURE 80 A vertical turbine pump design with enclosed impellers and; (a) enclosed line shafting and (b) open-line shafting (Flowsolve Corporation)

for providing tension on the cover pipe and introducing a lubricant into it. Below ground discharge is taken from a tee in the column pipe, and the driving head functions principally as a stand for the driver and a support for the column pipe.

Liquid in a vertical turbine pump is guided into the impeller by the suction case or head. This may be a tapered section for the attachment of a conical strainer or suction pipe, or it may be a bell mouth.

Semiopen and enclosed impellers are both commonly used. For proper clearances in the various stages, the semiopen impeller requires more care in assembly on the impeller shaft and more accurate field adjustments of the vertical shaft position in order to obtain the best efficiency. Enclosed impellers are favored over semiopen ones because wear on the latter reduces capacity, which cannot be restored unless new impellers are installed. Normal wear on enclosed impellers does not affect impeller vanes, and worn clearances may be restored by replacing wearing rings. The thrust produced by semiopen impellers may be as much as 150% of that by enclosed impellers.

Occasionally, in power plants, the maximum water level that can be carried in the condenser's hot well will not give adequate NPSH for a conventional horizontal condensate pump mounted on the basement floor, especially if the unit has been installed in a space originally allotted for a smaller pump. Building a pit for a conventional horizontal condensate



FIGURE 81 Vertical turbine can pump for condensate service (Flowserve Corporation)

pump or a vertical dry-pit pump that will provide sufficient submergence involves considerable expense. Pumps of the design shown in Figure 81 have become quite popular in such applications. This is basically a vertical turbine pump mounted in a tank (often called a can) that is sunk into the floor. The length of the pump has to be such that sufficient NPSH will be available for the first-stage impeller design, and the diameter and length of the tank have to allow for proper flow through the space between the pump and tank and then for a turn and flow into the bell mouth. Installing this design in an existing plant is naturally much less expensive than making a pit because the size of the hole necessary to install the tank is much smaller. The same basic design has also been applied to pumps handling volatile liquids that are mounted on the operating floor and not provided with sufficient NPSH.

Propeller Pumps Originally, the term *vertical propeller pump* was applied to vertical wet-pit diffuser or turbine pumps with a propeller or axial-flow impellers, usually for installation in an open sump with a relatively short setting (see Figures 82 and 83). Operating heads exceeding the capacity of a single-stage axial-flow impeller might call for a pump of two or more stages or a single-stage pump with a lower specific speed and a mixed-flow impeller. High enough operating heads might demand a pump with mixed-flow impellers and two or more stages. For lack of a more suitable name, such high-head designs have usually been classified as propeller pumps also.

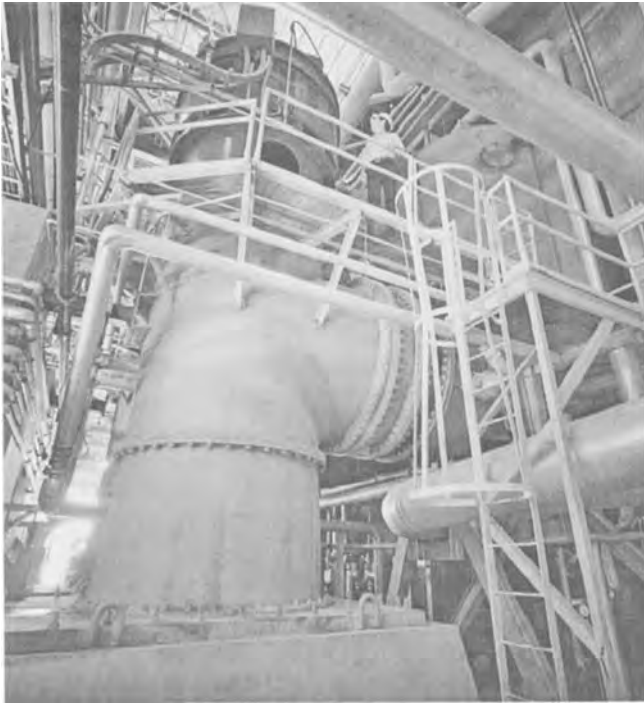


FIGURE 82 Vertical propeller pump installed (Flowserve Corporation)

Although vertical turbine pumps and vertical modified propeller pumps are basically the same mechanically and could even be of the same specific speed hydraulically, a basic turbine pump design is suitable for a large number of stages. A modified propeller pump design, however, is basically intended for a maximum of two or three stages.

Most wet-pit drainage, low-head irrigation, and storm water installations employ conventional propeller or modified propeller pumps. These pumps have also been used for condenser circulating services, but a specialized design dominates this field. As large power plants are usually located in heavily populated areas, they frequently have to use badly contaminated water (both fresh and salt) as a cooling medium. Such water quickly shortens the life of fabricated steel. Cast iron, bronze, or an even more corrosion-resistant cast metal must therefore be used for the column pipe assembly. This requirement means a very heavy pump if large capacities are involved. To avoid the necessity of lifting this large mass for maintenance of the rotating parts, some designs (one of which is illustrated in Figure 84) are built so that the impeller, diffuser, and shaft assembly can be removed from the top without disturbing the column pipe assembly. These designs are commonly designated as *pullout* designs.

Like vertical turbine pumps, propeller and modified propeller pumps have been made with both open- and enclosed-line shafting. Except for condenser circulating services, enclosed shafting, using oil as a lubricant but with a grease-lubricated tail bearing below the impeller, seems to be favored. Some pumps handling condenser circulating water use enclosed shafting but with water (often from another source) as the lubricant, thus eliminating any possibility of oil getting into the circulating water and coating the condenser tubes.

Propeller pumps have open propellers. Modified propeller pumps with mixed-flow impellers are made with both open and closed impellers.



FIGURE 83 Vertical propeller pump with above ground discharge (Flowserve Corporation)



FIGURE 84 Vertical pull-out design allows the rotating element and critical non-rotating wear components to be removed for inspection and replacement without removing the complete pump (Flowserve Corporation)

Volute Pumps A variety of wet-pit pumps are available. The liquid pumped, be it clean water, sewage, abrasive liquids or slurries, dictates whether a semiopen or an enclosed impeller will be used, whether the shafting will be open or closed to the liquid pumped, and whether the bearings will be submerged or located above the liquid.

Figure 85 illustrates a single-volute pump with a single-suction enclosed nonclog impeller, no pump-out vanes or wearing-ring joints on the back side of the impeller, and enclosed shafting. The pump is designed to be suspended from an upper floor by means of a drop pipe and for pumping sewage or other solid-laden liquids. To seal against leakage along the shaft at the point where it passes through the casing, a seal chamber or a stuffing box is provided. The design of a stuffing box can be either like the one shown in Figure 86, which uses rings of packing and a spring-loaded gland, or the one shown in Figure 87, which uses U-cup packing requiring no gland. The pump shown in Figure 85 uses two

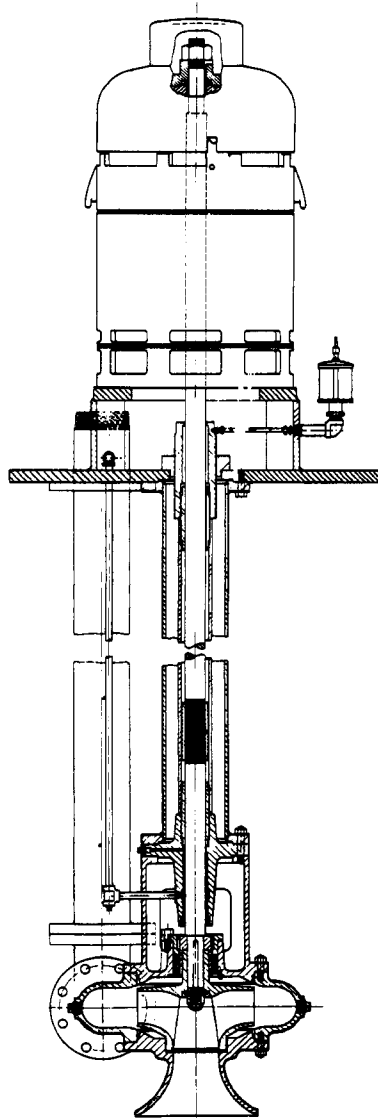


FIGURE 85 A section of a vertical wet-pit, nonclogging pump (Flowserve Corporation)

sleeve bearings above the impeller. The bottom bearing is grease-lubricated, and a seal is provided to prevent grease leakage as well as to keep out any grit. The upper bearing connects the shaft cover pipe to the pump-bearing bracket and the upper end of the cover pipe to the floorplate. This bearing is gravity-feed oil-lubricated. If intermediate bearings are required, they are also supported by the cover pipe and are oil-lubricated. The pump thrust is carried by the motor, which can be either a hollow-shaft or a solid-shaft construction. The latter type requires the use of a rigid coupling between the pump and motor shafts.

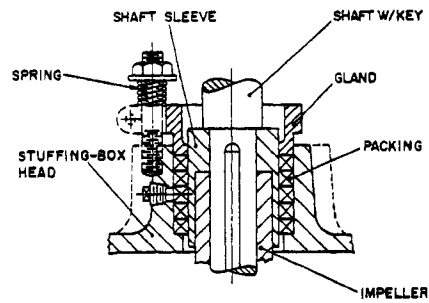
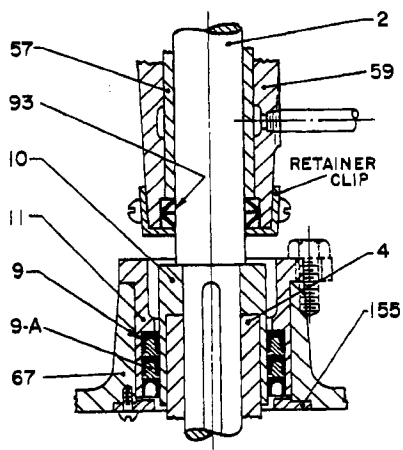


FIGURE 86 A typical stuffing box arrangement

In most applications, these volute-type pumps have been replaced with vertical wet-pit pumps with the stuffing box/seal chamber in the discharge head.

A design that uses open shafting and no seal chamber or stuffing box at the pump casing, incorporating its own thrust bearing, is shown in Figure 88. It may also offer a recessed impeller option, suitable for pumping heavy concentrations of solid material (such as sludges or slurries) or in certain food-processing applications. Pumped liquid leakage from the casing is relieved back to the suction through holes in the support pipe. The seal chamber or stuffing box at the driver floor elevation is used only when gas tight construction is desired. The lower and any intermediate sleeve bearings are grease-lubricated as shown, but gravity-feed oil lubrication is also available in other designs. The upper antifriction thrust bearing is grease-lubricated. A solid shaft motor and a flexible shaft are used.

Figure 89 illustrates what is called a *cantilever-shaft pump*, which has the unique feature of having no bearings below the liquid surface. The shaft is exposed to the liquid pumped. External antifriction grease-lubricated bearings are provided above the floor and are properly spaced to support the rigid shaft. They carry both the thrust and the radial load. A flexible coupling is used between the pump and the solid-shaft motor. The stuffing box at the floorplate may be eliminated if holes are provided in the drop pipe to maintain the liquid level in the pipe even with the sump liquid level. Either semiopen or enclosed impellers may be used.



STATIONARY PARTS		
REF. NO.	NO. OF PCS.	NAME OF PARTS
9	3	U-CUP SEAL RING
9-A	2	SPACERS
11	1	SEAL RING ADAPTER
57	1	LOWER PUMP BEARING
59	1	BEARING BRACKET
67	1	STUFFING-BOX HEAD
93	2	OIL SEAL
155	1	SEAL-RING RETAINER

ROTATING PARTS		
REF. NO.	NO. OF PCS.	NAME OF PARTS
2	1	SHAFT W/KEY
4	1	IMPELLER
10	1	SHAFT SLEEVE

FIGURE 87 A stuffing box arrangement with U-cup packing (Flowserve Corporation)



FIGURE 88 A vertical wet-pit volute pump with open shafting, no seal chamber (stuffing box), and its own thrust bearing (Flowserve Corporation)

An interesting design of the wet-pit pump is shown in Figure 90. It uses a single-stage, double-suction impeller in a twin-volute casing. Because the axial thrust is balanced, the thrust bearing need carry only the weight of the rotating element. The pump requires no stuffing box or mechanical seal. The shaft is entirely enclosed, and the bearings are externally lubricated, either with oil or with water. The lower bearing receives its lubrication from an external pipe connection.

The term sump pump ordinarily conveys the idea of a vertical wet-pit pump that is suspended from a floorplate or sump cover. It could be supported by a foot on the bottom of a well, be motor-driven and automatically controlled by a float switch, and be used to remove drainage collected in a sump. The term does not indicate a specific construction, for both diffuser and volute designs are used. These may be single-stage or multistage and have open or closed impellers of a wide range of specific speeds.

For small capacities driven by fractional-horsepower motors, *cellar drainers* can be used. These are small and usually single-stage volute pumps with single-suction impellers (either top or bottom suction) supported by a foot on the casing. The motor is supported well above the impeller by some form of column enclosing the shaft. These drainers are made as complete units, including float, float switch, motor, and strainers.

The larger sump pumps are usually standardized but obtainable in any length, with covers of various sizes (on which a float switch may be mounted) and the like. Duplex units, that is, two pumps on a common sump cover (sometimes with a hole for access to the sump), are often used. Such units may operate their pumps in a fixed order, or a mechanical or electric alternator may be used to equalize their operation.

The Application of Vertical Wet-Pit Pumps Like all pumps, the vertical wet-pit pump has advantages and disadvantages. One advantage is that installation does not require a separate dry pit to collect the pumped liquid. If the impeller (first-stage impeller in multistage pumps) is submerged, no priming problem exists and the pump can be automatically controlled without fear of its ever running dry. Moreover, the available NPSH is greater (except in closed tanks) and often permits a higher rotative speed for the same

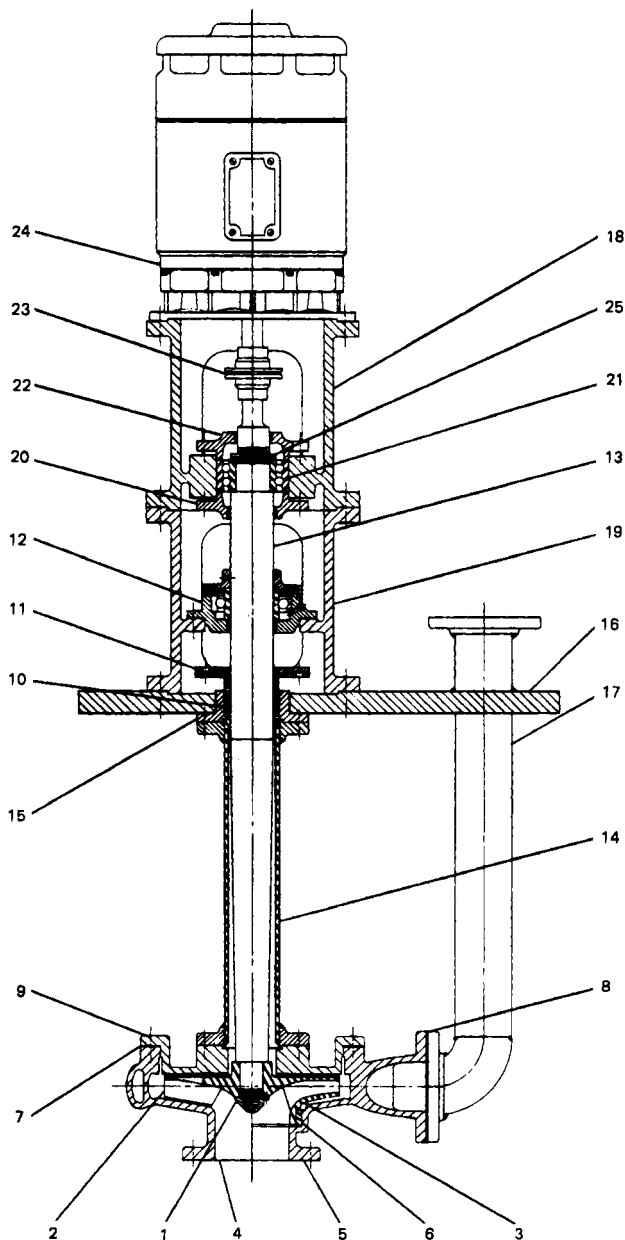


FIGURE 89 Vertical wet-pit cantilever-type volute pump: (1) impeller nut, (2) open impeller, (3) closed impeller, (4) open casing, (5) closed casing, (6) casing wearing ring, (7) casing gasket, (8) discharge flange gasket, (9) back cover, (10) packing or mechanical seal, (11) packing or seal gland, (12) radial bearing, (13) shaft, (14) supporting pipe, (15) separate stuffing box, (16) sump cover, (17) discharge pipe, (18) upper motor pedestal, (19) lower motor pedestal, (20) lower thrust bearing cover, (21) thrust bearing, (22) upper thrust bearing cover, (23) coupling, (24) motor, (25) bearing locknut and washer (Laurence Pump)



FIGURE 90 Double-suction, wet-pit pump (Flowserve Corporation)

service conditions. A second advantage is that the motor or driver can be located at any desired height above any flood level.

It has the following mechanical disadvantages: (1) the possibility of freezing when idle, (2) the possibility of damage by floating objects if the unit is installed in an open ditch or similar installation, (3) the inconvenience of lifting out and dismantling for inspection and repairs, no matter how small, and (4) the pump bearings have a relative short life unless the water and bearing design are ideal. In summary, the vertical wet-pit pump is the best pump available for some applications. It's not ideal but can be the most economical for certain installations, a poor choice for some, and the least desirable for still others.

Axial Thrust in Vertical Pumps with Single Suction Impellers The subject of axial thrust in horizontal pumps has already been discussed in this section. When pumps are installed in a vertical position, additional factors need be taken into account when determining the amount and direction of the thrust to be absorbed in the thrust bearings.

The first and most significant of these factors is the weight of the rotating parts, which is a constant downward force for any given pump, completely independent of the pump operating capacity or total head. Since in most cases the single-suction impellers of vertical pumps are mounted with the suction eye facing downward, the normal hydraulic axial thrust is exerted downward and the weight of the rotating parts is additive to this axial thrust.

The second factor involves the dynamic force (or change in momentum) caused by the change in the direction of flow, from vertical to either horizontal or partly horizontal, as the pumped liquid flows through the impeller. This force acts upward and balances a small portion of the hydraulic downthrust and of the rotor weight. The magnitude of this force for water having a specific weight (force) of 62.34 lb/ft³ (9.79 kN/m³) is

$$\text{in USCS units} \quad F_u = \frac{KV_e^2 A_e}{2g(2.31)}$$

$$\text{in SI units} \quad F_u = \frac{KV_e^2 A_e}{2g(1.02)}$$

where F_u = upthrust, lb (N)

K = constant related to the impeller type

V_e = velocity in the impeller eye ft/s (m/s)

A_e = net eye area, in² (cm²)

g = 32.2 ft/s² (9.81 m/s²)

The constant K is 1.0 for a fully radial-flow impeller, less than 1.0 for a mixed-flow impeller, and essentially 0 for a fully axial-flow impeller.

Under normal operating conditions, the upward thrust caused by the change of momentum is hardly significant in comparison with the downward thrust caused by the

unbalanced pressures acting on the single-suction impeller. Consider the example of an impeller with the following characteristics:

Capacity	2500 gpm (568 m ³ /h)
Total head	231 ft (70.4 m)
Net pressure	100 lb/in ² (6.89 bar)
Unbalanced eye area	40 in ² (258 cm ²)

If we neglect the effect of the pressure distribution on the shrouds of the impeller, the downward thrust is

$$\text{in USCS units} \quad 100 \times 40 = 4000 \text{ lb}$$

$$\text{in SI units} \quad 6.89 \times 10^5 \times \frac{258}{10,000} = 17,800 \text{ N}$$

The upward force caused by the change of momentum is

$$\text{in USCS units} \quad \frac{1.0 \times \left(\frac{2500 \times 0.32}{40} \right)^2 \times 40}{64.4 \times 2.31} = 107.5 \text{ lb}$$

$$\text{in SI units} \quad \frac{1 \times \left(\frac{568 \times 10,000}{258 \times 3600} \right)^2 \times 258}{2 \times 9.81 \times 1.02} = 481 \text{ N}$$

This is certainly negligible relative to the downward hydraulic axial thrust.

The situation is quite different, however, during the startup of a vertical pump. Although the motor may get up to full speed in just a few seconds, it takes a certain amount of time for the total head to increase from zero that corresponding to the normal operating capacity. Consequently, the pump will be operating in a very high capacity range. Since the upward force caused by the change of momentum varies as the square of the capacity while the downward axial thrust caused by pressure differences is very low, there can be a momentary net upward force, or upthrust. This means that thrust bearings intended to accommodate the axial thrust of vertical pumps must be capable of accommodating some thrust in the upward direction in addition to the normal downward thrust. This is particularly true for pumps with relatively low heads per stage and with short settings because in these units the rotor weight does not at all times compensate for the upthrust from the change in momentum.

Particular care must be exercised in defining the range of vertical movement allowed for the thrust bearings because any such movement must remain within the displacement limits of any mechanical seal used in the pump.

In rather rare cases, a close-coupled vertical pump is operated at such a high capacity that a continuous net upthrust is generated. Such operations can damage the pump because the line shaft is operated in compression and may buckle, causing vibration and bearing wear. The manufacturer's comments should be invited if such an operation is contemplated.

Shaft Elongation in Vertical Pumps The elongation of a vertical pump shaft is caused by three separate phenomena: (1) the tensile stress caused by the weight of the rotor, (2) the tensile stress caused by the axial thrust, and (3) the thermal expansion of the shaft.

In most cases, the tensile stress created by the axial thrust is several times greater than that created by the weight. In a typical example of a 16,000-gpm (3636-m³/h) pump designed for a 175-ft (53.3-m) head and 50 ft (15.25 m) long, the elongation caused by the weight of a 1600-lb (726-kg) impeller will be of the order of 0.0033 in (0.084 mm). The elongation caused by the axial thrust will be approximately 0.0315 in (0.8 mm).

The elongation caused by thermal expansion has to be considered from two angles. First, if the shaft and the stationary parts are built of materials that have essentially

the same coefficient of expansion, both will expand equally and no significant relative elongation will take place. Second, if the pumps are operated in essentially the same range of temperatures in which they were assembled, no significant relative expansion will take place, even if dissimilar coefficients of expansion are involved. Whatever the case, the pump manufacturers take these factors into consideration by providing the necessary vertical end-play between the stationary and rotating pump components.

Loads on Foundations of Vertical Pumps If the motor support is integral with the pump discharge column, as in Figures 80, 82, and 83, and if the hydraulic thrust is carried by the motor thrust bearing, this thrust is not additive to the deadweight of the pump and of its motor plus the weight of the water contained in the pump, insofar as the load on the foundations is concerned. This is because the pump and motor mounted in this fashion form a self-contained entity and all internal forces and stresses are balanced within this entity. If the pump and motor are supported separately, however, as in Figure 91, and are joined by a rigid coupling that transmits the pump hydraulic

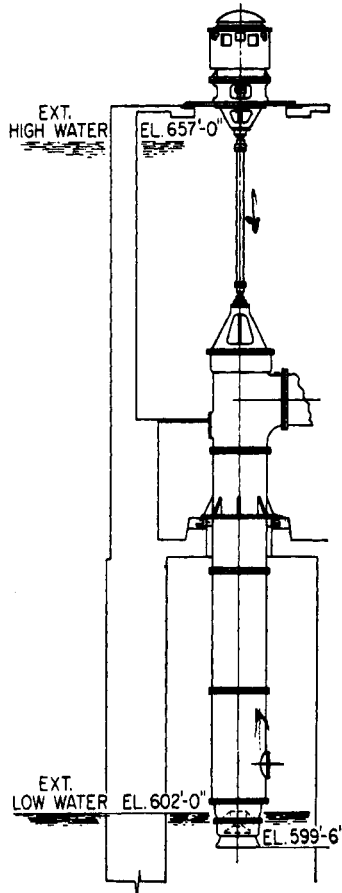


FIGURE 91 The vertical pump and driving motor supported separately (Flowserve Corporation)

thrust to the motor thrust bearing, the foundations will carry the following loads when the pump is running:

- *The foundation supporting the motor:* the weight of the motor plus the weight of the pump rotor plus the axial hydraulic thrust developed by the pump
- *The foundation supporting the pump:* the weight of the pump stationary parts plus the weight of the water in the pump, including the water in the discharge pipe supported by the foundation, less the axial hydraulic thrust developed by the pump

Since when the pump is idle the foundation supporting the pump is not benefited by the reduction in load equivalent to the axial hydraulic thrust, both running and idle operating conditions must be considered in this case.

Typical Arrangements of Vertical Pumps A pump is only part of a pumping system. The hydraulic design of the system external to the pump will affect the overall economy of the installation and can easily have an adverse effect upon the performance of the pump itself. Vertical pumps are particularly susceptible because the small floor space occupied by each unit offers the temptation to reduce the size of the station by placing the units closer together. If the size is reduced, the suction arrangement may not permit the proper flow of water to the pump suction intake. Recommended arrangements for vertical pumps are discussed in Section 13.1.

SPECIAL PURPOSE PUMPS

In addition to the more or less general purpose pumps described on the preceding pages, literally hundreds of centrifugal pumps are intended for very specific applications. Although it is impossible to describe every one of these special purpose designs, many of them are discussed and illustrated in Section 12, covering a variety of pump services. However, several specific designs are seeing rapidly increasing usage and so are discussed in greater detail here.

Submersible Motor-Driven Wet-Pit Pumps The installation of conventional vertical wet-pit pumps with the motor located above the liquid level may require a considerable length of drive shafting, particularly in the case of deep settings. The addition of this shafting, of the many line bearings, and possibly of an external lubrication system may represent a major portion of the total installed cost of the pumping unit. Furthermore, shaft alignment becomes more critical, and shaft elongation and power losses increase rapidly as the setting is increased, especially for deep-well pumps.

A great variety of submersible motors have been developed to obviate these shortcomings. They are described in Section 9.1.1, and a classification of the various types of such motors is presented in that section. Submersible wet-pit pumps eliminate the need for extended shafting, shaft couplings, a mechanical seal or stuffing box, a subsurface motor stand, and, in some cases, an expensive pump house. Both vertical turbine and volute-type wet-pit pumps may be so driven.

Figures 92 and 93 illustrate, respectively, the external appearance and a cross section of a vertical turbine pump driven by a submersible motor located at the bottom of the pump. The pump suction is through a perforated strainer located between the motor and the first-stage impeller bowl. There is, of course, no shafting above the pump and the pump-and-motor unit is supported by the discharge pipe only. No external lubrication is required. The motor is completely enclosed and oil-filled and is provided with a thrust bearing to carry the pump downthrust. A mechanical seal is provided at the motor shaft extension, which is connected to the pump shaft with a rigid coupling. Only a discharge elbow and the electric cable connection are seen above the surface support plate. On occasion, this type of pump is used horizontally as a booster pump in a pipeline, and in such cases the elbow at the discharge is eliminated.

Vertical wet-pit volute-type sump pumps can also be obtained with close-coupled submersible motors (see Figure 94) for drainage, sewage, process, and slurry services.

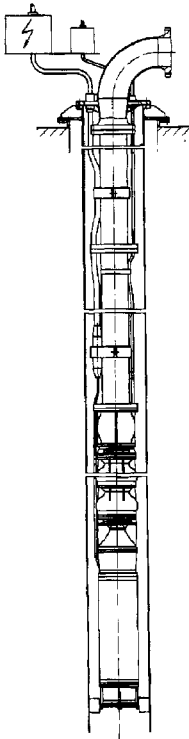


FIGURE 92 Vertical turbine pump driven by a submersible motor (Flowsolve Corporation)

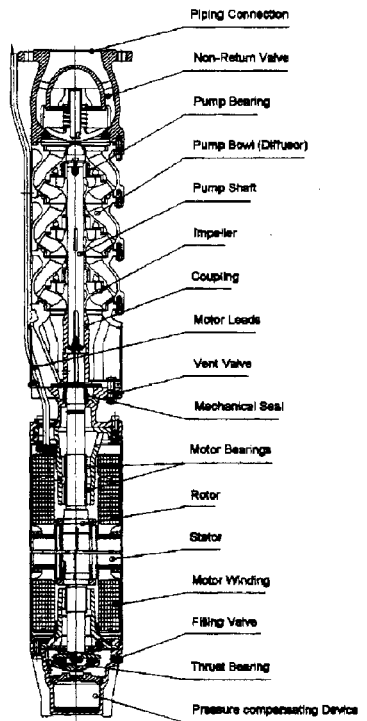


FIGURE 93 Cross-sectional view of a submersible pump (Flowsolve Corporation)

Small portable pumps are available with flexible discharge hoses and built-in water-level motor control switches activated by trapped air pressure. Motors for these pumps are usually oil-filled and have a single mechanical shaft seal but are also available in a hermetically sealed design. The submersible motors are cooled by the liquid in which they are immersed and therefore should not be run dewatered, although some motors can operate for short periods (10 to 15 min) this way.

In a pump-motor combination (see Figure 95), the motor is cooled by the pumped liquid as it moves through a passage around the sides of the motor. This design also uses a pressurized oil-seal chamber to assure positive sealing.

Sealless Pumps Sealless pumps are available for pumping corrosive, volatile, radioactive, and otherwise hazardous liquids. Canned motor pumps are an assembly of a standard centrifugal pump and a squirrel-cage induction motor in a hermetically sealed unit. Modifying a recirculating flow system in a canned motor pump can allow it to be used in applications at up to 1000°F (538°C).

Magnetic drive pumps are an assembly of a rotor, an impeller, product lubricated bearings, and a magnetic carrier inside an isolation shell or diaphragm. This rotor is driven by magnets outside the shell or diaphragm. No mechanical connection exists between the driven magnets and the driving magnets. No seals exist and thus we have the term "sealless." Sealless pumps are described in detail in Section 2.2.3.

Straight-Radial-Vane High-Speed Pumps For handling volatile liquids at low flow rates and high heads, the straight-radial-vane impeller in a diffuser casing offers several

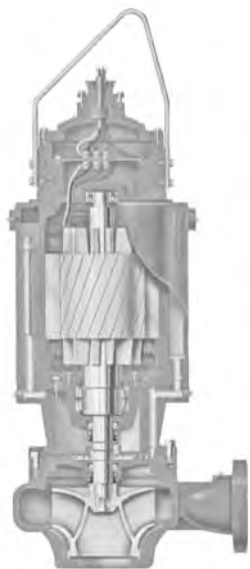


FIGURE 94 A section of a submersible, solids handling pump (Flowserve Corporation)

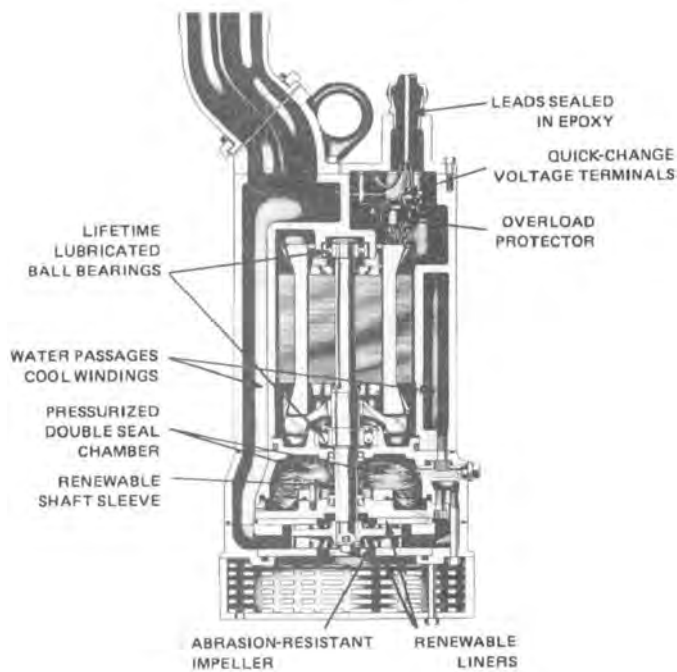


FIGURE 95 A portable submersible pump (Peabody Barnes)

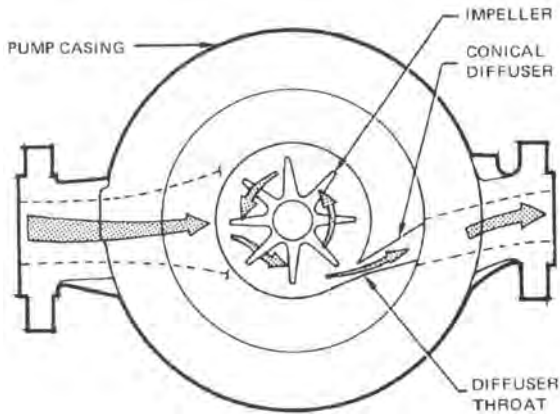


FIGURE 96 A straight-radial-vane impeller in a single emission-point, conical-diffuser circular casing (Sundyne Corporation)

advantages. Volatile, low-specific-gravity, poor-lubricity liquids require larger running clearances, which is not possible with conventional high-head multistage centrifugal or positive displacement pumps. High-head pumping of these liquids can be handled by operating this completely open impeller at very high speeds through an integral gear increaser and with very large impeller-to-casing clearance, typically 0.030 to 0.070 in (0.76 to 1.8 mm). Tests of one manufacturer's design have shown this clearance can increase to 0.125 in (3.2 mm) with virtually no change in performance, and consequently there is no need to provide adjustment for impeller axial clearance. A pump of this design can also run "dry," as liquid lubricity is not required to lubricate the bearings, which can be separately lubricated (providing the mechanical seal is lubricated).

Figure 96 illustrates one manufacturer's design of a radial-vane high-speed pump. The impeller rotates in a circular casing that has a single emission point leading to a conical diffusion section. The advantage of this type of casing is that the conversion to pressure occurs outside the circular housing, thus eliminating any recirculation forces that would require a close clearance between the impeller and casing. For higher flow ranges, a double emission point design, as shown in Figure 97, is used. This additional emission acts like a double-volute casing in conventional centrifugal pumps.

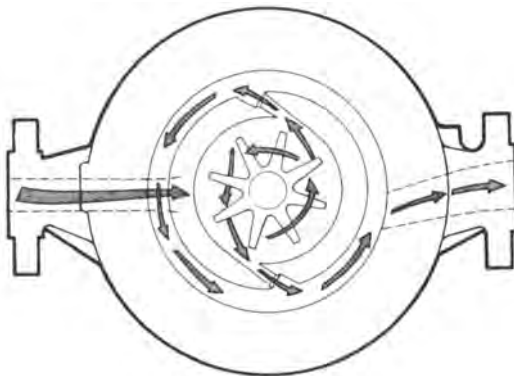


FIGURE 97 A straight-radial-vane impeller in a double-emission-point, conical-diffuser circular casing (Sundyne Corporation)

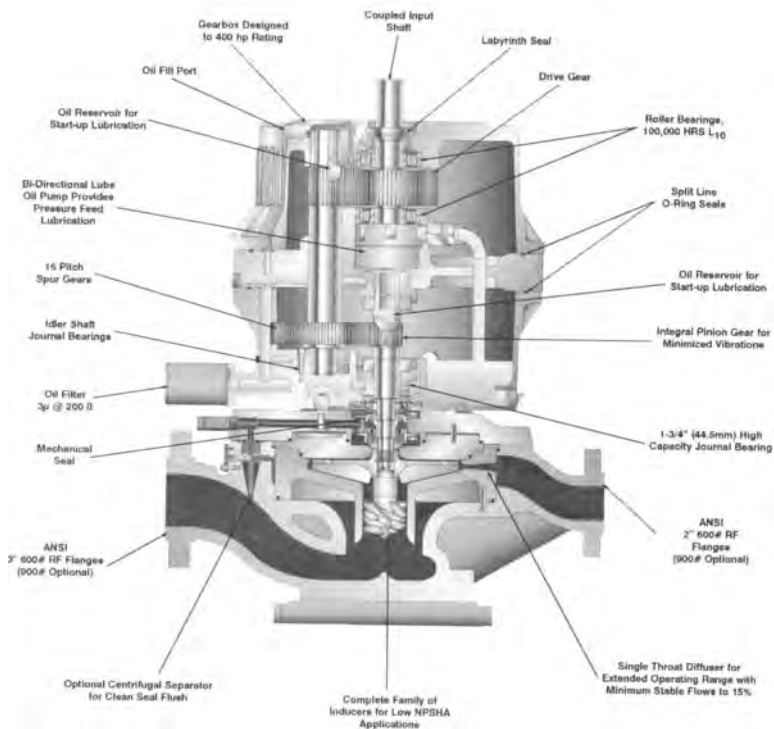


FIGURE 98 A section of a straight-radial-vane impeller and inducer in a conical-diffuser circular casing with an integral gear increaser with the following: (1) pump casing, (2) impeller, (3) impeller bolt, (4) impeller tab washer, (5) inducer, (6) diffuser, (7) diffuser cover, (8) upper throttle bushing, (9) seal housing, (10) single shaft sleeve, (11) pump seal rotating face, (12) gearbox seal rotating face, (13) pump mechanical seal, (14) gearbox mechanical seal, (15) separator orifice, (16) separator fitting, (17) gearbox output housing, (18) gearbox input housing, (19) bearing plate, (20) interconnecting shaft, (21) low-speed shaft, (22) input gear, (23) lower idler gear, (24) high-speed shaft, (25) pinion gear, (26) idler shaft, (27) O-ring packings (Sundyne Corporation).

Figure 98 is a sectional view of this pump with an integral speed increaser. Because of the high rotational speed, a single-stage pump can achieve the head normally associated with multistage centrifugal or positive displacement pumps. With this smaller liquid end and an integral gear box, cost and space savings can be appreciable.

2.2.2

CENTRIFUGAL PUMP PRIMING

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When a pump is first put into service, its passageways are filled with air. If the suction supply is above atmospheric pressure, this air will be trapped in the pump and compressed somewhat when the suction valve is opened. Priming is accomplished by venting the entrapped air out of the pump through a valve provided for this purpose. A centrifugal pump is primed when the passageways of the pump are filled with the liquid to be pumped. The liquid replaces the air, gas, or vapor in the passageways. This may be done manually or automatically.

Unlike a positive displacement pump, a centrifugal pump that takes its suction from a supply located below the pump, which is under atmospheric pressure, cannot start and prime itself (unless designed to be self-priming, as described later in this section). At its rated capacity, a positive displacement pump will develop the necessary pressure to exhaust air from its chambers and from the suction piping. Centrifugal pumps can also pump air at their rated capacity, but only at a pressure equivalent to the rated head of the pump. Because the specific weight of air is approximately $\frac{1}{800}$ that of water, a centrifugal pump can produce only $\frac{1}{800}$ of its rated liquid pressure. For every 1 ft (1 m) water has to be raised to prime a pump, the pump must produce a discharge head of air of approximately 800 ft (m). It is therefore apparent that the head required for a conventional centrifugal pump to be self-priming and to lift a large column of liquid (and in some cases to discharge against an additional static liquid head) when pumping air is considerably greater than the rating of the pump. Centrifugal pumps that operate with a suction lift can be primed by providing: (1) a foot valve in the suction line, (2) a single-chamber priming tank in the suction line or a two-chamber priming tank in the suction and discharge lines, (3) a priming inductor at the inlet of the suction line, or (4) some form of vacuum-producing device.

FOOT VALVES

A foot valve is a form of check valve installed at the bottom, or foot, of a suction line. When the pump stops and the ports of the foot valve close, the liquid cannot drain back to the

suction well if the valve seats tightly. A foot valve does not always seat tightly, and the pump occasionally loses its prime. However, the rate of leakage is generally small, and it is possible to restore the pump to service by filling and starting it promptly. This tendency to malfunction is increased if the liquid contains small particles of foreign matter, such as sand, and foot valves should not be used for such service. Another disadvantage of foot valves is their unusually high frictional loss.

The pump can be filled through a funnel attached to the priming connection or from an overhead tank or any other source of liquid. If a check valve is used on the pump and the discharge line remains full of liquid, a small bypass around the valve permits the liquid in the discharge line to be used for repriming the pump when the foot valve has leaked. Provision must be made for filling all the passageways and for venting out the air.

PRIMING CHAMBERS

The Single-Chamber Tank A single-chamber primer is a tank with a bottom outlet that is level with the pump suction nozzle and directly connected to it. An inlet at the top of the tank connects with the suction line (Figure 1). The size of the tank must be such that the volume contained between the top of the outlet and the bottom of the inlet is approximately three times the volume of the suction pipe. When the pump is shut down, the liquid in the suction line may leak out, but the liquid in the tank below the suction inlet cannot run back to the supply. When the pump is started, it will pump this entrapped liquid out of the priming chamber, creating a vacuum in the tank. The atmospheric pressure on the supply will force the liquid up the suction line into the priming chamber.

Each time the pump is stopped and restarted, a quantity of liquid in the priming tank must be removed to create the required vacuum. Because of possible back siphoning, the liquid volume in the tank is reduced. Unless the priming tank is refilled, it has limited use. Automatic refilling of the priming tank cannot occur if the pump has a discharge check valve unless there is sufficient backflow from the discharge system before the check valve seats.

Because of their size, the use of single-chamber priming tanks is restricted to installations of relatively small pumps, usually to a 12-in (305-mm) suction line (approximately 2000 gpm [450 m³/h]).

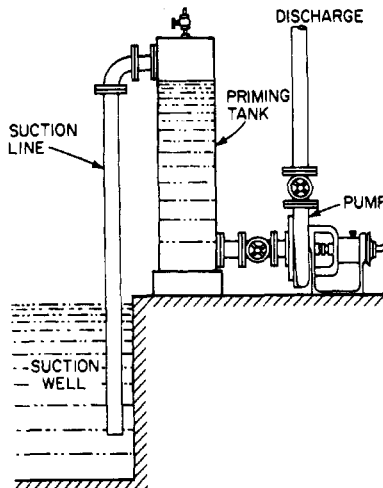


FIGURE 1 Single-chamber priming tank

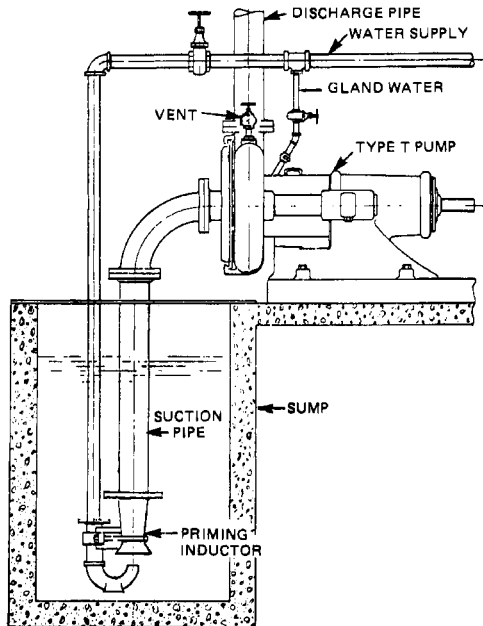


FIGURE 2 Priming inductor (Nagle Pumps)

PRIMING INDUCTORS

If a separate source of liquid of sufficient capacity and pressure is available, it can be used to fill the suction line of the pump to be primed through the use of an inductor, as shown in Figure 2. For water, the pressure must be equal to about 4 lb/in² for each foot (90 kPa for each meter) of head necessary to prime the pump, measured from the lowest liquid level in the sump from which priming must be accomplished to the top of the pump. The amount of liquid necessary depends on the pressure.

Because the inductor is a positive-pressure device, leaks in the suction line or at the pump seal or stuffing box are not critical. The service liquid can be left on or turned off after pumping is initiated. If left on, the head developed by the pump (therefore the flow also) will be increased.

An additional feature of the priming inductor is that operation is possible even when the suction line is covered with sediment. When the pump and suction line are filled, back-flow results, mixing the solids and the liquid.

TYPES OF VACUUM DEVICES

Almost every commercially made vacuum-producing device can be used with systems in which pumps are primed by evacuation of air. Formerly water-jet, steam-jet, or air-jet primers had wide application, but with the increase in the use of electricity as a power source, motor-driven vacuum pumps have become more popular.

Ejectors Priming ejectors work on the jet principle, typically using steam, compressed air, or water as the operating medium. A typical installation for priming with an ejector

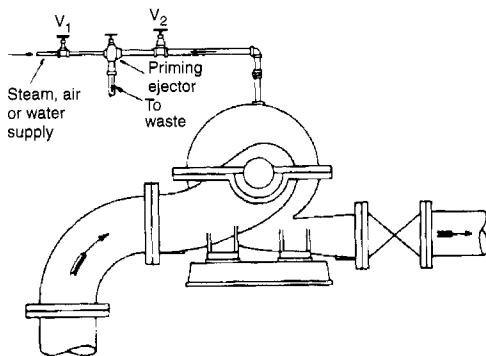


FIGURE 3 Arrangement of priming with an ejector

is shown in Figure 3. Valve V_1 is opened to start the ejector, and then valve V_2 is opened. When all the air has been exhausted from the pump, liquid will be drawn into and discharged from the ejector. When this occurs, the pump is primed and valves V_2 , and V_1 are closed, in that order.

An ejector can be used to prime a number of pumps if it is connected to a header through which the individual pumps are vented through isolating valves.

Dry Vacuum Pumps Dry vacuum pumps, which may be of either the reciprocating or the rotary oil-seal type, cannot accommodate mixtures of air and liquid. When they are used in priming systems, some protective device must be interposed between the centrifugal pump and the dry vacuum pump to prevent liquid from entering the vacuum pump. The dry vacuum pump is used extensively for central priming systems.

Wet Vacuum Pumps Any rotary, rotative, or reciprocating pump that can handle air or a mixture of air and liquid is classified as a wet vacuum pump.

The most common type of wet vacuum pump is a centrifugal displacement type of pump consisting of a round, multiblade rotor revolving freely in an elliptical casing partially filled with liquid. The curved rotor blades project radially from the hub and, with the side shrouds, form a series of pockets and buckets around the periphery.

The rotor revolves at a speed high enough to throw the liquid out from the center by centrifugal force. This forms a solid ring of liquid revolving in the casing at the same speed as the rotor, but following the elliptical shape of the casing. It will be readily seen that this forces the liquid to alternately enter and recede from the buckets as the rotor at high velocity.

If a solid stream of liquid circulates in this pump in place of air or of an air and liquid mixture, the pump will not be damaged but it will require more power. For this reason, in automatic priming systems using this type of vacuum pump, a separating chamber or trap is provided so liquid will not reach the pump.

Liquid needed for sealing a wet vacuum pump can be supplied from a source under pressure, with the shutoff valve operated manually or through a solenoid connected with the motor control. It is, however, preferable to provide an independent sealing liquid supply by mounting the vacuum pump on a base containing a reservoir. This is particularly desirable in locations where freezing may occur, as a solution of antifreeze can be used in the reservoir.

CENTRAL PRIMING SYSTEMS

If there is more than one centrifugal pump to be primed in an installation, one priming device can be made to serve all the pumps. Such an arrangement is called a central priming

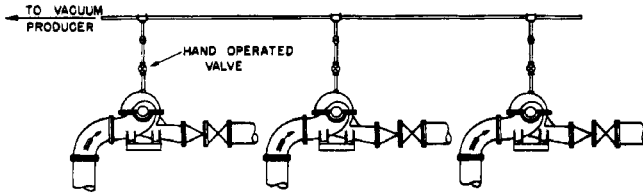


FIGURE 4 Connections for a central priming system

system (Figure 4). If the priming device and the venting of the pumps are automatically controlled, the system is called a central automatic priming system.

Vacuum-Controlled Automatic Priming System A vacuum-controlled automatic priming system consists of a vacuum pump exhausting a tank. The pump is controlled by a vacuum switch and maintains a vacuum in the tank of 2 to 6 in Hg (50 to 150 mm Hg) above the amount needed to prime the pumps with the greatest suction lift. The priming connections on each pump served by the system are connected to the vacuum tank by automatic vent valves and piping (Figure 5). The vacuum tank is provided with a gage glass and a drain. If liquid is detected in the vacuum tank as the result of leakage in a vent valve, the tank can be drained. Automatic vent valves consist of a body containing a float that actuates a valve located in the upper part. The bottom of the body is connected to the space being vented. As air is vented out of the valve, water rises in the body until the float is lifted, and the valve is closed.

A typical vent valve designed basically for vacuum priming systems is illustrated in Figure 6. The valve is provided with auxiliary tapped openings on the lower part of the body for connection to any auxiliary vent points on the system—for instance, when air is to be exhausted simultaneously from the high point of the discharge volute and the high point of the suction passageways. When one or more of these vent points are points of higher pressure, such as the top of the volute of the pump, an orifice is used in the vent line to limit the flow of liquid. Otherwise, a relatively high constant flow of liquid from the discharge back to the suction would take place, causing a constant loss. Where a unit is used more or less constantly, a separate valve should be used for each venting point.

The system shown in Figure 5 is the most commonly used central automatic priming system. It can use either wet or dry motor-driven vacuum pumps. Most central priming systems are provided with two vacuum pumps. The usual practice is to have the control of one vacuum pump switched on at some predetermined vacuum and the control of the second switched on at a slightly lower vacuum.

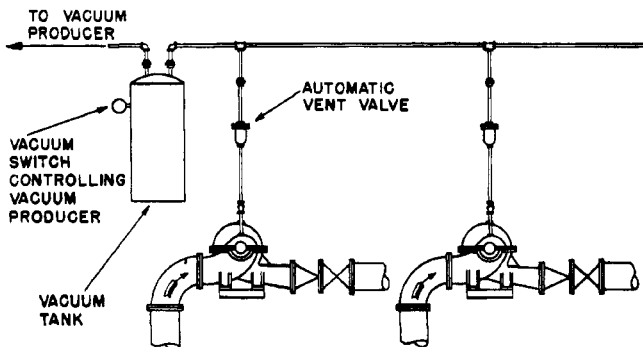


FIGURE 5 Vacuum-controlled central automatic priming

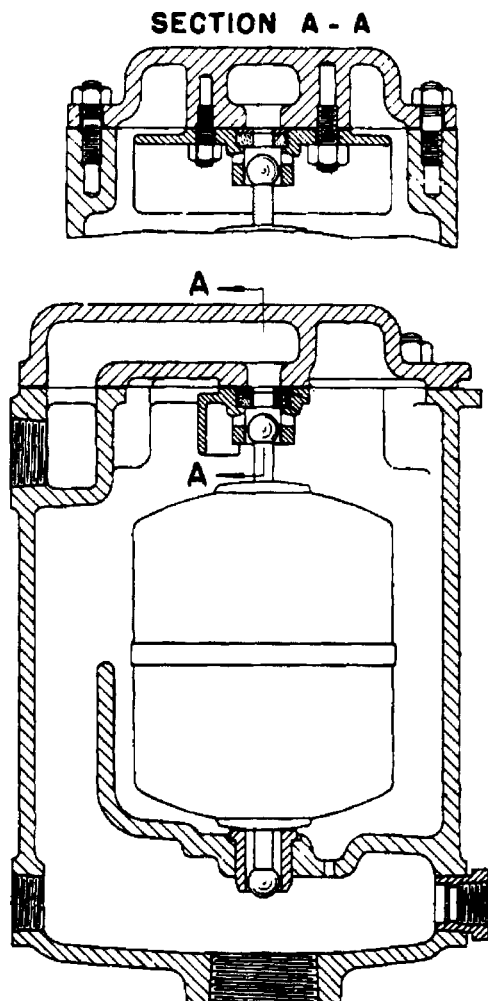


FIGURE 6 Automatic vent valve (Nash Engineering)

Combination dry reciprocating-type vacuum pump and vacuum tank units are commercially available in various sizes. The operation of a typical unit can be explained as follows, referring to Figure 7.

The primer automatically stops and starts itself to maintain a minimum vacuum in its tank at all times, regardless of whether the centrifugal pumps it serves are operating or not.

A vacuum header is run from the primer to the centrifugal pumps, and connection is made to the priming valve mounted on each pump. The vacuum in the primer tank removes the air from the pump and suction line via the vacuum header and priming valve. Liquid rises up the suction line and fills the pump. The liquid cannot enter the vacuum header because the float mechanism in the priming valve closes when the liquid reaches it.

If, for any reason, the liquid level in the pump drops, the priming valve immediately opens and the vacuum restores the liquid level. The pumps are thus kept permanently primed.

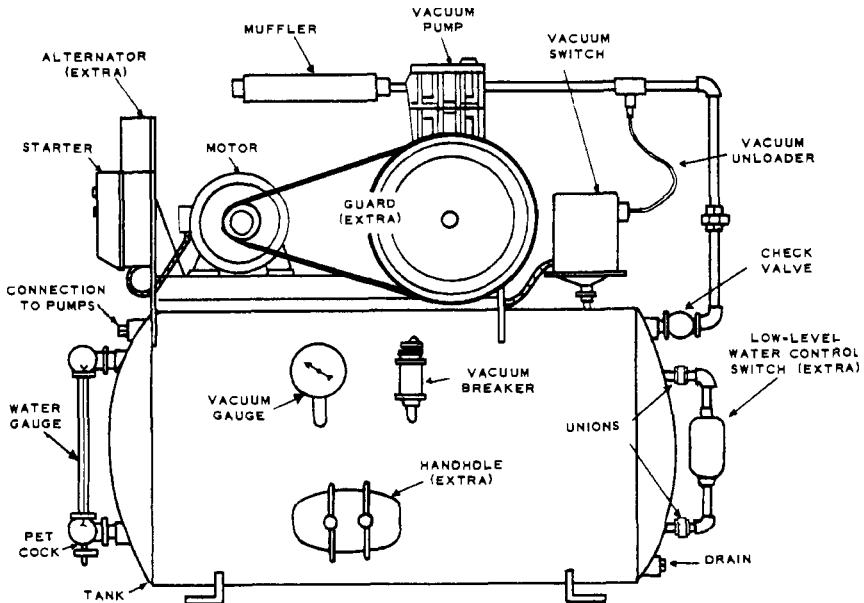


FIGURE 7 Combination dry reciprocating-type vacuum pump and vacuum tank (Apco/Valve and Primer)

Automatic priming systems using ejectors are also feasible, and several such systems are commercially available.

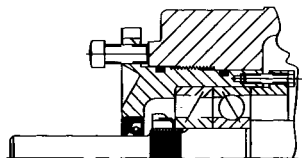
SELF-PRIMING PUMPS

The basic requirement for a self-priming centrifugal pump is that the pumped liquid must be able to entrain air in the form of bubbles so the air will be removed from the suction side of the pump. This air must be allowed to separate from the liquid after the mixture of the two has been discharged by the impeller, and the separated air must be allowed to escape or to be swept out through the pump discharge. Such a self-priming pump therefore requires, on its discharge side, an air-separator, which is a relatively large stilling chamber, or reservoir, either attached to or built into the pump casing. Alternatively, a small air bleed line can be installed from the discharge pipe between the pump and the discharge check valve back to the suction source.

There are two basic variations of the manner in which the liquid from the discharge reservoir makes the pump self-priming: (1) recirculation from the reservoir back to the suction and (2) recirculation within the discharge and the impeller itself.

Recirculation at Discharge This form of priming is distinguished from the preceding method by the fact that the priming liquid is not returned to the suction of the pump but mixes with the air either in the impeller or at its periphery. The principal advantage of this method, therefore, is that it eliminates the complexity of internal valve mechanisms.

One such self-priming pump is illustrated in Figure 8. An open impeller (A) rotates in a volute casing (B), discharging the pumped liquid through passage C into the reservoir (D). When the pump starts, the trapped liquid carries entrained air bubbles from the suction to the discharge chamber. There, the air separates from the liquid and escapes, whereas the liquid in the reservoir returns to the impeller through the recirculation port



OPTIONAL
ANGULAR-CONTACT BEARINGS

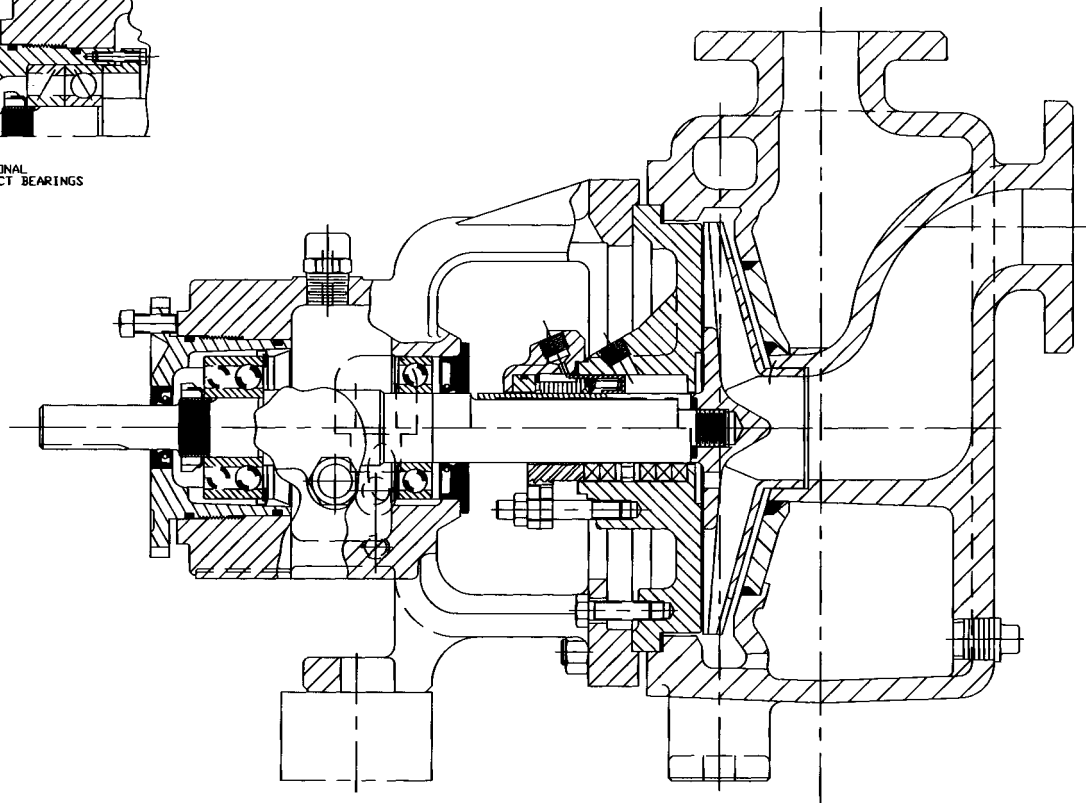


FIGURE 8 Self-priming pump with recirculation at discharge (Flowserve Corporation)

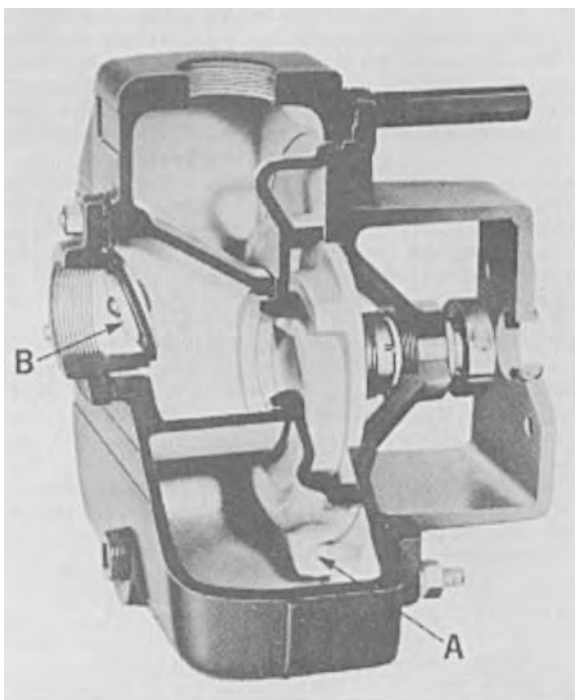


FIGURE 9 Self-priming pump with recirculation at discharge (Peabody Barnes)

(F), reenters the impeller and, after mixing once more with air bubbles, is discharged through passage C. This operation is repeated continuously until all the air in the suction line has been expelled. After the pump is primed, a uniform pressure distribution is established around the impeller, preventing further recirculation. From this moment on, the liquid is discharged into the reservoir bath at C and at F.

Figure 9 illustrates another form of self-priming pump with recirculation at the discharge. In this arrangement, the return of the liquid to the impeller periphery takes place through a communicating passage (A) located at the bottom of the pump casing. After the pump is primed, liquid is delivered into the reservoir at the discharge both through its normal volute discharge and through port A. In the pump illustrated in Figure 9, a check valve at the pump suction (B) acts to prevent the draining of the pump after it has been stopped.

Regenerative Turbine Pumps Because these pumps can handle relatively large amounts of gas, they are inherently self-priming as long as sufficient liquid remains in the pump to seal the clearance between the suction and discharge passages. This condition is usually met by building a trap in the pump suction.

SPECIAL APPLICATIONS

Systems for Sewage Pumps A pump handling sewage or similar liquids containing stringy material can be equipped with automatic priming, but special precautions must be used to prevent carry-over of the liquid into the vacuum-producing device.

One approach is to use a tee on the suction line immediately adjacent to the pump suction nozzle, with a vertical riser mounted on the top outlet of the tee. This riser is blanked at the top, thus forming a small tank. The top of the tank is vented to a vacuum system through a solenoid valve. The solenoid valve in turn is controlled electrically through electrodes located at different levels in the tank. The solenoid valve closes if the liquid reaches the top electrode and opens if the liquid level falls below the level of the lower electrode.

Another solution permits the use of an automatic priming system with a separate motor-driven vacuum pump controlled by a discharge-pressure-actuated pressure switch. An inverted vertical loop is incorporated in the vacuum pump suction line to the pump being primed. This prevents the sewage from being carried over into the vacuum pump because this pump shuts down before the liquid reaches the top of the loop.

Systems for Air-Charged Liquids Some types of liquid have considerable dissolved gas that is liberated when the pump handles a suction lift. In such installations, an air-separating tank (also called a *priming tank* or an *air eliminator*) should be used in the suction line. One type (Figure 10) uses a float-operated vent valve to permit the withdrawal of air or other gas. Another common arrangement uses a float valve mounted on the side of the tank to directly control the starting and stopping of the vacuum pump. Unless the air-separating tank is relatively large and the vacuum pump is not oversized, there is danger of frequent starting and stopping of the vacuum pump in such a system. When sand is present as an impurity in the liquid, the air-separating tank can be made to also function as a sand trap.

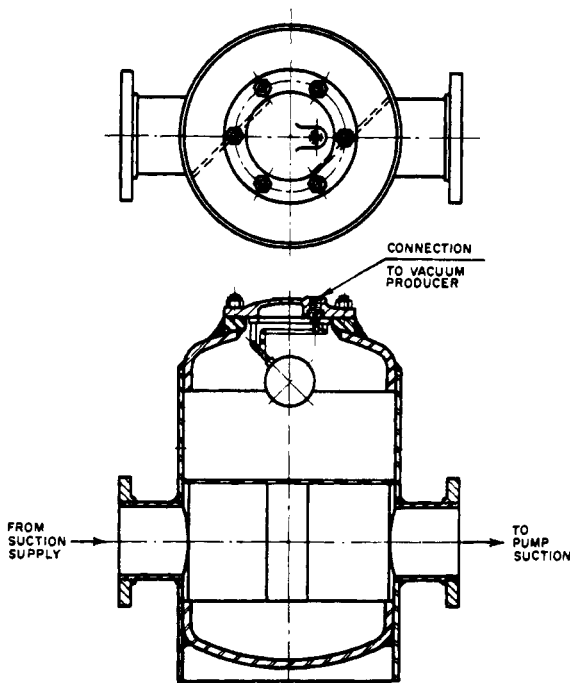


FIGURE 10 Air separating chamber

Systems for Units Driven by Gasoline or Diesel Engines An automatic priming system using motor-driven vacuum pumps can be used for centrifugal pumps driven by diesel engines if a reliable source of electric power is available in the station. An auxiliary vacuum pump driven by a gasoline engine might be desirable for emergency use in case of electric power failure. Alternatively, a direct-connected wet vacuum pump with controls similar to those used in motor-driven automatically primed units is very satisfactory.

The choice of the priming device for a gasoline-engine centrifugal pump depends on the size of the pump, the required frequency of priming, and the portability of the unit. Most portable units are used for relatively low heads and small capacities, for use in pumping out excavations and ditches, for example. Self-priming pumps of various types are most satisfactory for this service and are preferable to regular centrifugal pumps.

It is possible to utilize the vacuum in the intake manifold of a gasoline engine as a means of priming or keeping the pump primed. The rate at which the air can be drawn from the pump in this manner is relatively low, so many of these units use foot valves. They are initially primed by filling the pump manually. Provision must be made to prevent liquid from being drawn over into the manifold.

TIME REQUIRED FOR PRIMING

The time required to prime a pump with a vacuum-producing device depends on: (1) the total volume to be exhausted, (2) the initial and final vacuums, and (3) the capacity of the vacuum-producing device over the range of vacuums that will exist during the priming cycle. The calculations for determining the time necessary to prime a pump are complicated. To permit close approximations, jet primers are usually rated in net capacity for various lifts. It is necessary to divide the volume to be exhausted by the rating to obtain the approximate priming time. Unless such a simplified method is available, the selection of the size of a primer is best left to the vendor of the equipment.

Central automatic priming systems are usually rated for the total volume to be kept primed. The time initially required to prime each unit served by the central system is not usually considered, as the basic function of the system is to keep the pumps primed and in operating condition at all times.

PREVENTION OF UNPRIMED OPERATION

Various controls may be used to prevent the operation of a pump when it is unprimed. These controls depend upon the type of priming system used. For many installations, a form of float switch in a chamber connected with the suction line is used. If the level in the chamber is above the impeller eye of the pump, the float switch control allows the pump to operate. If the liquid falls below a safe level, the float switch acts through the control to stop the pump, to prevent its being started, to sound an alarm, or to light a warning lamp. Such a valve and switch are illustrated in Figure 11.

GENERAL

Because a great number of automatic priming devices and systems are available, care should be taken to use the type or variation best suited to the application. The discussion of priming in this section does not, of course, cover all makes and modifications available for every specific application.

An automatic priming system will often allow units to be operated with excessive air leakage into the suction lines. This is poor practice because it requires the operation of the vacuum producer for greater periods of time than normally necessary.

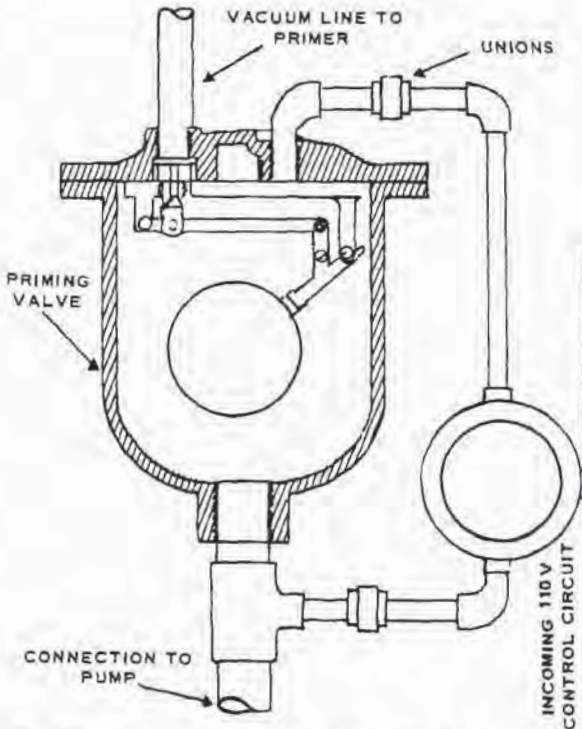


FIGURE 11 Priming valve with liquid level control switch (Apco/Valve and Primer)

FURTHER READING

Apco/Valve and Primer Corp. *Automatic Pump Primers*, Technical Bulletin 721, Schaumburg, IL, 1978.

Hrivnak, S. J. "Why Doesn't Your Self-Priming Pump Work?" *Pumps and Systems Magazine*, June, 1999.

Karassik, I. J., and Carter, R. Chap. 21 in *Centrifugal Pumps: Selection, Operation, and Maintenance*. McGraw-Hill: New York, 1960.

2.2.3 SEALLESS PUMPS

FREDERIC W. BUSE
STEPHEN A. JASKIEWICZ

Sealless pumps are developed to eliminate the liquid leakage to the atmosphere that occurs from pumps that employ packing or mechanical seals. This leakage is usually toxic or dangerous to the environment; sometimes the leakage is valuable. Eighty percent of the applications are for pressures below 200 lb/in² (13.8 bar) and below 250°F (120°C).

Sealless pumps are divided into two categories: magnetic drive pumps and canned motor pumps. The two categories compete against themselves in certain applications, but for the most part they each have their own market niche into which they are applied. Both have encapsulated inner driven mechanisms. On a magnetic drive pump (see Figure 5 in Section 2.2.3.1), the impeller is mounted to an inner magnet carrier. The inner and outer magnetic carriers are sealed by what is called a shell, which contains pump internal pressure. On a canned motor pump (see Figure 1 in Section 2.2.3.2), the impeller is mounted directly to the motor rotor. The atmospheric sealing element between the motor stator and rotor is called a liner or "can." Both magnetic drive and canned motor pumps use product lubricated bearings of compatible design and materials. These bearings are usually cooled and lubricated by the pump liquid.

The following sections describe both magnetic drive and canned motor pumps.

SECTION 2.2.3.1 MAGNETIC DRIVE PUMPS

FREDERIC W. BUSE
GREGORY M. DENAULT

The principle of a magnetic drive pump is the elimination of the seal or packing in an overhung pump (Figure 1) and cutting the shaft in two at the seal location (Figure 2a). On the inner or wet end half of the shaft (Figure 2b), an inner magnet assembly (a) is placed on a shaft (b) supported by product lubricated bearings (c). On the other cut portion of the shaft, an outer magnet assembly (d) is placed on the power shaft (e). Between the inner and outer magnet assemblies, a static seal (f) isolates the shell (g) or diaphragm and the pumped liquid from the atmosphere. The magnetic flux from the outer magnet assembly drives the inner magnet assembly and impeller(s). The outer assembly is mounted directly to either its own bearing assembly in a frame housing (Figure 3) or motor shaft (Figure 4). Figure 5 shows a cross section of a magnetic drive frame mounted pump.

Magnetics A magnetic circuit usually consists of two sets of permanent magnets and inner and outer conducting rings (Figure 6). The conducting rings can be cast iron, ductile iron, carbon steel, or a 400 series stainless steel.

Magnet Materials The first permanent magnets, developed in the 1940s, were made of aluminum-nickel-cobalt (AlNiCo) and used in small chemical pumps. Development of rare earth magnets in the 1980s made it possible to have a small power package that could drive larger pumps. The two types of rare earth materials commonly used are neodymium iron boron (NdFeB) and samarium cobalt (SmCo). The SmCo is four times stronger than AlNiCo. NdFeB, at 70°F (21°C), is 20% stronger than SmCo. The advantage of SmCo is the maximum service temperature of 550°F (288°C), almost twice that of NdFeB, which is 300°F (149°C). Figure 7 shows the strength versus temperature characteristics of the two materials. The cost of SmCo, however, is about twice that of NdFeB.

The magnets have a maximum temperature at which they lose all magnetism in an irreversible process; the magnets do not regain magnetism as they cool. This temperature is the Curie temperature and is shown for each magnet material in Table 1.

Conduction Ring The amount of transmittal torque depends on the overall gap (Figure 8) between the poles of the magnets and the thickness of the conducting ring (Figure 9). If

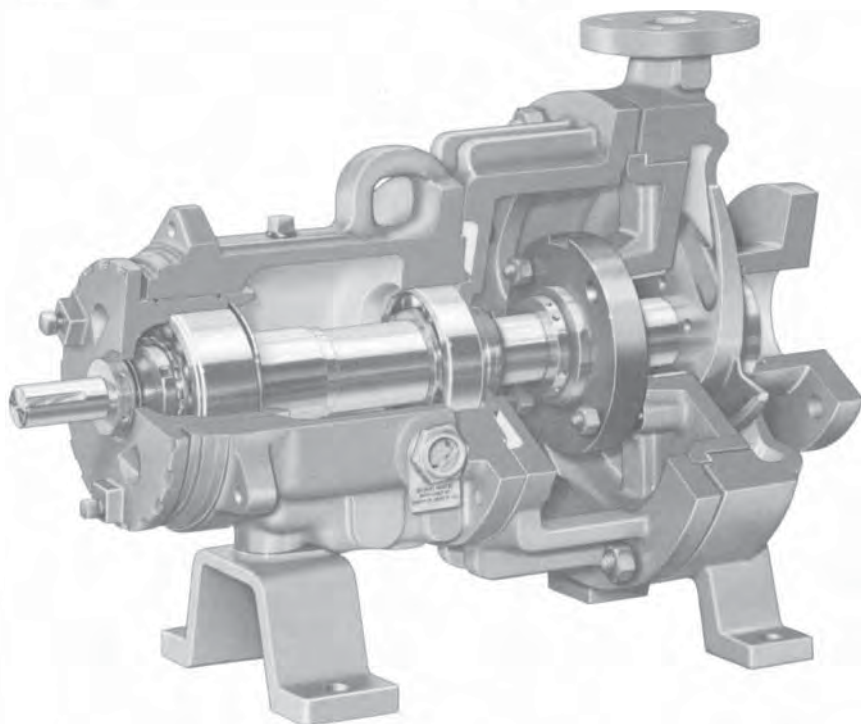


FIGURE 1 Pump with conventional bearing housing and mechanical seal (Flowserve Corporation)

the thickness of the conducting ring is too small, it will become saturated with flux and the torque capability will be reduced.

Gap The “overall gap” (Figure 8) is made up of the air gap, containment shell thickness, liquid gap, and encapsulation. Gap dimensions are based on the pressure requirement for the shell, the number of magnets (single or dual), and the material of the shell (metallic or nonmetallic) in the gap.

EXAMPLE Dimensions shown as in (mm)

Mean diameter (on radius dimensions)	4.6 (117)	6.0 (152.4)
Air gap minimum	.030 (0.762)	.045 (1.143)
Containment shell thickness: Hastelloy C	.040 (1.016)	.060 (1.524)
Polymer	.120 (3.048)	.150 (3.810)
Liquid gap	.035 (0.889)	.035 (0.889)
Encapsulation	.030 (0.762)	.030 (0.762)
Overall gap (basis Hastelloy C)	.135 (3.429)	.170 (4.318)
Overall gap (basis polymer)	.215 (5.461)	.260 (6.604)

In the previous example, the same overall gap was maintained for polymer and Hastelloy C shells. This allows for interchangeability of the magnetic assemblies independent of shell material.

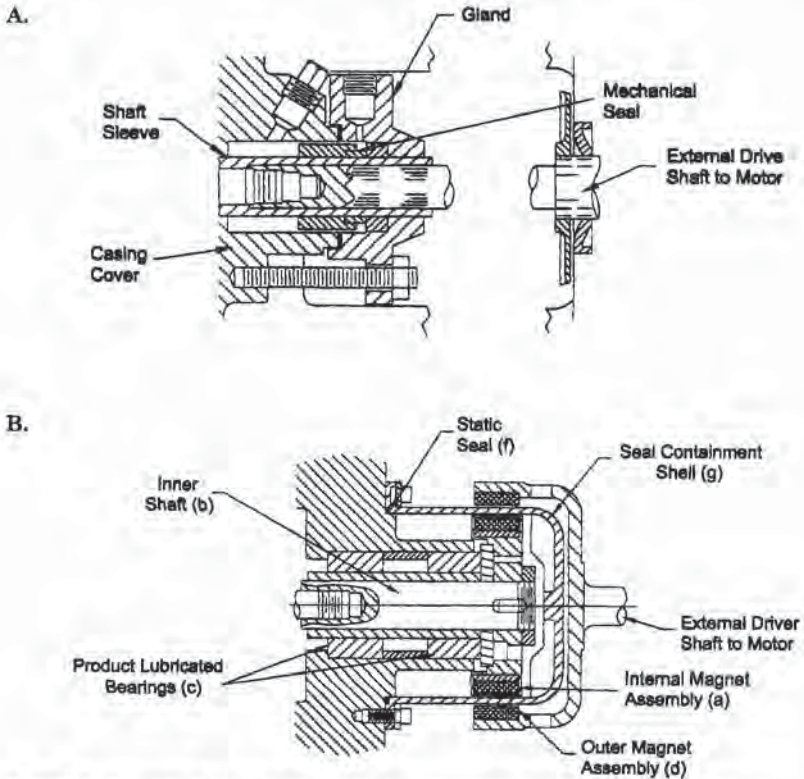


FIGURE 2 Comparison of typical mechanical seal arrangement and sealless pump drive configuration: (a) arrangement for typical mechanical through casing cover, (b) sealless drive configuration

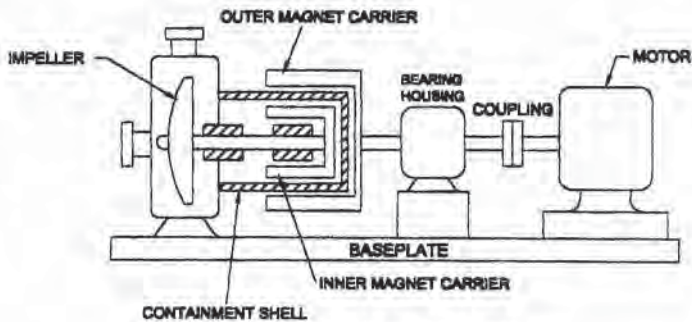


FIGURE 3 Frame-mounted magnet drive arrangement

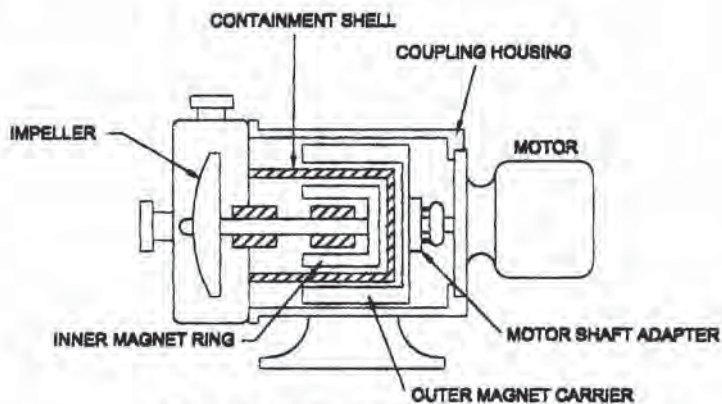


FIGURE 4 Close-coupled magnet drive arrangement

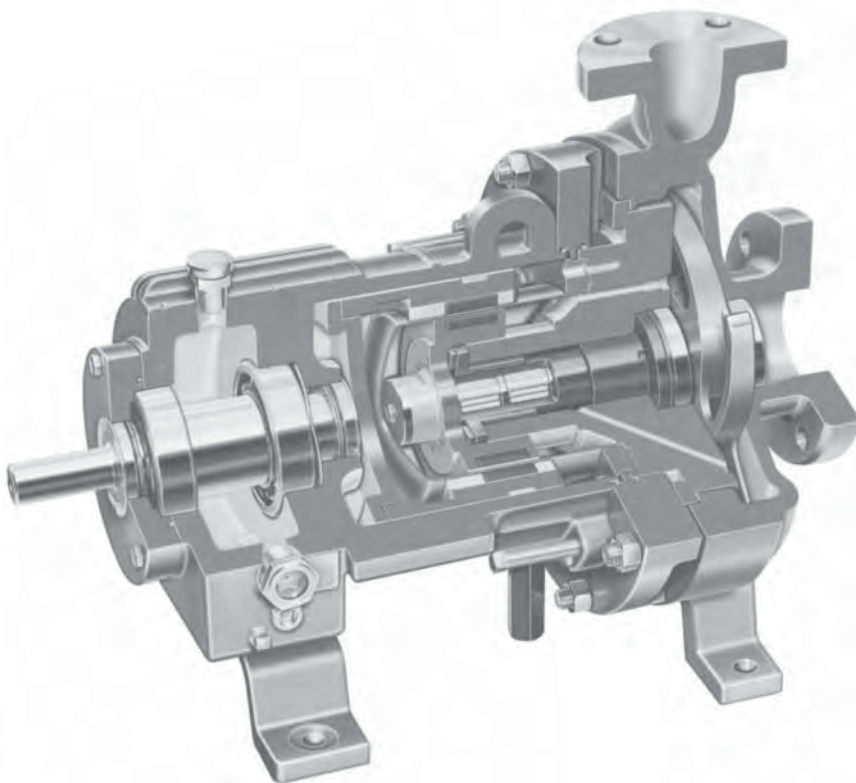


FIGURE 5 Cross-section of a frame-mounted magnet-driven pump (Flowserve Corporation)

FLUX

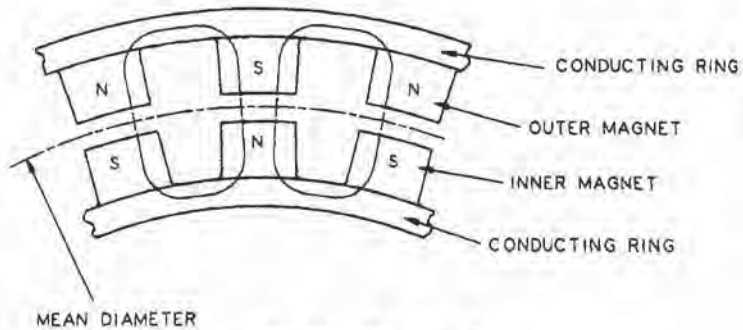


FIGURE 6 Magnetic circuit

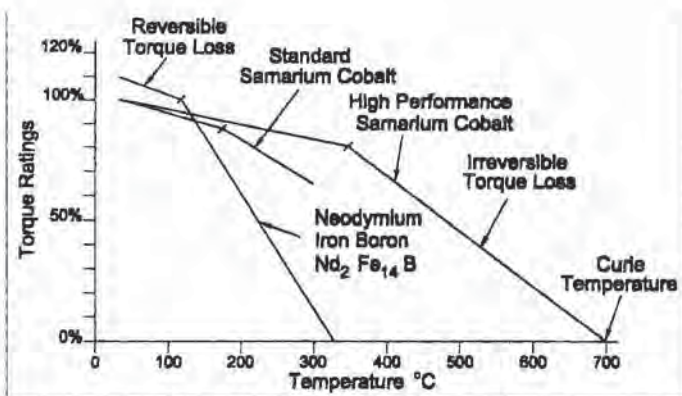


FIGURE 7 Magnet strength versus temperature

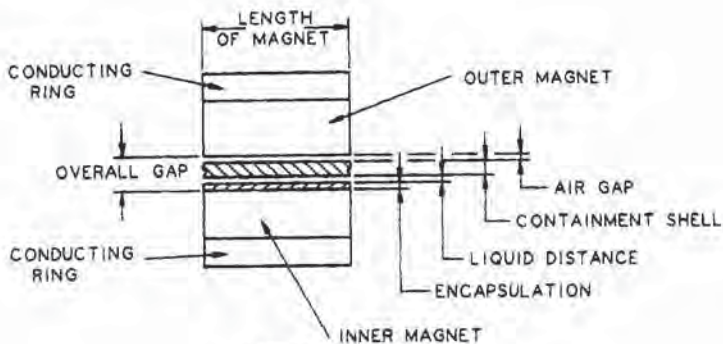


FIGURE 8 Magnet configuration

TABLE 1 Operating and Curie temperatures for magnet materials

Type of Magnet Material	Operating Temperature (°F/°C)	Curie Temperature (°F/°C)
AlNiCo	660/305	800/425
NdFeB	250–300/120–137 (Depending on grade)	600/310
SmCo	500–660/260–350 (Depending on grade)	1300/700

Transmittal Torque The torque transmitted by the magnets depends on the following:

- Flux density of the magnets, B_g
- Operating temperature of the magnets (which will change B_g)
- Length of the magnet poles, L
- Number of magnets per ring, M
- Mean ratios between the ID of the outer assembly magnets and OD of the inner assembly magnets, r
- Overall gap between the ID and OD of the magnets in the assembly, g
- A constant, K , which changes as a function of the specific design

Torque is determined from the following relationship, with appropriate units:

$$T = \frac{K \times B_g^2 \times L \times M \times r}{g}$$

For a given design type and configuration, the torque varies inversely as the square of the overall gap. Depending on costs and specific design construction, an assembly ring of magnets can have either one continuous length of magnets of a series of 1, 2, 3, or more rows of individual magnets.

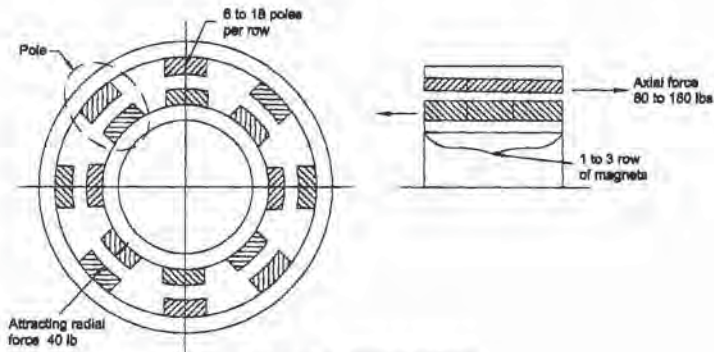


FIGURE 9 Rows of magnets

TORQUE CAPABILITY

The ultimate torque is the static "breakaway torque." To determine this, the magnet carriers are assembled so the inner carrier is locked in position. Then a torque is applied by bar and weights or by a torque wrench to the outer carrier. The torque value at which the two carrier assemblies break loose from each other radially—or "decouple"—is called the "breakaway torque." The designer has to account for the driver start-up acceleration time, start-up torque, and an appropriate safety factor to apply to the breakaway torque to determine the allowable applied torque.

Table 2 gives examples of the torque capability of assemblies composed of NdFeB blocks of magnet 0.75 in wide $\times$ 0.38 in high $\times$ 1.125 in long (19 mm $\times$ 9.65 mm $\times$ 28.6 mm) with a 0.180-in (4.6-mm)-thick conducting ring.

Basic Dimensioning of Magnet Blocks The magnet blocks can be made too long or too short, resulting in handling, magnet molding, or flux density problems. The ratio of the dimensions for magnet blocks for successful designs is as follows:

- Block width is two to three times the thickness.
- Block length is three to five times the thickness.
- Doubling the block thickness will increase the strength by approximately 20%.

For reference: 1.0 in³ (16.387 cm³) of NdFeB per assembly at a mean diameter of 4.0 in (101.6 mm) with a 0.25 in (6.35 mm) overall gap produces approximately 7 ft $\cdot$ lb (9.5 N $\cdot$ m) of torque. A 3% reduction in flux density is equal to a 6% reduction in torque.

Characteristics of magnet material:

Density: 0.273 lb/in³ (7.56 g/cm³)

Tensile: 12×10^3 lb/in² (844 kg/cm²)

Compression: 110×10^3 lb/in² (7.7×10^3 kg/cm²)

Flex stress: 36×10^3 lb/in² (2.53×10^3 kg/cm²)

Coefficients of thermal expansion are shown in Table 3.

Radial and Axial Magnet Forces The inner and outer carriers have to be restrained radially by bearings from contacting each other. In the example of "Torque Capability," a single row of 18 magnets with a 6 in (152 mm) mean diameter, the radial force with magnets concentric is 40 lb (18 kgf). When the magnets are offset by 0.005 in (0.127 mm), the radial force is 55 lb (25 kgf) (Figure 9). When the magnets are against each other (no gap), the radial force is 80 lb (36 kgf).

In the previous example, it takes 60 lb (27 kgf) in an axial direction to separate a single row of concentric magnets and 180 lb (82 kgf) for three rows. It is strongly advisable that provisions be made for personnel to address these loads during assembly and disassembly of the carriers.

Encapsulation of Inner Carrier Magnetics Encapsulation can be accomplished with either metallic or polymer materials. The encapsulation of the inner magnet and conducting ring is probably the most expensive and extensive process in a magnetic drive sealless

TABLE 2 Torque capability of magnet assemblies

Mean Diameter in (mm)	Magnets/Row	Overall Gap in (mm)	Breakaway Torque ft-lb (N $\cdot$ m)
4.3 (109)	12	0.240 (6.10)	30 (40)
6.0 (152)	18	0.260 (6.60)	60 (80)

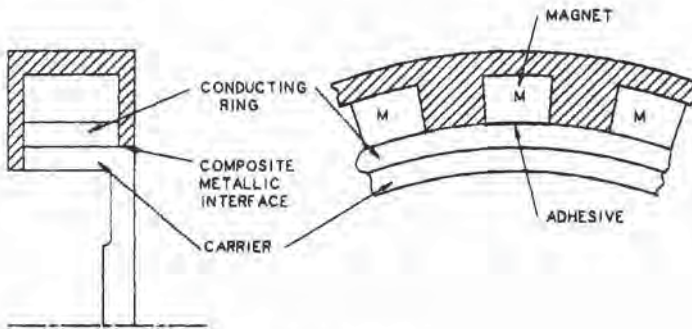
TABLE 3 Coefficients of thermal expansion (in/in/°F × 10⁻⁶)/(cm/cm/°C × 10⁻⁶)

Magnet material	Parallel to Axis	Perpendicular to Axis
NdFeB	5 (9)	9 (16.2)
SmCo	20 (36)	16 (28.8)

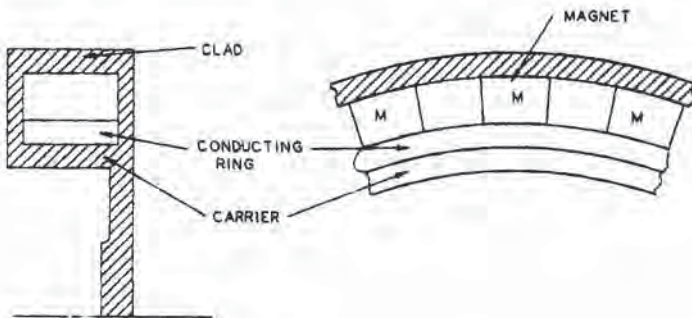
pump. After encapsulation, the carrier should be nondestructively tested to confirm 100% effectiveness.

The pros and cons of polymer encapsulation (Figure 10a) are as follows:

1. Limited to 250 to 300°F (120 to 150°C)
2. The overall gap is increased because of the required thickness of the polymer.
3. Magnets must be restrained mechanically on the conducting ring by adhesives or high-strength polymers around the magnets.
4. Polymers like PFA/PTFE or PEEK are more corrosive-resistant than most metals.
5. Production polymer construction is much less expensive than metallic construction.
6. Polymer tooling is expensive.



(A) NONMETALLIC COMPOSITE



(B) METALLIC

FIGURE 10 Encapsulation of magnets using (A) nonmetallic composition and (B) metallic clad

The pros and cons of metallic encapsulation (Figure 10b) are as follows:

1. Temperature rating can be 550°F (288°C).
2. Thickness of the encapsulation material over the magnets can be as little as 0.030 in (0.76 mm).
3. Welding of the components can be conventional, electron beam, or laser. However, care must be taken with conventional welding to prevent the arc from jumping toward the magnet flux.
4. If castings are used for the inner carrier, porosity can be a problem.
5. Gassing of polymers from welding heat coming out of the seams can be a problem.
6. Adhesives are not required to keep the magnets in place at 3600 rpm with metallic encapsulation; the outer shield performs this function.

Encapsulation of Outer Carrier Magnets The magnets for the outer carriers do not have to be encapsulated (Figure 11). However, material like NdFeB has an affinity for water absorption that results in rusting and swelling. The magnets can then break loose and move in position relative to one another. It is highly recommended that they be encapsulated with an epoxy or metal sheathing for atmospheric protection and handling.

Construction The outer carrier can be a casting or fabrication. The carrier is attached to the power end shaft in the bearing housing (Figure 3) or directly to a motor shaft, which is then called *close coupled construction*. When attached to the bearing housing shaft, there is very little axial or radial load applied to the bearings. This lightly loaded condition can result in internal skidding of the rolling element bearings within their races, resulting in premature bearing failure. Therefore, it is best to preload the bearings with a spring to prevent skidding. This can be accomplished outside of the bearing by a spring-loading feature (Figure 11).

Containment Shell The containment shell shape and thickness depends on working pressure, material, and temperature. The shell thickness is usually uniform. To keep the length of the shell to a minimum, a square-ended shell can be used. However, unless the end of the shell is made extra thick, it would have a relatively low pressure capability. Therefore, an end shape which is elliptical or spherical is used to obtain higher pressure capability (Figure 12).

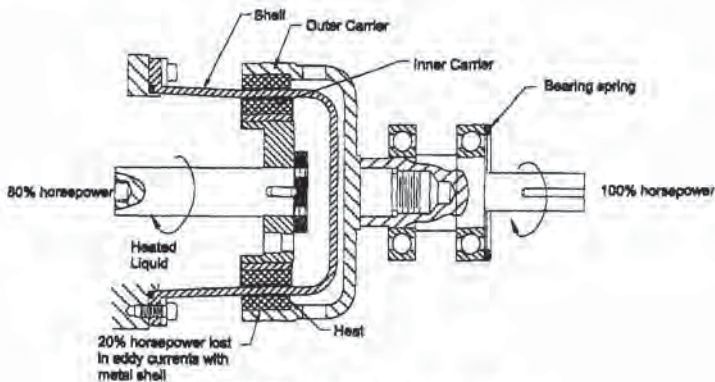
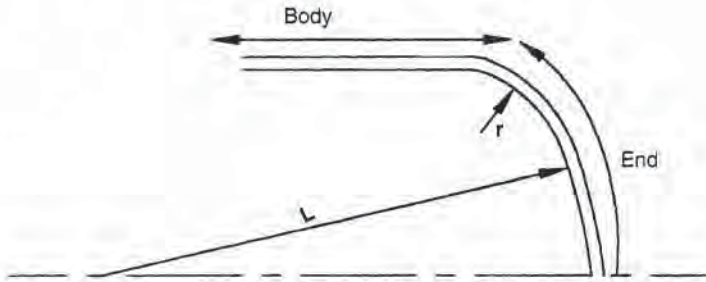


FIGURE 11 Inner and outer carrier

FIGURE 12 Shell ratio L/r for pressure

Examples of how allowable shell pressure varies with the shape of the end plate are shown in Table 4.

The classes of materials used for containment shells (Figure 11) include metals, polymers, and ceramics. The characteristics of containment shells of various materials are shown in Table 5. The main advantage of polymer or ceramic shells is that there are no eddy current losses. Therefore, cooling of the magnets is not required.

Eddy Currents Metallic or metallic-lined shells will produce eddy currents. Depending on the thickness of the shell, the eddy current losses (P_L) can amount to as much as 20% of the total power.

$$P_L = \frac{K \times T \times L \times N^2 B_g^2 D^3 M}{R}$$

where K = a constant, depending on the design

T = thickness of the shell

L = length of magnets (times the core of magnets) N = speed, rpm

B_g = flux density of the magnets

D = mean diameter

M = number of sets of magnets

R = electrical resistivity, microhms per cm^3 (electrical resistivity for various shell materials is given in Table 6)

The effect of material selection on heat buildup is shown in Table 7. The tabulation shows the temperature rise of the air space inside a shell with only the outer carrier spinning

TABLE 4 Pressure capabilities of shells with various end shapes for the same thickness and material

Shape	L/r	Allowable pressure—lb/in ² (kPa)	Ratio of allowable pressure to that for a flat plate
Flat	533	9 (62)	1.0
Flat	267	23 (158)	2.55
Elliptical	28	113 (779)	12.6
Elliptical	8	209 (1440)	23.2
Spherical	1	815 (5620)	90.5

TABLE 5 Characteristics of containment shells of various materials

Material	Metal	Reinforced Polymer	Ceramic	Metal with PTFE
Construction	Welded or hydroformed	Injection molded	Set and fired	Spray coated
Temperature limit—°F (°C)	−300 to + 750 (−150 to +400)	−40 to +250 (−4 to + 120)	32 to 2000+ (0 to 1100+)	32 to 350 (0 to 175)
Thickness—in (mm)	0.030 to 0.040 (0.76 to 1.0)	0.170 (4.3)	0.250 to 0.380 (6.35 to 9.6)	0.050 (1.27)
Heat conductivity*	Hastelloy C = 71 AISI 316 = 7.5	3	6	Higher than metal alone
Creep	None	None	None	Some
Thermal shock	1000+ (537+)	375 (190)	500 (260)	—
Eddy currents	316 is twice Hastelloy C	None	None	Same as metal

*Approximate thermal conductivity in Btu/h/ft²/°F. Multiply by 0.488 to get calories/h/cm²/°C

TABLE 6 Electrical resistivity of various shell materials

Shell Material	Electrical Resistivity (<i>R</i>), microhms per cm ^a
Hastelloy C	130
AISI 316L	74
Inconel 625	129
Nimonic 90	115
Titanium	53
K-Monel	58
Alloy 20	75

TABLE 7 Effect of material selection on heat build-up

Material of Shell	Thickness in (mm)	Time min	kW	°F (°C)	°F/min (°C/min)
AISI 316	0.43 (10.9)	Start	0.40	80 (27)	—
AISI 316	0.43 (10.9)	5.0	0.30	500 (260)	100 (38)
Hastelloy C	0.43 (10.9)	0.5	0.24	170 (77)	340 (171)
Hastelloy C	0.43 (10.9)	5.0	0.21	370 (188)	74 (23)

around a metallic shell (no inner carrier in place) at 3550 rpm. The carrier has 12 magnets, 1.25 in (3.175 cm) long, with a mean diameter of 4.25 in (11.43 cm).

This illustrates the difference between AISI 316 and Hastelloy C shells at 3550 rpm. At 1750 rpm, the power loss for AISI 316 would be one-fourth that at 3550 rpm, which may not result in excess power consumption.

There are also axially laminated metal shells available that substantially reduce eddy current (I²R) losses. These laminations work on the principle that when a shell length is cut in half, the eddy current losses are reduced to one-fourth their original value. If the one-half shell length is cut in half again, the resulting pieces have eddy current losses one-sixteenth that of the original shell. A shell design made of segments sealed together to make a full-length shell will thus reduce total eddy current losses.

Dual Containment As a precaution against leakage due to a breach in the primary containment shell, a dual or double-containment arrangement can be used. This double layering of shells (Figure 13) usually consists of a combination of a nonmetallic and metallic shell. The pressure rating of the secondary shell is equal to that of the primary shell. It is designed to operate at least 48 to 120 h after a breach in the primary shell occurs. A pressure monitor is inserted in the flange of the secondary shell to detect pressure buildup from primary shell leakage.

Bearings The internal shaft system of the pump is supported by one or more bearings. Some designs use a rotating shaft; others use a mandrel on which the bearings rotate. The bearing, which consists of a journal and bushing, is made of various materials, depending on the loads and pumpage (used for product lubrication). The bearing loads are from the weight of the components and hydraulic forces from the impeller and inner carrier. The impeller forces are both radial and axial.

Most magnetic drive sealless pumps are single-stage volute pumps that have the same radial bearing loads as comparable conventionally sealed pumps. The main difference with the sealless pump is that there is almost no overhang from the impeller to the first bearing. The load on the bearing, therefore, is almost equal to that of the impeller. In a conventionally sealed pump, the load on the radial bearing is almost twice that of the impeller. This can be seen by comparing Figures 1 and 5.

The axial load will depend on whether the impeller is enclosed or semiopen (Figure 14). Enclosed impellers usually have horizontal front rings and may also have a back ring or pump-out vanes. Semiopen impellers have no rings, but they usually employ scallops in the shroud to reduce the effective pressure area (Figure 15). Axial thrust force is further reduced by employing pump-out vanes (POV) or pump-out slots (POS) on the back shroud of the impeller to reduce the amount of pressure on the impeller back shroud.

The enclosed impeller can have less radial and axial load than a semiopen impeller. This is usually accomplished by employing back rings (Figure 14b). To ensure positive

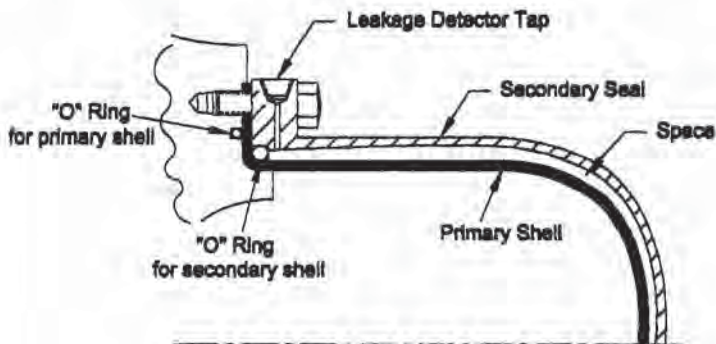


FIGURE 13 Dual containment

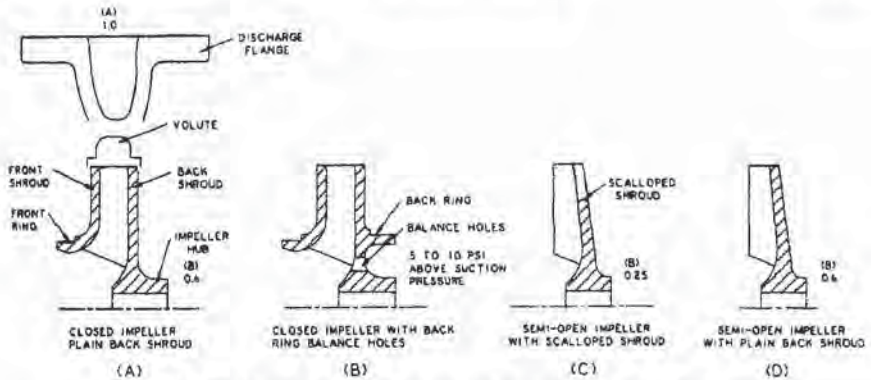
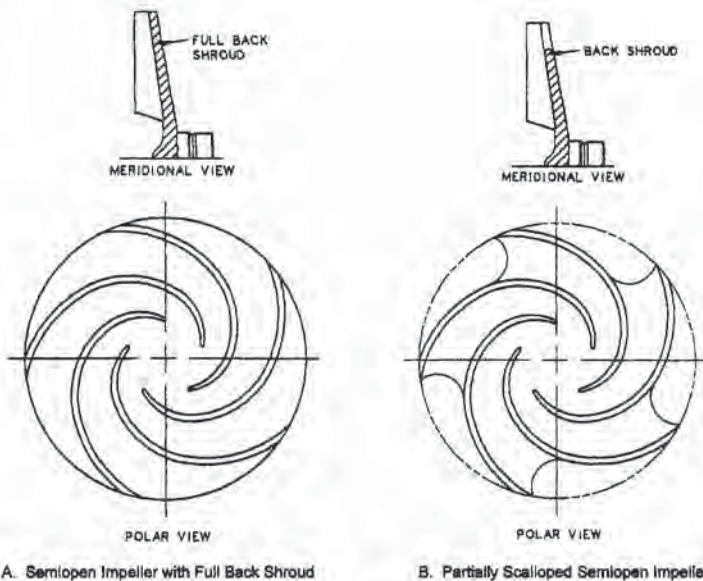


FIGURE 14A through D Various enclosed and semiopen impeller configurations

pumping in the lubrication flow path in magnetic drive pumps, POV-POS on an enclosed or semiopen impeller are recommended.

Liquids pumped in chemical or petroleum plants may have low viscosity, low specific gravity, or low specific heat. These characteristics can result in boundary lubrication rather than hydrodynamic lubrication of the product lubricated bearings. Therefore, bearings are selected using pressure-velocity (PV) values. Some limiting PV values for various bearing material combinations are shown in the following section on bearing materials.

Bearing Materials Table 8 shows PV values for various bearing journal and thrust face materials. When loads exceed a PV value of 150,000, the bearing should have an alignment compensator built in to correct for inaccuracies in perpendicularity, concentricity, and parallelism of parts.



A. Semiopen Impeller with Full Back Shroud

B. Partially Scalloped Semiopen Impeller

FIGURE 15A and B Semiopen impellers with a full back shroud and with a partially scalloped back shroud

TABLE 8 PV values for various bearing journal and thrust face materials

Bushing Materials	Journal/Thrust Face PV ($\times 10^3$)*
Carbon graphite vs. AISI 316	150
Carbon graphite vs. chrome oxide hardened coating	250
Silicon carbide vs. hardened coating	250
Silicon carbide vs. carbon graphite	300
Silicon carbide vs. silicon carbide	500
PEEK—carbon-filled vs. hardened coating	150
Polyimide-carbon filled vs. hard coating	150

P = Net load/projected area (minus area for slots), psi

V = Velocity at the shaft diameter or mean diameter of the thrust face in ft/min

*Note: Multiplying the PV values in the table by 2.1 will give values equal to P in kPa and V in m/min.

When designing the bearing, thermal expansion and heat conductivity properties must be considered. Some designs have a temperature range of -300 to $+750^\circ\text{F}$ (-150 to $+400^\circ\text{C}$). Table 9 lists bearing material characteristics to facilitate selection of the proper material for the intended application.

Particles For the softer bearing materials, the maximum particle size should be no more than 10% of the diametral clearance for a bearing with no grooves and no more than 20% for a bearing with grooves. Polymers and graphites are much softer than silicon carbides or hard coatings, and they are not recommended for liquids with hard particles. Silicon carbide versus silicon carbide will grind up most particles. For particle concentrations of 50 parts per million (ppm) or less, particle hardness is generally not a factor in bearing performance.

Strainers of 100 mesh are recommended to reduce the amount and size of particles going through the flow path. These strainers are installed for internal or external injection. Remember: Particles that will pass through a 100 mesh screen may be as large as 0.006 in (0.15 mm), whereas nominal diametral bearing clearances of 0.002 in (0.05 mm) are common.

TABLE 9 Bearing material characteristics

Material	Thermal Expansion (note 1)	Heat Conductivity (note 2)	Hardness (note 3)
AISI 316	9.6	7.5	160 BHN
Hastelloy C	6.7	71	170 BHN
Carbon graphite	2.6	5	95 Vickers
Silicon carbide— self-sintered	2.2	85	2400 Vickers
Silicon carbide with carbon	2.0	75	2400 Vickers
PEEK—Carbon-filled	8.0	6	
Polyimide—Carbon-filled	5.0		100 to 150 BHN
Aluminum oxide			1800 Vickers
Chrome oxide			1800 Vickers

Note 1: $\text{in/in}/^\circ\text{F} \times 10^{-6}$ (multiplied by 1.411 to get $\text{cm/cm}/^\circ\text{C}$)

Note 2: $\text{BTU/h}/\text{ft}^2/^\circ\text{F}$ (multiply by 0.488 to get $\text{calories/h}/\text{cm}^2/^\circ\text{C}$)

Note 3: $R_C \times 25 = \text{Vickers}$; $\text{BHN} \times 10 = R_C$

Note also that the bearing material combination of silicon carbide against silicon carbide or against carbon is electrically conductive.

Running Dry Most sealless pump failures occur because the pump system is not monitored and the pump is allowed to run dry. When this happens, the bearings will run dry. When the bearings run dry, hard, brittle bearing materials such as self-sintered silicon carbide will fail within minutes. Graphite silicon carbide may run dry for as many as 10 to 20 min with no damage. The polymers, which are poor heat conductors, expand inwardly and seize against the mating surface. The graphites will also move inwardly, but they tend to wear rather than seize, which results in excessive clearances when the pump is stopped and cooled. One optional design uses Teflon strips that expand inwardly when the unit runs dry so the shaft runs on the Teflon rather than on the silicon carbide, thus helping to avoid bearing failure.

Another option is to employ an external circulating tank, with no external running parts, that will provide the bearings with external lubricating liquid should dry operation occur. This type of external tank system has allowed pumps to operate for two hours or more without incurring damage to the bearings.

Flow Path The amount and direction of flow for cooling of the magnets and lubrication of the bearings is critical to the operation of a sealless pump (Figure 16). It is preferable for the liquid to lubricate the bearings before being heated by the magnets. This reduces the possibility of vaporization of the liquid occurring at the thrust-bearing faces. The amount of liquid circulated through the system is usually between 1 and 8 gpm (4 and 30 L/min). It is usually channeled to the front and back bearings, the thrust bearing face, across the magnets, and to the impeller hub. Some manufacturers have computer programs that calculate the flow, pressure, and temperature of the cooling/lubricating flow at various critical locations along the flow path. These programs can also account for the effects of the liquid specific gravity, specific heat, and viscosity. The local pressure and temperature is used to determine the vapor pressure at that point to ensure that the liquid is not flashing. The programs can also calculate axial thrust and determine if the lubrication

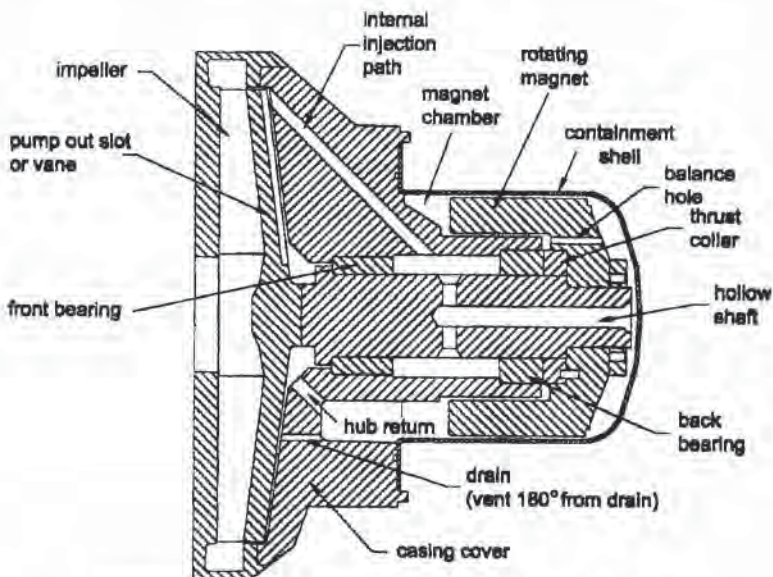


FIGURE 16 Magnetic sealless pump components for the internal flow system.

at the thrust bearing face is hydrodynamic or boundary. This is done at various flow rates, impeller diameters, and pump speeds.

The temperature rise of the liquid as it travels through the cooling-lubricating flow path depends to a great extent on the liquid's characteristics, such as specific gravity, specific heat, vapor pressure, and viscosity. With a nonmetallic shell on ambient water service, a typical temperature rise might be 1 to 2°F (0.5 to 1°C). For a metallic shell, with its much higher eddy current losses, the temperature rise could be as much as 8 to 12°F (4 to 7°C).

Performance The head capacity curve of a typical two-pole speed sealless pump matches that of a conventionally sealed pump; however, the overall efficiency is lower. With a non-metallic shell, the efficiency may be only about two points less at the best efficiency point flow rate, but as much as six points less at one-half the best efficiency point flow rate. When a metallic shell is used, the efficiency at best efficiency point flow rate may be as much as 8 to 12 points lower than a comparable pump with a mechanical seal (Figure 17).

APPLICATION ADVANTAGES OF SEALLESS PUMPS:

- No leakage to the environment
- No loss of valuable liquids
- Lower noise levels
- High suction pressure does not affect the axial thrust
- Can handle liquids from 0 to 4 toxicity rating
- Because of no leakage, there is much less chance of a fire
- Easier to obtain construction permits and permits for continued operation
- Less external piping required

APPLICATIONS THAT SHOULD BE REVIEWED BEFORE SEALLESS PUMPS ARE APPLIED:

- Dirty liquids
- High temperature

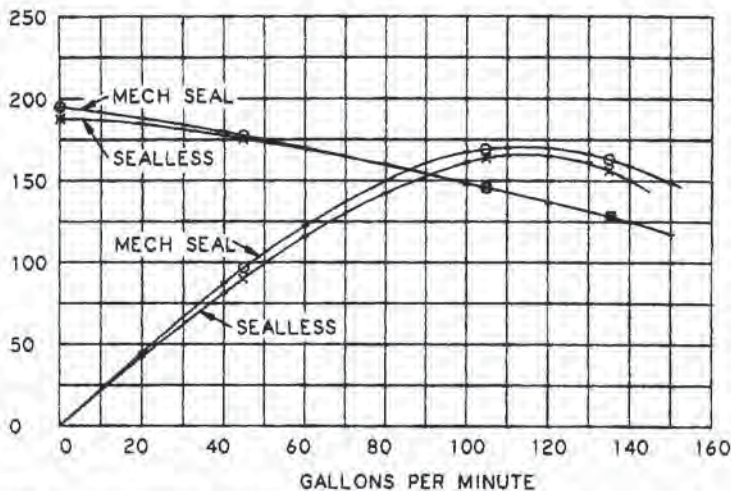


FIGURE 17 Typical performance for a 1.5 x 1.0 x 6 pump at 3550 rpm, comparing characteristics for sealless versus mechanical seal construction

- Liquids that solidify
- Viscous liquids above 200 centipoise
- Oversize drivers that can cause decoupling during acceleration
- Cavitation of liquid in the impeller eye that can result in excess thrust
- Excessive entrained gas

FURTHER READING

American National Standard for Sealless Centrifugal Pumps, ANSI/HI 5.1-5.6-2000, Hydraulic Institute, Parsippany, NJ www.pumps.org.

Hydraulic Institute ANSI/HI 2000 Edition Pumps Standards, Hydraulic Institute, Parsippany, NJ www.pumps.org.

Sealless Pumps for Petroleum, Heavy Duty Chemical, and Gas Industry Services, API Standard 685, 2000, The American Petroleum Institute, 1220 L Street, Northwest, Washington, D.C. www.api.org.

Specification for Sealless Horizontal End Suction Centrifugal Pumps for Chemical Process, ANSI/ASME B73.3M-1997, The American Society of Mechanical Engineers, 345 East 7th Street, New York, NY www.ASME.org.

Eierman, R. "A User's View of Sealless Pumps—Their Economics, Reliability, and the Environment." *Proceedings of the Seventh International Pump Users Symposium*. Texas A&M University, College Station, TX, March 1990, pp. 127–133.

Hernandez, T. "A User's Engineering Review of Sealless Pump Design Limitations and Features." *Proceedings of the Eighth International Pump Users Symposium*. Texas A&M University, College Station, TX, March 1991, pp. 129–145.

Littlefield, D. "Sealless Centrifugal Pumps." *Proceedings of the Eleventh International Pump Users Symposium*. Texas A&M University, College Station, TX, March 1994, pp. 115–119.

Guinzburg, A., and Buse, F. "Computer Simulation of the Flowpath in Magnetic Seal-less Pumps." *Proceedings of the Fifteenth International Pump Users Symposium*. Texas A&M University, College Station, TX, March 1998, pp. 1–9.

SECTION 2.2.3.2 CANNED MOTOR PUMPS

STEPHEN A. JASKIEWICZ

A canned motor pump (CMP) is a combination pump and induction motor built together into a single hermetically sealed unit (see Figure 1). For single stage, centrifugal, canned motor pumps (see Figure 2) the impeller (A) is normally of the closed type, and is mounted on one end of the rotor shaft, which extends from the motor section into the pump casing. The rotor (B) is submerged in the fluid being pumped and is, therefore, "canned" to isolate the motor components from contact with the process fluid. The stator (C) is also "canned" to isolate the motor windings from the fluid being pumped. Bearings (D) are submerged in the system fluid and are continually lubricated. The outer stator band assembly (G) in conjunction with a sealed electrical connection (F) provides a positive secondary containment in the event of the primary containment, stator liner (E), failure.

The canned motor pump has only one moving part, a combined rotor-impeller assembly that is driven by the magnetic field of an induction motor. A portion of the pumped fluid is channeled to flow through the rotor cavity to cool the motor and lubricate the bearings. For an external circulated pump a self-cleaning filter can be provided to remove any solids from the recirculation fluid before it enters the bearing section of the motor. The stator windings and rotor armature are protected from contact with the recirculating fluid by a corrosion resistant, nonmagnetic, alloy liner (E) which completely seals or "cans" the stator winding.

Canned motor pumps can be modified for pumping hot fluids, up to 1000°F (538°C), volatile liquids, fluids with high melting points, liquids with solids, slurries and high pressure system pressures. To be used in these types of applications the recirculation flow must be conditioned to prevent any solids or vapors from entering or forming in the motor section of the pump.

Because there is only one rotating assembly, canned motor pumps do not require shaft or coupling alignments that are necessary for other types of pumps. Centerline supported pump casings are also not required for pumping liquids at elevated temperatures since the entire pump will "move" with the piping. Pumps can be installed in either the horizontal

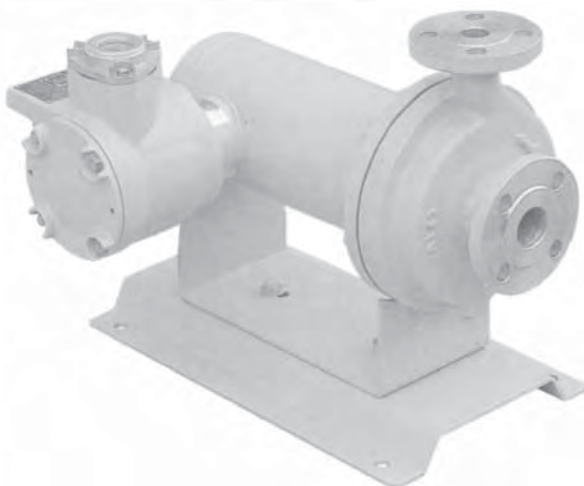


FIGURE 1 Typical canned motor pump (Teikoku Electrical Manufacturing Corp.)

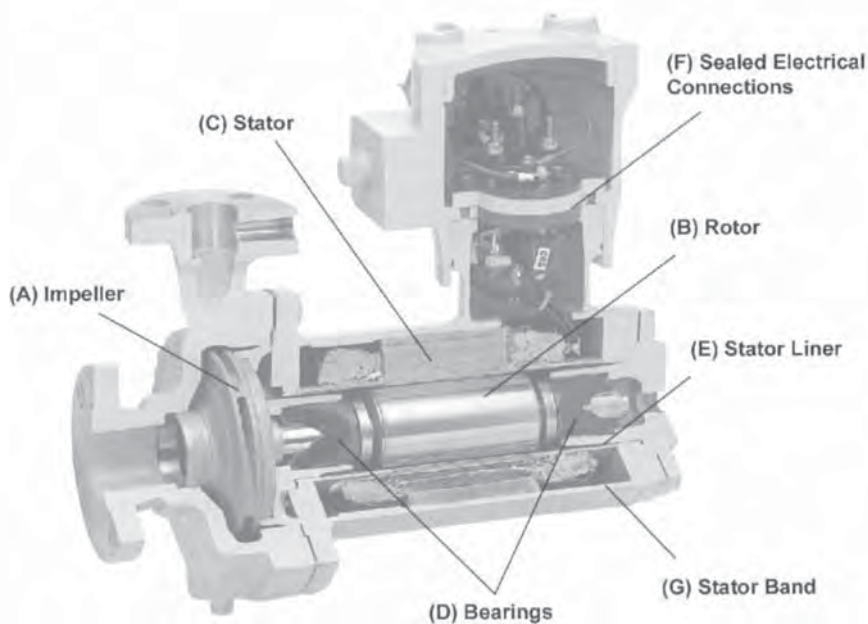


FIGURE 2 Components of a canned motor pump (Teikoku Electrical Manufacturing Corp.)

or vertical position, pump casing up or down. Mounting in the vertical position offers the advantage of greatly reducing the radial loads on the bearings. This feature is very significant with high horsepower motors or when pumping liquids with very low viscosities.

BASIC DESIGN

Stator Assembly The stator assembly of canned motor pumps (see Figure 3) consists of a set of one- or three-phase windings (A). Stator laminations (B) are constructed of low-silicon grade steel. Laminations and windings are mounted inside the cylindrical stator band (C). End bells (D and E), welded to the stator band, close off the ends of the stator assembly. The stator liner (F) is supported on the outside diameter by the steel lamination of the motor. Backup sleeves (G) are provided to strengthen those areas of the stator liner not supported by the stator laminations. The stator liner is, in effect, a cylindrical "can" placed in the stator bore and welded to the rear end bell and front end bell shroud to hermetically seal off the windings from contact with the liquid being pumped. Terminal leads (H) from the windings are brought out through a pressure tight lead connector (I) mounted on the stator band and terminated in an electrical connection box.

Motors are designed and manufactured specifically for use in canned motor pumps or components of conventional motors are modified. A wide variety of motor insulations are available ranging from temperature limits of 266°F (130°C) to above 700°F (371°C).

Canned motor pumps can be documented and labeled for use in explosion proof locations. Because the pump and motor are one unit, the complete assembly must be tested and approved for explosion proof applications. Explosion proof pumps are rated either as Class I, Group D, Division 1 or Class 1, Group C and D, Division 1.

Canned motor pumps perform very well when used with variable speed drives (VSDs). Unless the motor windings and bearing support systems are specifically designed for high speeds the speed range is usually limited to 1000 to 4200 revolutions per minute (rpm).

Rotor Assembly The rotor assembly is a squirrel cage induction rotor constructed and machined for use in canned motor pumps (see Figure 4). It consists of a machined corrosion

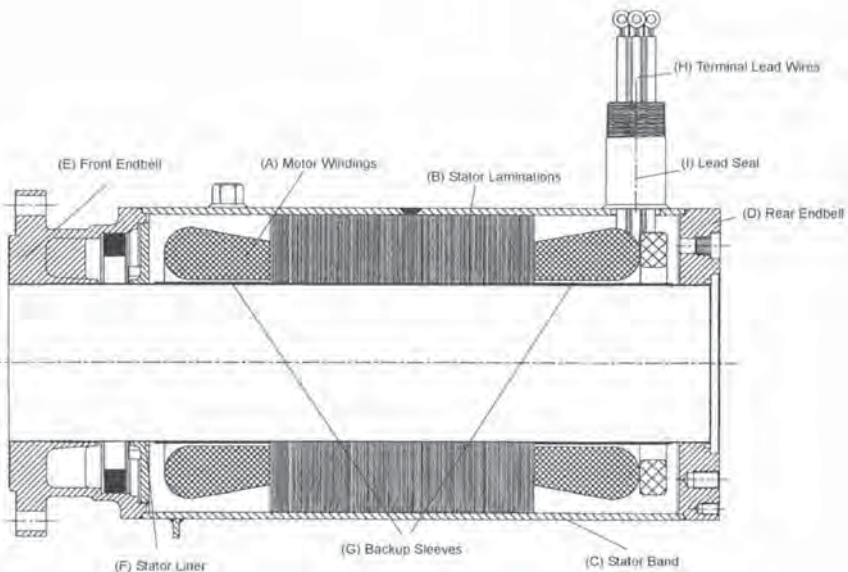


FIGURE 3 Stator assembly (Chempump, a Division of Teikoku USA Inc.)

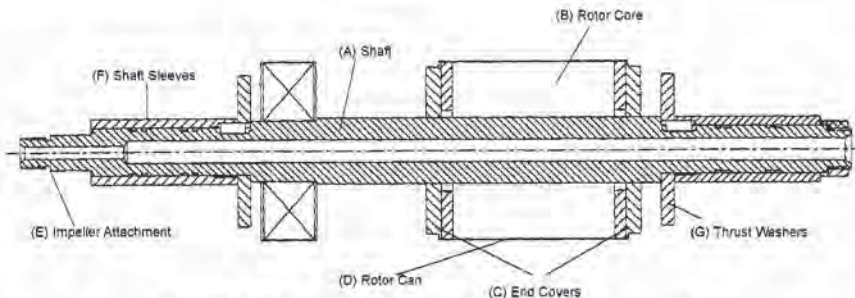


FIGURE 4 Rotor assembly (Chempump, a Division of Teikoku USA Inc.)

resistant shaft (A), laminated core (B) with copper or aluminum bars and end rings, corrosion resistant end covers (C), and a corrosion resistant sleeve (D). Various methods are used to attach the impeller to the motor shaft (E).

The rotor end covers are welded to the shaft and rotor sleeve, which surrounds the outside of the rotor, thus hermetically sealing off the rotor core from contact with the liquid being pumped.

Some manufacturers offer replaceable shaft sleeves (F) and axial thrust surfaces (G) for longer service life and ease of maintenance. Shaft sleeves can be constructed entirely of a hardened material such as silicon carbide and stellite or just the journal area can be coated with a hardened material such as tungsten carbide or chrome oxide.

Bearings Only two bearings are required for most canned motor pumps. These bearings are normally cooled and lubricated by the pumped fluid; therefore, the bearing material must be compatible with the process fluid. A multitude of materials are available, such as various grades of carbon graphite, antimony, silicon carbide, aluminum oxide and a variety of polymers. Bearing selection is dependent on the compatibility with the process fluid, the amount of solids present, and the pumping temperature.

A hydrodynamic bearing is the most common type of bearing used. The components of the bearings are both stationary (connected to the stator) and rotating with the rotor assembly. The fluid pumped fluid passes between the stationary bearing and shaft journal bearing causing the rotating assembly to run on a thin film of liquid; this should prevent contact between the journal and bearing. Most (stationary) bearings are grooved, either helical or straight grooves, on the inside diameter to increase the flow of process fluid through the journal area, thereby increasing the thickness of the fluid film and decreasing the temperature of the bearings.

Internal Clearances The overall gap between the stator windings and the rotor armature is critical to the design and operation of the canned motor pump. Increasing the gap between the iron of the motor windings and the rotor armature decreases the efficiency of the motor. Materials of construction of the stator liner also have an effect on motor efficiency. Stainless steels and Hastelloy are the most common materials used for stator liners and rotor sleeves. While stainless steel is less expensive, Hastelloy has higher corrosion resistance, is a stronger material and has lower electrical losses. Motor efficiency in canned motor pumps is not only important for energy cost considerations but also in the amount of heat input to the recirculation fluid.

The stator liner is a wetted, pressure boundary component; typically ranges in thickness from 0.010 to 0.040 in (0.254 to 1.016 mm), depending on the design pressure of the pump and other construction factors. For higher-pressure applications, the liner may remain at the same thickness, but its outside diameter is supported by the motor laminations and by backup sleeves located on both sides of the motor. By doing this, canned motor pumps have been designed to withstand working pressures up to 8000 lb/in² with 0.015-in (0.381-mm) stator liners, heavy walled backup sleeves, and reinforced stator cores.

The rotor armature, also a wetted component, is protected from the process fluid by a sleeve and two end covers. The thickness of the rotor sleeve typically ranges from 0.010 to 0.25 in (0.254 to 6.35 mm). The running clearance between the rotating motor armature and the stationary stator liner is usually about 0.020 in (0.508 mm) diametrically.

The total diametrical clearance between the journal and bearings ranges from 0.0025 to 0.067 in (0.0635 to 1.702 mm), and sometimes higher, depending on the specific manufacturer's design, bearing materials and power of the motor.

Secondary Containment Canned motor pumps offer a level of safety and process fluid containment unavailable with any other type of pump. Positive secondary containment of the process fluid is a built-in feature with canned motor pumps when the motor lead wires are housed in a pressure-retaining lead seal. In the event of a failure of the primary containment shell (stator liner) the outer stator band becomes a secondary containment vessel preventing the process fluid from entering the environment.

The outer stator band is far removed from the rotation element, making it impossible for the rotating element to make contact. The secondary containment system should be designed and tested to same pressure and temperature rating as the pump. Materials of construction of the stator band assembly should also be considered when pumping a highly corrosive fluid. When pumping dangerous fluids that require secondary containment, many users will install a device to detect the loss of the primary containment liner to shut the pump down. The most common device specified is a pressure switch mounted to the stator band assembly.

PRINCIPLE OF OPERATION

Flow Path Most canned pumps, when pumping relatively clean, nonvolatile fluids, will channel a small portion of the process fluid through the motor section. This fluid cools and lubricates the bearings and also removes heat generated by the motor. The circulation path can be either external or internal to the pump. With external circulation (see Figure 5), the recirculation fluid is piped outside of the pump, through a filter, then into the motor section of the unit. The filter assembly (see Figure 6) is self-cleaning and located in the discharge flange of the pump. Pumps having internal circulation have the recirculation

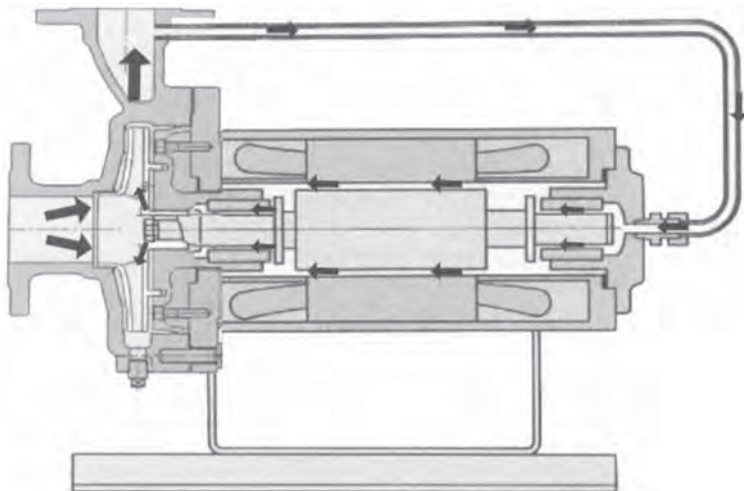


FIGURE 5 External circulation. (Teikoku USA Inc.)

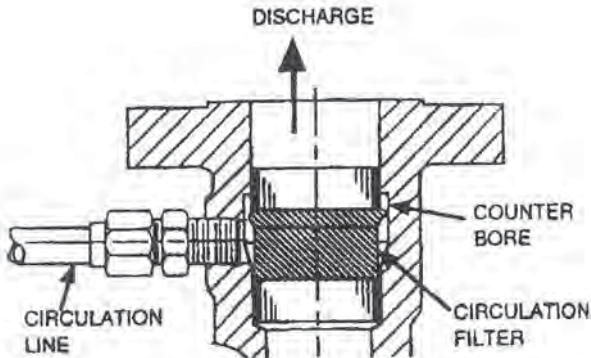


FIGURE 6 Filter assembly (Chempump, a Division of Teikoku USA Inc.)

contained within the pump. Filtering the recirculation liquid is not available with internal circulation.

In either external or internal circulation, the flow path is from the high-pressure area of the pump (pump discharge or pump chamber at the tip of the impeller) returning to the low-pressure area (eye of the impeller). The amount of liquid recirculated through the motor section ranges from 1 to 16 gpm (4 to 60 L/m) depending on the size of the pump and the pressure developed by the liquid end.

Thrust Balance The most common type of bearings used in canned motor pumps are product lubricated hydrodynamic bearings. With these types of bearings, the rotating assembly is allowed to float axially. This movement is known as "end play" and is defined as the movement of the rotor, in the axial direction, between the forward and rear contact points (normally the thrust bearings). A principle of hydraulic thrust balance is commonly used to position the rotation assembly between the forward and rear mechanical contact points. Based on hydraulic forces generated by the rotating impeller, automatic thrust balance is accomplished by the pressure of the pumped fluid. Pressure chambers are designed either within the pump chamber using the movement of the impeller or rear of the pump controlling the rate of return of the process fluid.

Figure 7 illustrates hydraulic thrust balance within the pump casing. Balance chambers are located on the front and rear of the impeller. When a change in axial load changes the position of the impeller, either forward or rear, there is an equalizing change of hydraulic pressure in the balance chambers, which immediately returns the rotation assembly into a balance position.

With the use of hydrodynamic bearings and hydraulic thrust balance, the rotating assembly is "floating" in the pumped fluid with no mechanical contact. For this reason it is best to operate canned motor pumps continuously or with a minimum of start-stops.

Applications Canned motor pumps can be used for every pumping application from clean, nonvolatile fluids to highly volatile, high-temperature, high-pressure, and slurry services. Numerous bearing and shaft materials are available along with various modifications to the internal bearing lubrication system to prevent vapors and solids from entering or forming in the bearing system.

When pumping volatile fluids, the motor section can be pressurized by an auxiliary impeller located on the rotor. The recirculation fluid, which normally returns to the eye of the impeller, is channeled to the pressurized section of the liquid end increasing the pressure in motor section (see Figure 8). This design allows a volatile fluid to remain liquid inside the motor cavity even with a temperature increase caused by motor heat. Another method to handle fluids near their boiling point is to reverse the recirculation flow path. Instead of returning the heated liquid back to the eye of the impeller the recirculation liquid

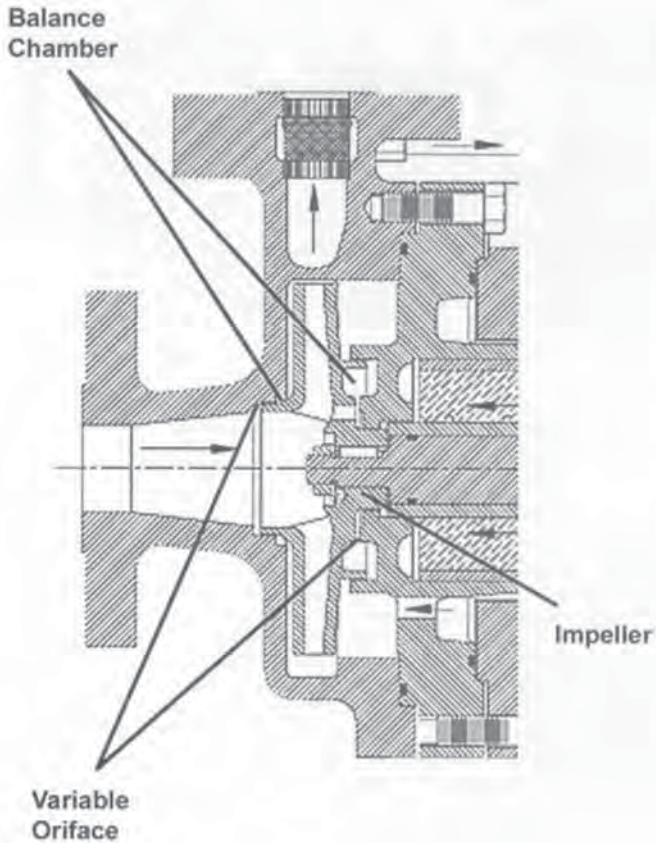


FIGURE 7 Hydraulic thrust balance (Chem pump, a Division of Teikoku USA Inc.)

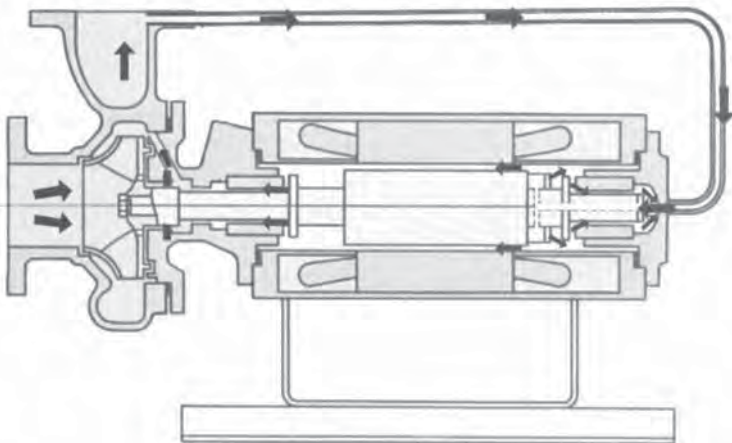


FIGURE 8 Pressurized circulation (Teikoku USA Inc.)

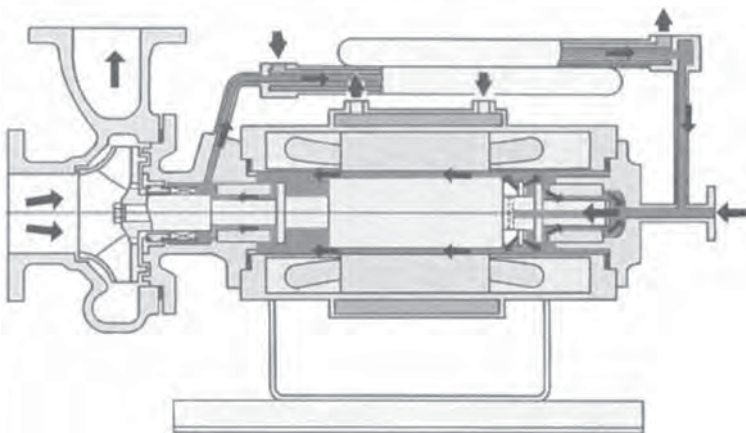


FIGURE 9 Externally cooled motor (Teikoku USA Inc.)

exits the pump and is returned to the suction vessel. The reverse circulation option provides a continuous vent of the motor chamber.

For pumping fluids at temperatures above 400°F (200°C) water-cooled and non-water-cooled designs are available. The water-cooled design isolates the motor and bearings from the pumped fluid (see Figure 9). The circulating fluid in the motor section is used for lubrication of the bearings and cooling of the motor. This fluid remains in the motor section and is circulated through the rotor cavity by an auxiliary impeller, which is an integral part of the rotor assembly. The fluid in the motor section is forced across the rotor and through the bearings then through an auxiliary heat exchanger. The heat exchanger is cooled by water or a suitable heat transfer fluid. The non-water-cooled design has a motor insulation system constructed of materials designed to operate at high temperatures. The material of the bearings must also be designed for temperatures in excess of 650°F (350°C). The power lead connection is extended away from the motor so the electrical connection box is at ambient temperatures (see Figure 10).

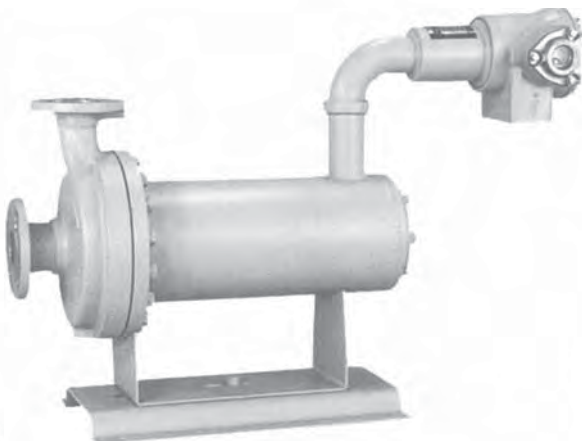


FIGURE 10 High temperature non-water-cooled motor (Teikoku Electrical Manufacturing Corp.)

The slurry design has the same modifications as the high-temperature, water-cooled design with the addition of a closure seal between the motor and the pump to prevent any solids from entering the motor chamber. This design completely isolates the slurry in the pump casing from the liquid in the motor. A clean fluid is back flushed through the motor to lubricate the bearings and cool the motor. A small portion of the flush fluid flows, passes through the closure seal into the process fluid. The amount of liquid mixing with the process fluid varies depending on the type of closure seal, but is usually much less than 1 gpm (4 L/m).

Pressure-Temperature Profile The single most important consideration in the application and successful operation of canned motor pumps is bearing environment control. The process fluid cools and lubricates the bearings and removes the heat generated by the motor. The fluid cooling the bearings must remain in a liquid state to provide adequate bearing lubrication.

A number of variables must be considered when considering the condition of the recirculation fluid. These variables include: vapor pressure, specific heat, specific gravity, viscosity, pump efficiency, motor efficiency, motor load, recirculation flow rate, and the recirculation flow pressure. The physical properties of the fluid should be provided as a function of temperature. Figure 11 illustrates the heat rise calculations for a specific application. Figure 12 is an example of a typical pressure-temperature profile of a sealless pump. If the flow rate varies, a profile should be calculated for each design condition.

A pressure-temperature profile should be calculated for any application where there is any doubt concerning the condition of the recirculation fluid. As a minimum it is recommended to have a pressure-temperature profile for maximum, minimum, and design flow and temperature conditions.

Installation Installation of a canned motor pump is much less costly than other pumps because canned motor pumps do not require any special base plates or mounting pads. In fact, many canned motor pumps are stilt mounted or bolted directly into the system piping. There is only one shaft in a canned motor pump, therefore, alignment is not necessary nor is pump orientation. The unit can be mounted either horizontally or vertically, with the pump casing up or down. The sleeved bearings are located in the bearing housing. Alignment of the bearings is accomplished by a registered fit between the bearing housings and the stator assembly so all internal alignment is accomplished at final assembly.

Diagnostics Diagnostic systems are available to monitor and display the condition of the internal wear components of the pump. More advanced diagnostic systems will continuously indicate both axial and radial wear by monitoring the position of the rotating assembly. Continuous wear indication allows the pump user to trend the wear of the pump and schedule standard wear parts replacement long before a major failure would occur. Remote outputs are also available on these diagnostic systems for the user to link the pump wear data to a data control system (DCS). These outputs include a digital, RS-485 signal, an analog 4-20 mA, or a relay designed to shut down the pump or signal an alarm when a certain amount of bearing wear has occurred.

These diagnostic tools are the single most important advancement in the technology of pumping equipment. Having access to data on the condition of the pump provides the user with the opportunity to perform predictive and preventative maintenance on equipment and systems. Figure 13 illustrates the type of information available when using an RS-485 output linked to a data collection system.

Industrial Standards Due to the unique design of canned motor pumps (pump and motor combined into a single unit), they are designed to conform to many international standards. Motors can be supplied to meet many international standards as well. Third party testing and certification are available from:

- Underwriter Laboratories (UL)
- Factor Mutual (FM)

Heat Rise Calculations

PUMP MODEL NC-A60-10-N5
 FLUID TRICHLOROSILANES
 Bypass, GPM 0

CIRCULATION PATH	STANDARD	REVERSE	PRESSURIZED
Desired Flow (GPM)	300	300	300
TDH (FT)	240	240	240
Temperature, inlet (F)	150	150	150
Specific Heat, (btu/lb F)	0.26	0.26	0.26
Specific Gravity	1.24	1.24	1.24
Hydraulic Horsepower	22.5	22.5	22.5
Total Pump Flow (GPM)	300	300	300
Pump Efficiency - Note 1.	58	58	58
Motor Output (HP)	38.9	38.9	38.9
Delivered Pump Head, (FT)	240	240	240

Motor Efficiency - Note 3.	80		80
Motor Input (KW)	36.2		36.2
Overall Efficiency	46.4		46.4
Impeller Diameter (IN)	8.5		8.5
Motor Circulation Flow (GPM)	7		6
Process Wire-Water Eff.	46.4	46.4	46.4
Total Circ rate (GPM) - Note 2.	9	6	6
Suction Pressure PSIA at NPSH= 61 FT	77.3	77.3	77.3

TEMPERATURE RISE

Standard circ (pump)	1.4	0.8	1.4
temp. pump exit	151.4	150.8	151.4
circ flow (motor)	23.2	25.6	25.6
total temperature rise	24.6	26.4	26.9
max temp. circ. flow	174.6	176.4	176.9
Est. P vap, psia	63	64.6	
Percent max circ flow		0.9	
Min. Motor pressure, psig		112.1	
Circ press/ differential		0.27	

Note 1. Efficiency at "Total pump flow"

Note 2. Total Circ Flow includes front bearing flow (does not affect temp rise).

Note 3. Motor efficiency includes windage loss (correct for specific gravity)

Note 4. System heat added with reverse circulation is "total heat" less "motor heat".

HEAT BALANCE

Total heat input (btu/min)	1104.7	1104.7	1104.7
Standard circ (pump)	1105.9	645.5	1105.9

FIGURE 11 Heat rise (Chempump, a Division of Teikoku USA Inc.)

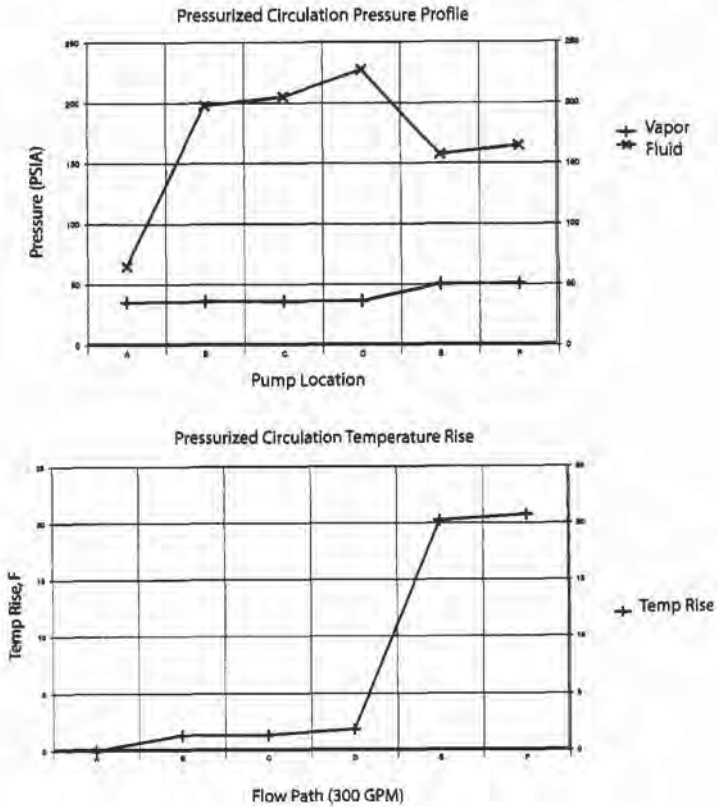
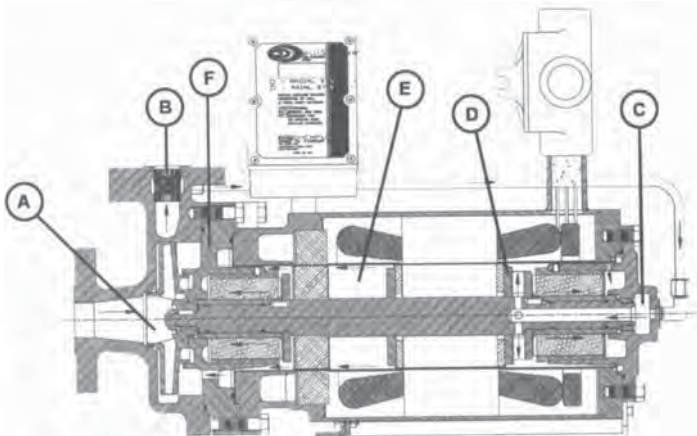


FIGURE 12 Pressure-temperature profile (Chempump, a Division of Teikoku USA Inc.)

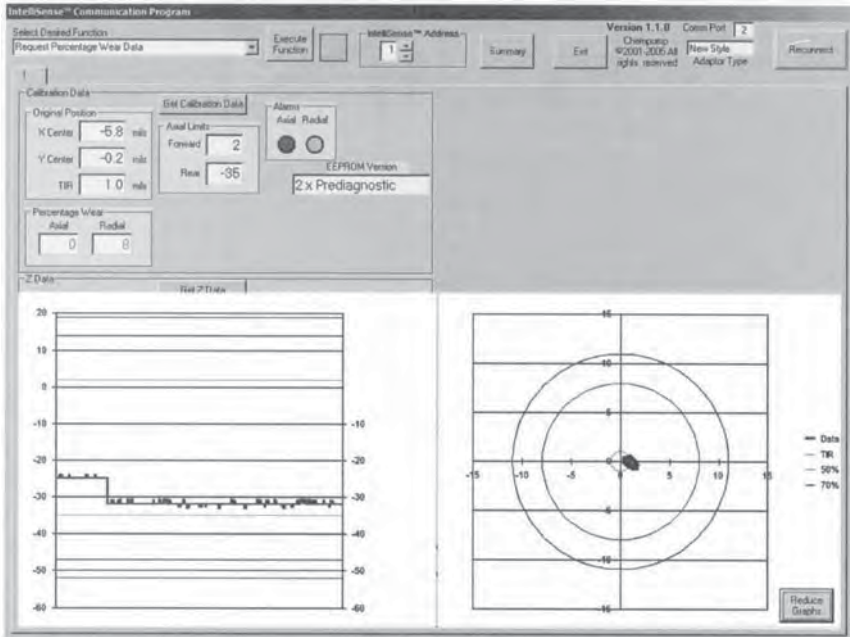


FIGURE 13 Rotor position output data

- Canadian Standards Association (CSA)
- Japanese Industrial Standards (JIS)
- ATEX
- CE
- Institute of Electrical and Electronics Engineers (IEEE)

Materials can be purchased to meet the standards of the American Society of Testing and Materials (ASTM). Certified Material Test Reports (CMTR) can also be supplied.

Designs can conform to:

- American Society of Mechanical Engineers (ASME NCA-4000)
- Canadian Standards Association (CSA-Z299.3)
- Military Standard (MIL-I-45208)

Complete pumps can meet the requirements of:

- ANSI/HI 5.1-5.6-2000, "American National Standard for Sealless Centrifugal Pumps"
- ASME-B73.3-3M-2003, "Specification for Sealless Horizontal End Suction Centrifugal Pumps for Chemical Process"
- API Standard 685, "Sealless Centrifugal Pumps for Petroleum, Heavy Duty Chemical and Gas Industry Services"
- ISO 15783-2002, "Sealless Rotodynamic Pumps-Class II"
- ISO 13709-2003, "Centrifugal pumps for petroleum, petrochemical and natural gas industries"

FURTHER READING

American National Standard for Sealless Centrifugal Pumps, ANSI/HI 5.1-5.6-2000, Hydraulic Institute, Parsippany, NJ. www.pumps.org.

Hydraulic Institute ANSI/HI 2000 Edition Pump Standards, Hydraulic Institute, Parsippany, NJ. www.pumps.org.

Sealless Pumps for Petroleum, Heavy Duty Chemical, and Gas Industry Services, API Standard 685, 2000, American Petroleum Institute, 1220 L Street, Northwest, Washington, D.C. www.api.org.

International Organization of Standards, ISO 15783:2002 and 13709:2003. 1 rue de Varembe, Case postale 56, CH-1211 Geneve 20, Switzerland. www.iso.org.

Jaskiewicz, S., and Rosenbaum, M. "Intelligent Sealless Pumps—A Unique Evolution." *World Pumps*, Vol. 2003, Number 442, July 2003, pp. 16–21.

Littlefield, D. "Sealless Centrifugal Pumps." *Proceedings of the Eleventh International Pump Users Symposium*. Texas A&M University, College Station, TX, March 1994, pp. 115–119.

Le Blue, J., Jr., and Lobach J. "Continuous Monitoring of Sealless Pumps." *Proceedings of the Fifteenth International Pump Users Symposium*. Texas A&M University, College Station, TX, March 1998, pp. 125–131.

Sealless Rotodynamic Pumps—Class II, ISO 15783:2002. International Organization for Standardization (ISO), Geneva, Switzerland. www.iso.ch

DISPLACEMENT PUMPS

SECTION 3.1

POWER PUMP THEORY

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A power pump is a positive displacement machine consisting of one or more cylinders, each containing a piston or plunger. The pistons or plungers are driven through slider-crank mechanisms and a crankshaft from an external source. The capacity of a given pump is governed by the rotational speed of the crankshaft.

Unlike a centrifugal pump, a power pump does not develop pressure; it only produces a flow of fluid. The downstream process or piping system produces a resistance to this flow, thereby generating pressure in the piping system and discharge portion of the pump. The flow fluctuates at a rate proportional to the pump speed and number of cylinders. The amplitude of the fluctuations is a function of the number of cylinders. In general, the greater the number of cylinders, the lower the amplitude of the flow variations at a specific rpm.

All power pumps are capable of operating over a wide range of speeds, thereby making it possible to produce a variable capacity when coupled to a variable speed drive. Each pump has maximum suction and discharge pressure limits that, when combined with its maximum speed, determine the pump's power rating. The pump can be applied to power conditions that are less than its maximum rating but at a slight decrease in mechanical efficiency.

The power pump is a positive displacement device. When operating, it will continue to deliver flow independent of the pressure in the discharge piping system. Unlike a centrifugal pump, a power pump will not "deadhead" or "go back on its curve" in response to increasing discharge pressure. When this pressure exceeds the design limits of the pump, mechanical failure—often catastrophic—will result. For this reason, all piping systems incorporating power pumps must have discharge pressure relief devices to limit the pressure in the piping system and avoid pump failure. These devices must be located between the discharge connection on the pump and the first isolation valve in the piping system.

This section has been organized to provide sufficient power pump theory to properly select a pump for most applications. It is recognized that the proper pump metallurgy for the pumped fluid must be used, but due to the vast number of possible fluids, selection of pump metallurgy is outside the scope of this section.

The subject of Net Positive Suction Head (NPSH) is mentioned in several places in this section and is covered in more detail in Section 3.4, "Displacement Pump Performance, Instrumentation, and Diagnostics." For basic power pump selection, it is only necessary to understand that the NPSH available from the suction system must be sufficiently above the NPSH required by the pump to operate properly.

SELECTION THEORY

Power Brake horsepower (bhp) is a function of a pump's capacity (flow rate), differential pressure, and overall efficiency. It is an essential criterion for selecting the drive components but is not valuable for pump selection. A large pump operating well below its design rating can meet the same horsepower requirements as a smaller pump running at a higher speed. Unless the application requires a derated pump, it is usually more economical to select a pump at the upper end of its design rating.

The brake horsepower for the pump is

$$\text{In USCS units} \quad bhp = (Q \times Ptd \times 100)/(1714 \times OE)$$

$$\text{In SI units} \quad kW = (Q \times Ptd \times 100)/(36 \times OE)$$

where Q = delivered capacity (flow rate), gpm (m^3/h)

Ptd = differential pressure (discharge – suction), lb/in^2 (bar)

OE = overall efficiency, %

NOTE: One bar equals 10^5 Pa.

Capacity The capacity Q is the total volume of fluid delivered per unit of time. This fluid includes liquid, entrained gases, and solids at the specified conditions.

Displacement Displacement D , gpm (m^3/h), is the calculated capacity (volume flow rate) of the pump with no slip losses. For single-acting plunger or piston pumps, this is

$$\text{In USCS units} \quad D = A \times m \times n \times s / 231$$

$$\text{In SI units} \quad D = A \times m \times n \times s \times 6 \times 10^{-5}$$

where A = cross-sectional area of plunger or piston, in^2 (mm^2)

m = number of plungers or pistons

n = rpm of pump

s = stroke of pump, in (mm) (half the linear distance the plunger moves in one revolution)

$$231 = \text{in}^3/\text{gal}$$

For double-acting plunger or piston pumps, this is

$$\text{In USCS units} \quad D = (2A - a)m \times n \times s / 231$$

$$\text{In SI units} \quad D = (2A - a)m \times n \times s \times 6 \times 10^{-5}$$

where a is the cross-sectional area of the piston rod, in^2 (mm^2).

Pressure The pressure Ptd used to determine brake horsepower is the differential pressure or discharge pressure minus the suction pressure. In most applications, the suction pressure is small relative to the discharge pressure. However, when pumping some compressible liquids, such as methane and propane, the suction pressure may be 20 to 30% of the discharge pressure. For accurate brake horsepower calculations, always include the suction pressure. Figure 1 shows a typical performance curve for a power pump.

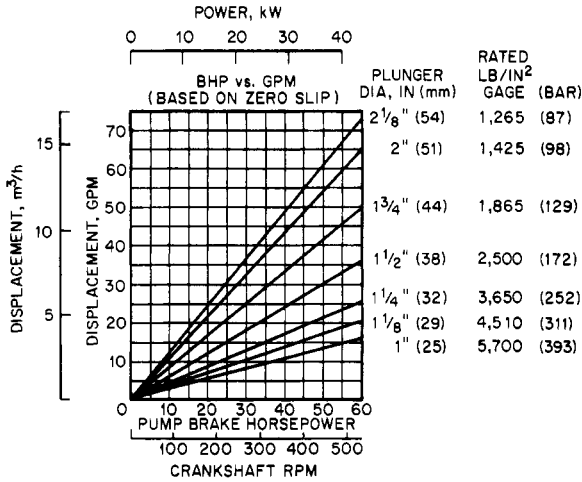


FIGURE 1 Typical power pump performance: brake horsepower versus gallons per minute (kilowatts versus m³/h) based on zero slip (Flowserve Corporation)

Slip Slip (*S*) is the loss of capacity due to internal and external pump leakage. External leakage occurs primarily through the stuffing box via the packing. Internal leakage is primarily the backflow past the suction and discharge valves. Backflow occurs when a valve remains open for a fraction of a second as the plunger or piston reverses direction. A small amount of leakage may occur across the piston in a double acting pump from the high pressure side to the low pressure side. Slip is expressed as a percentage loss of the displacement flow rate and is typically 1 to 4%:

$$S = 100 \times (B + V + L) / D$$

where *S* = Slip, %

B = Leakage through the stuffing box

V = Backflow across the valves

L = Internal leakages

Fluid viscosity, pump speed, and discharge pressure can all have an effect on slip, which is shown in Tables 1 and 2.

Mechanical Efficiency The mechanical efficiency of a power pump is

In USCS units $ME = \text{displacement power} / \text{power in} = P_{\text{displ}} / P_{\text{in}} = D \times Pdt / 1714 \times P_{\text{in}}$

In SI units $ME = \text{displacement power} / \text{power in} = P_{\text{displ}} / P_{\text{in}} = D \times Pdt / 36 \times P_{\text{in}}$

where *P_{in}* is the input power from the driver, bhp (kW).

The mechanical efficiency of a power pump is the sum of all the frictional losses in the fluid and power ends. These include the plungers and packing, the crossheads, the rod seals, and the bearings. The efficiency of a single acting pump often exceeds 90%, while a

TABLE 1 Slip in a pump with a plate valve

Centistokes	100	1,000	2,000	6,000	10,000	12,000
Slip, %	8	8.5	9.5	20	41	61

TABLE 2 Slip as a function of pump speed and pressure

Pressure, lb/in ² (bar)	Slip, %		
	At 440 rpm	At 390 rpm	At 365 rpm
4000 (275)	11	22	34
3000 (207)	9	20	31
2000 (138)	7	18	30
1000 (69)	7	15	27.5

double-acting piston pump will be 88% due to the additional piston and rod seals. If the pump is equipped with internal gearing, an additional 2% loss is common.

Most power pumps are designed to accept a range of plunger or piston sizes. When the larger plungers are used, the increased diameter of the packing/seals and plunger/liners will result in higher frictional losses than with smaller components. As a rule, doubling the plunger/piston diameter will decrease the mechanical efficiency by 8%. Mechanical efficiency is also affected by speed and, to a lesser extent, by developed pressure, as indicated in Tables 3 and 4.

Speed Pump speed, or, more correctly, stroke rate, is one of the most critical selection criteria for power pumps. The rotating and reciprocating parts of the power end, as designated, are often capable of speeds twice that of the actual pump rating. The maximum pump speed is determined by the design of the fluid end, the hydraulic capability of the anticipated suction system, and the required life of the plungers, packing, and valves. Most power pump standards limit the plunger speed from 140 to 280 ft/min (0.71 to 1.42 m/s). The plunger speed is

In USCS units $Sp = s \times n/6$

In SI units $Sp = s \times n/30,000$

where Sp = plunger or piston speed, ft/min (m/s).

s = stroke of the pump, in (mm) (half of the linear distance the plunger or piston moves in one revolution)

n = rpm of pump

All pumps have a minimum speed limit, usually determined by a decrease in the adequate lubrication to the bearings in the power end.

Volumetric Efficiency Volumetric efficiency (VE) is the ratio of the discharge volume to the suction volume, expressed as a percentage, plus the slip. It is proportional to the ratio r and the developed pressure where r is the ratio of the internal volume of fluid between valves when the plunger or piston is at the top of the peak of its back stroke ($C + D$) to the plunger or piston displacement (D) (see Figure 2).

Since the discharge volume cannot be readily measured at discharge pressure, it is taken at suction pressure. Taking the discharge volume at the suction pressure results in a higher volumetric efficiency than using the calculated discharge volume at discharge

TABLE 3 The effect of speed on mechanical efficiency at a constant developed pressure

% of full speed	44	50	73	100
ME, %	93.3	92.5	92.5	92.5

TABLE 4 The effect of pressure on mechanical efficiency at constant speed

% of full-load developed pressure	20	40	60	80	100
ME, %	82	88	90.5	92	92.5

pressure because of fluid compressibility. Compressibility becomes important when pumping water or other liquids over 6000 lb/in² (414 bar), and it should be taken into consideration when determining the actual delivered capacity into the discharge system.

Figure 3 shows the approximate volumetric efficiency for water (not including slip).

Based on the expansion back to suction capacity,

$$VE = 100 \times (1 - Ptd \times \beta \times r) / (1 - Ptd \times \beta) - S$$

Based on discharge capacity,

$$VE = 100 \times (1 - Ptd \times \beta \times r) - S$$

where β is the compressibility factor of the liquid being pumped. Figure 4 shows approximate values of β for various liquids.

When the compressibility factor is not known, but the suction and discharge density can be determined in pounds per cubic foot, the following equations can be used for calculating VE:

Based on suction capacity,

$$VE = 100 \times [r - (p_d/p_s)(r - 1)] - S$$

Based on discharge capacity,

$$VE = 100 \times [1 - r(1 - p_s/p_d)] - S$$

where p_s = suction density

p_d = discharge density

S = slip

Torque The average torque required by a power pump is independent of the pump speed, assuming the suction and discharge pressures are held constant. This means that the pump is a "constant torque" device and, unlike a centrifugal pump, will have a flat speed-torque curve. The torque required at the input shaft is as follows:

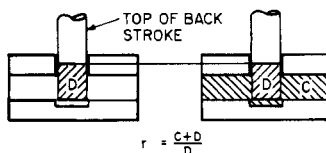
In USCS units $M = p \times 5250/n$

In SI units $M = p \times 9.549/n$

where M = pump torque, lb-ft (N · m)

n = speed, rpm

p = power, bhp (kW)

**FIGURE 2** The ratio r (Flowserve Corporation)

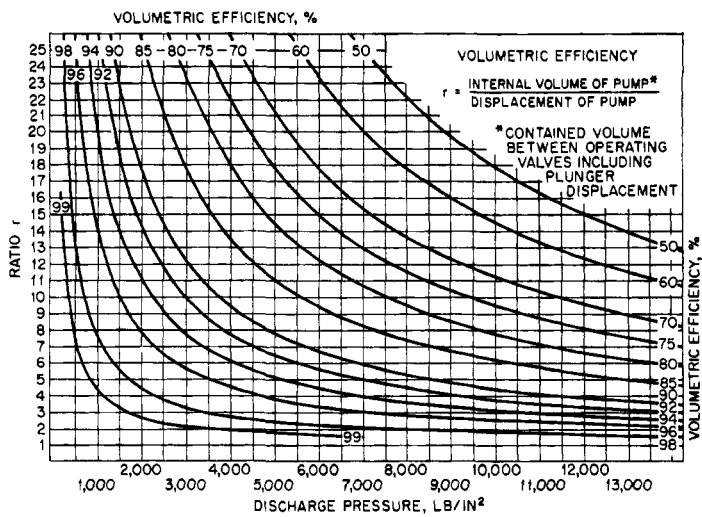


FIGURE 3 Volumetric efficiency (Flowserve Corporation) ($\text{lb/in}^2 \times 0.69 = \text{bar}$)

Break-away torque, or start-up torque, is the torque required to initiate the motion of the pump to enable it to accelerate to a constant speed. If the pump is started against an open bypass to pump suction, the torque requirement is 25% of the running torque. If the pump is started against full discharge pressure, the torque requirement is 125% of the running torque. When selecting a drive system, the break-away torque of the pump must

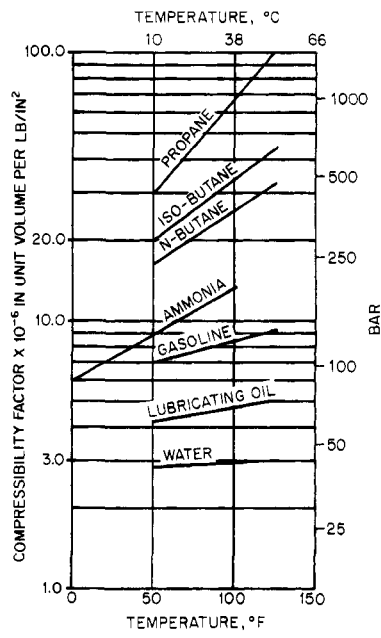


FIGURE 4 Compressibility factor (Flowserve Corporation) ($\text{bar} \times 14.5 = \text{lb/in}^2$)

TABLE 5 The pulses per revolution and pulse amplitude for various pump types

Pump Type	No. of Cylinders	Pulses per Rev.	Pulse Amplitude
Duplex, double-acting	4	8	+24% - 22%
Triplex, single-acting	3	6	+ 6% - 17%
Quintuplex, single-acting	5	10	+ 2% - 5%
Septuplex, single-acting	7	14	+ 1.2% - 2.6%
Nonoplex, single-acting	9	18	+0.6% - 1.5%

be considered. Special “high torque” motors are often required if the pump is to be started under the load.

If a variable frequency drive (VFD) is used for pump speed/capacity control, torque fluctuations or pulsations may require the drive manufacturer to provide special electronic dampening circuits. The normal pulsating flow of the pump causes the input power to fluctuate at approximately the same magnitude as the flow pulses. The frequency of these pulses per revolution of the crankshaft is twice the number of cylinders (see Table 5).

If the pump has internal gearing, or if the pump is being driven by an external gear reducer, the torque pulsation values listed should be divided by the gear ratio.

Derating Power pumps can be derated in order to achieve acceptable performance or part life in a variety of special applications. The most common derating is pump speed. Experience has shown that the life of fluid-end expendable parts, such as plungers, packing, and valves, can be extended if the pump speed/plunger velocity is reduced when pumping certain fluids. Some guidelines for acceptable plunger speeds are shown in Table 6.

Pumping high-viscosity fluids may also require the speed of a power pump to be derated. Some attempts have been made to establish standards for speed derating as a function of the fluid viscosity. Unfortunately, these methods have been based on power pump designs that were established for water or similar low-viscosity services. If a pump has been designed specifically for a high-viscosity service, only moderate speed derating may be required. Generally, pumps designed with large internal fluid passages, low-valve velocities, and larger plungers or pistons will need less derating.

Power pumps should be derated for a rod load if system or process “upsets” cannot be avoided. These are usually applications where the gas content of the pumped fluid can lead to abnormal pressure pulsations or cavitations in the fluid cylinder.

TABLE 6 Guidelines for acceptable plunger speeds

Fluid	Plunger Speed, ft/min (m/s)
Cold water	315–354 (1.6–1.8)
Hot water 140–194°F (60–90°C)	217–256 (1.1–1.3)
Hot water over 194°F (90°C)	157–217 (0.8–1.1)
Salt water	236–276 (1.2–1.4)
Cold oil	315–354 (1.6–1.8)
Hot oil	256–295 (1.3–1.5)
Crude oil	217–276 (1.3–1.5)
Liquid ammonia	197–236 (1.0–1.2)
Carbamate	138–158 (0.7–0.8)
Slurry	158–197 (0.8–1.0)
Fatty acid	236–295 (1.2–1.5)
Light hydrocarbons	177–236 (0.9–1.2)
Glycol	256–295 (1.3–1.5)
High-viscosity liquids	138–197 (0.7–1.0)

FLUID END THEORY

Pumping Cycle Unlike the relatively smooth, continuous flow of fluid through a centrifugal pump, the flow of fluid through a power pump occurs in a transitory dynamic condition called a *pumping cycle*. The event that initiates this cycle is the linear movement of the plunger or piston. In Figure 5, r is the radius of the crank in feet (meters), L is the length of the connecting rod in feet (meters), C equals L/r , and ω equals $(2\pi/60) \times \text{rpm}$. Thus, X , the linear movement of the piston or plunger is

$$X = r \left[1 - \cos \theta + L \left(1 - \sqrt{1 - \frac{r^2}{L^2} (\sin \theta)^2} \right) \right]$$

As the plunger (or piston) withdraws from the fluid cylinder or pumping chamber, the volume of the cylinder increases. The pressure in the cylinder decreases in response to the increased volume. Since most of the fluids handled by power pumps are relatively incompressible, very little plunger movement is required to cause a pressure drop. When the cylinder pressure drops sufficiently below suction pressure, the differential pressure begins to open the suction valve. The valve opens gradually and smoothly at the start of the suction stroke because the velocity and acceleration of the plunger are small.

Fluid flows through the suction valve assembly, following the plunger and filling the cylinder. As the plunger decelerates at the end of the suction stroke, the suction valve gradually returns to its seat. Ideally, the suction valve is completely closed as the plunger comes to a stop.

The motion of the slider-crank mechanism causes the plunger to reverse direction and start its discharge stroke. The fluid trapped in the fluid cylinder is compressed until the cylinder pressure exceeds the discharge pressure by an amount sufficient to begin to open the discharge valve. As with the suction valve, the discharge valve continues to open until it reaches its travel limit or until the velocity of fluid through the valve becomes constant. As the plunger decelerates, the valve moves back toward its seat. Again, ideally, the discharge valve closes when the motion of the plunger stops.

The number of pumping cycles in a single revolution of the crankshaft is the same as the number of cylinders in the pump. Every cylinder will "pump" in a sequence determined by the "firing order" of the crankshaft. The cylinders are arranged in parallel, with each one discharging into a common discharge manifold. In industry terms, the pump is usually identified by the number of plungers or pistons on the crankshaft. They are the same for single- or double-acting pumps (see Table 7).

Pulsations The pulsating characteristics of the fluid flowing into and out of power pumps are significantly influenced by the number of plungers or pistons. Discharge flow pulsations are the most critical because of the high energy potential generated when the system resistance reacts with the flow to create pressure. Since the magnitude of the discharge pulsation is mostly affected by the number of cylinders, increasing the number of cylinders will reduce the flow pulsations.

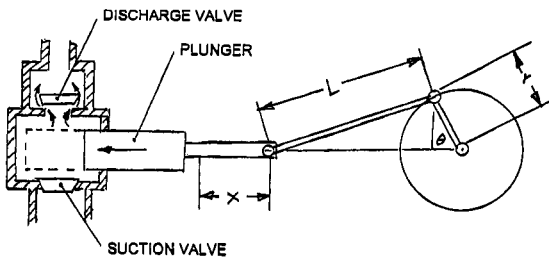


FIGURE 5 Slider-crank mechanism

TABLE 7 Industry terms for the number of plungers or pistons

Number	Term
1	Simplex
2	Duplex
3	Triplex
4	Quadruplex
5	Quintuplex
6	Sextuplex
7	Septuplex
9	Nonuplex

The acceleration and velocity of the plunger or piston will determine the rate that the fluid is discharged from the cylinder. The approximate velocity of the plunger or piston is

$$S = \frac{dx}{dt} = r \left(\sin \theta + \frac{\sin 2\theta}{2C} \right) \omega$$

The approximate acceleration of the plunger or piston is

$$Cp = \frac{d^2x}{dt^2} = r \left(\cos \theta + \frac{\cos 2\theta}{C} \right) \omega^2$$

The momentary rate of discharge capacity is the cross-sectional area A of the plunger or piston times velocity:

$$\text{For single-acting plunger or piston} \quad A = 0.785 \times D^2$$

$$\text{For double-acting piston} \quad A - a = 0.785 (D^2 - d^2)$$

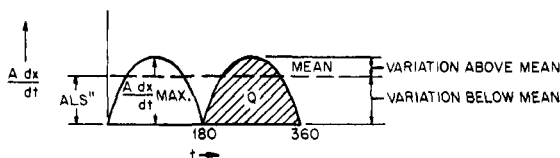
The total discharge Q is equal to ALS' , where S' is the number of effective strokes in a given time. The quantity Q is the area of the curve, the mean height of which is

$$\text{Total } Q/t = ALS'/t = ALS''$$

where S'' is the number of strokes per second and t is time in seconds.

Figures 6 and 7 show discharge rates for two types of pumps. Tables 8 and 9 show how variations from the mean are related to the change in the number of plungers with a C of approximately 6:1. This shows that pumps with an even number of cylinders have a higher flow variation than pumps with an odd number of cylinders. Table 9 shows the variations in a triplex with changes in $C = L/R$.

Valves A pump valve in its simplest form is a free-moving plug that is opened when the force of the liquid below the valve exceeds the force of the liquid above it. When the force above the valve becomes greater than the lower, the plug closes and forms an effective seal to fluid "backflow" and pressure loss. If the valve does not perform this function efficiently, the performance of the pump can be degraded to the point where no flow is produced.

**FIGURE 6** The discharge rate for a single double-acting pump (Flowsolve Corporation)

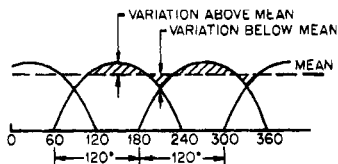


FIGURE 7 The discharge rate for a triplex single acting pump (Flowserve Corporation)

A valve will be in equilibrium when the forces above and below the valve are balanced (see Figure 8):

$$\text{Equilibrium} = P_1 \times A_1 = P_2 \times A_2 + S_f + M$$

where P = the pressure below the valve

A = the area below the valve exposed to P

P = the pressure above the valve

A = the area above the valve exposed to P

S_f = the force of the valve spring (if any)

M = the mass of the valve and half the mass of the spring

The significance of this simple equation becomes apparent when you remember that the valve is operating in a dynamic environment of constantly changing pressures in a time frame that is measured in tenths of a second. Add to that the requirement that the valve must pass a volume of fluid in each opening cycle with minimal pressure drop, and it becomes clear that the pump valve is the most critical component in the fluid end in terms of pump operability.

The pump designer will size the valves to provide a flow area, called the *spill area*, that prevents preestablished velocity limits from being exceeded. The spill area of various valve types is given in Table 10. The velocity of the fluid flowing through the spill area is called the spill velocity, V , shown in Table 11:

In USCS units V (ft/sec) = gpm through the valve $\times$ 0.642/spill area of the valve, in²

In SI units V (m/s) = m³/h through valve $\times$ 555.6/spill area of the valve, mm²

The quantity 0.642 (555.6) is used because all the liquid passes through the valve in half the stroke.

Valve Dynamics Valve dynamics is the mechanical response of the valve to the changes of pressure across the valve. Using a suction valve as an example, the valve starts its cycle

TABLE 8 The effect of the number of plungers on variations in capacity from the mean (C is approx. 6:1)

Type	Number of Plungers	% Above Mean	% Below Mean	Total %	Plunger Phase
Duplex	2	24	22	46	180 deg.
Triplex	3	6	17	23	120 deg.
Quadruplex	4	11	22	33	90 deg.
Quintaplex	5	2	5	7	72 deg.
Sextuplex	6	5	9	14	60 deg.
Septuplex	7	1	3	4	51.5 deg.
Nonuplex	9	1	2	3	40 deg.

TABLE 9 The effect of change in C on variations in capacity from the mean for a triplex pump

C	% Above Mean	% Below Mean	Total %
4:1	8.2	20.0	28.2
5:1	7.6	17.6	25.2
6:1	6.9	16.1	23.0
7:1	6.4	15.2	21.6

when the valve is closed and the plunger is at the start of its suction stroke (maximum insertion into the cylinder). As the plunger starts withdrawing from the cylinder, the internal volume in the cylinder starts to increase. This increasing volume results in decreasing cylinder pressure that, in turn, hydraulically unbalances the valve and the valve begins to open. There is usually a slight lag in the valve opening versus the start of the plunger motion of about 5 to 20° of crankshaft rotation. Traditionally, the opening valve lag is attributed solely to the inertia required to set the valve in motion and the preload of the valve spring. In the last few years, the theory of *valve stiction* has been proposed as an additional cause of valve opening lag.

Valve stiction, as the name implies, can be an additional force to overcome when the valve is trying to lift off its seat. One version of the theory is that a cohesive force exists between the fluid molecules and the sealing surfaces of the valve and seat. This can be demonstrated by trying to separate two wet, highly machined plates. The second stiction theory focuses on possible fluid dynamic conditions that may occur as fluid starts to flow across the valve seat. Valve seats do not have “knife edge” sealing surfaces; they have a width that distributes the stresses in the valve and seat. The stiction theory postulates that the flow of fluid from the smaller area at the center of the seat to the larger area around the seat causes a pressure drop across the seat. The pressure drop is the result of the radial divergence of the fluid and is strong enough to momentarily prevent the valve from opening. Extreme cases have even resulted in a circumferential ring of cavitation located at the center of the sealing areas of the seat and valve.

Field testing valves with special grooves and radial slots in the sealing face has proven successful in reducing stiction and opening valve lag in some cases. Narrowing the sealing faces may also prove effective, although part life may decrease due to increased stress and decreased wear area. Additional computer modeling and testing is required before the stiction theory can be completely validated.

As the valve continues to open, the spill area increases proportionally. It must be remembered that fluid is now flowing through the valve at an increasing rate while trying to fill the expanding cylinder volume created by the plunger withdrawal. Ideally, the motion of the valve will exactly correspond to the flow rate of the fluid through the valve, and the valve follows a smooth trajectory to a full open position (see Figure 9).

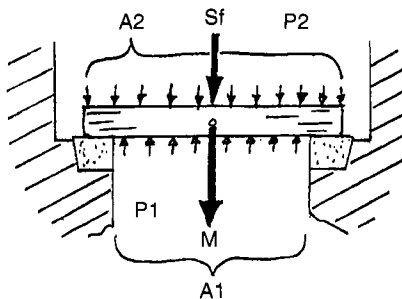
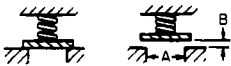
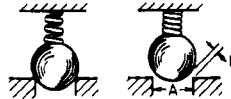
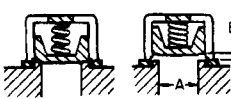
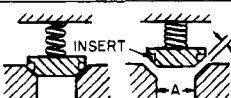
**FIGURE 8** Force balance of a typical valve

TABLE 10 Types of valves and their applications (Flowserve Corporation)

TYPE	SKETCH A = SEAT AREA B = SPILL AREA	PRESSURE PSI (BAR)	APPLICATION
PLATE		5,000 (345)	CLEAN FLUID. PLATE IS METAL OR PLASTIC
WING		10,000 (690)	CLEAN FLUIDS. CHEMICALS
BALL		30,000 (2,069)	FLUIDS WITH PARTICLES. CLEAR, CLEAN FLUID AT HIGH PRESSURE. BALL IS CHROME PLATED
PLUG		6,000 (414)	CHEMICALS
SLURRY		2,500 (172)	MUD, SLURRY. POT DIMENSIONS TO API-12. POLYURETHANE OR BUNA-N INSERT

This idyllic situation only occurs in very slow running pumps that are fitted with oversized valves. In modern higher speed pumps, the valve often accelerates at a rate fast enough to create a spill area greater than that required to maintain a constant flow velocity. At this point, the valve motion momentarily stops, and the valve may even start to close before the fluid flow once again matches the spill area. The valve then returns to the smooth trajectory until the maximum valve lift occurs.

After the plunger passes the mid point of its stroke, it starts decelerating. The flow of fluid through the valve is high enough to start to build pressure in the cylinder and unbalance the valve and start the valve closing. The valve spring has reached its maximum compression and, if properly designed, will help accelerate the valve back toward the seat. When the plunger reaches the end of the suction stroke, the valve should also be closing. The amount of crankshaft rotation that occurs between the time when the plunger reaches the end of its stroke and when the valve is completely closed is called the *delayed valve closing* and is normally 2 to 12° of crank rotation. Although a delayed valve opening may have little effect on overall pump performance, a delayed valve closing certainly will. A valve that is partially open when the plunger reverses direction will result in backflow. Backflow, as the name implies, is the reverse flow of fluid back across the valve and results in lower pump volumetric efficiency.

TABLE 11 Valve spill velocities

Valve	Spill velocity, ft/s (m/s)
Clean liquid suction valve	3–8 (0.9–2.4)
Clean liquid discharge valve	6–20 (1.8–6)
Slurry suction and discharge valve	6–12 (1.8–3.7)

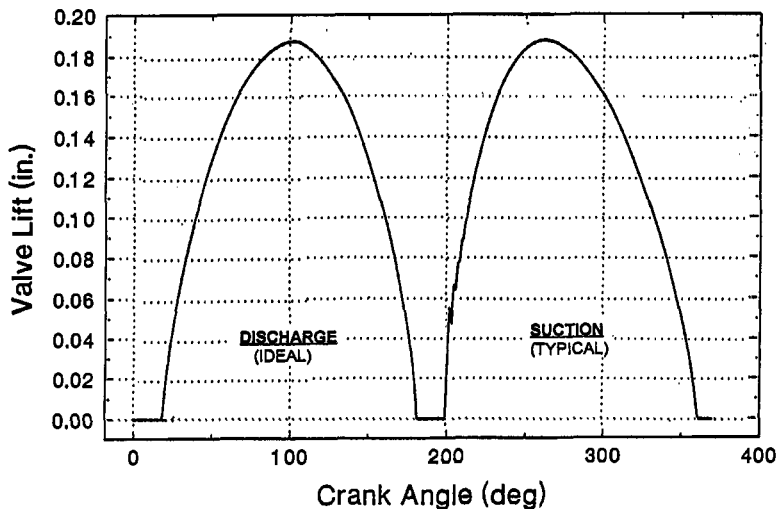


FIGURE 9 Valve lift versus crank angle (in $\times 2.54 =$ cm)

Many pump valves are designed with mechanical limits to the distance that the valve is allowed to open or lift. This is normally done to avoid overstressing the valve spring and to minimize the overall height of the valve assembly. Valve motion for a valve designed with a mechanical stop is shown in Figure 10. Another advantage of this design is that the valve does not have to travel as far during its closing cycle, compared to valves with no mechanical limits. The disadvantage is the potential for impact damage to the valve if it strikes the stop with excessive force.

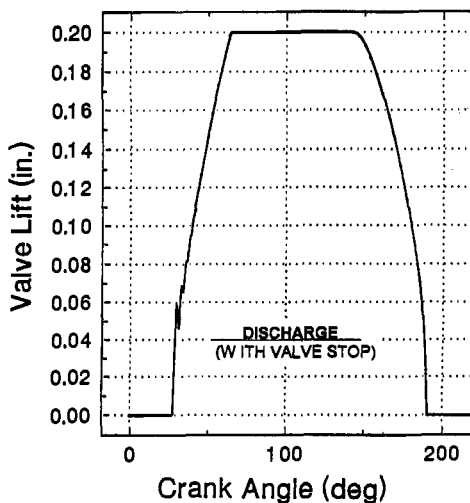


FIGURE 10 Valve lift versus crank angle for a valve with a mechanical stop (in $\times 2.54 =$ cm)

The importance of valve mass to overall valve dynamics has been the subject of much debate among pump designers over the years. There is no question that the mass of the valve must be overcome in order to open the valve. Test data are also available that shows no appreciable difference in the performance of pumps fitted with hollow ball valves versus identical pumps with solid ball valves. This apparent contradiction can be explained by studying valve acceleration at various parts of the valve cycle using advanced computer modeling.

It has already been explained above why a valve can hesitate, or stop opening, during the opening portion of its cycle. It's also been stated that when the valve spill area is large enough to establish a hydraulic balance, the valve will stop opening. What actually occurs is the valve starts decelerating as it comes closer to hydraulic equilibrium. If the valve is properly designed, it will contact its mechanical stop just as its acceleration/deceleration is very low. In that case, the valve mass has little significance on the impact force of the valve on the stop. The same holds true of the impact force of the valve on the seat as it closes. If the valve is properly designed, a 62-lb (28-kg) solid ball valve operating at 100 cycles per minute can strike the seat with less than 20 lb (9 kg) of force.

Although pump valves operate in a liquid medium, the shape of the valve is not an important design consideration for most applications. The relatively small distance that a valve travels is not sufficient for the fluid dynamics of a shape to have a measurable effect on valve dynamics. The only exception to this is valves in pumps handling high-viscosity liquids. For these applications, the ball valve, often spring-loaded, has proven to reduce closing valve lag and increase volumetric efficiency better than any other type of valve.

The most critical component in the optimization of valve dynamics is the valve spring. The pump designer normally selects a valve spring that will exert a certain amount of "pre-load" on the valve when it is closed. This pre-load helps the valve to close smoothly on the seat and avoid rebound (and possible backflow). Too high a preload in the suction valve may result in higher net positive suction head required (NPSHR). In the discharge valve, excessive preload can cause abnormally high pressure spikes in the fluid cylinder just before the valve opens.

The other valve spring design criterion is the spring rate. Every compression spring develops a predetermined resistance per unit length. This value is expressed in pounds per inch (kg per cm). As the valve is opening, the increasing spring force helps the valve obtain hydraulic balance faster. It also helps to limit the impact force of the valve on the stop. At the start of the closing cycle, the stored energy in the compressed spring helps the valve respond faster to the pressure changes in the fluid cylinder as the plunger starts to decelerate. Again, if the spring is properly designed, the closing valve lag will be minimized.

No published guidelines exist for the proper amount of valve spring pre-load or spring rate. Most pump designers use proprietary values, generated through a combination of in-house and field testing. However, although these values produce low NPSHR and high volumetric efficiency in most cases, the valve dynamics may not be close to being optimized. It should also come as no surprise that pumps fitted with "off-the-shelf" commercially available valves do not operate as well as pumps having optimized valve dynamics.

With the recent advent of advanced computer modeling of pump valves, it is now possible to optimize valve dynamics for a specific set of pump operating parameters. We may also see valve designs in the near future having variable rate valve springs, hydraulic dampening, and mechanisms that induce rotation as the valve opens and closes.

POWER END THEORY

The power end or drive end of a power pump consists of a crankshaft, connecting rods, crossheads, and bearings, all housed in a rigid structure referred to as the frame. Details of the design and construction of these components are covered in Section 3.2, "Power Pump Design and Construction." The slider-crank mechanism that converts rotational driving energy to the reciprocating motion that actuates the pistons or plungers can be found in reciprocating gas compressors, automotive engines, and stationary and marine engines. However, the stress loading pattern of power pump components is unique to this type of mechanism.

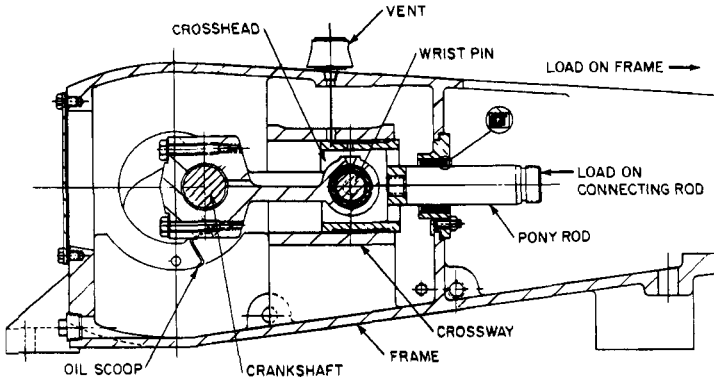


FIGURE 11 Power end, horizontal pump (Flowserve Corporation)

Applying Rod Load to the Power End As stated earlier, the loading on the power end is called the rod load and is the product of the area of the plunger multiplied by the maximum discharge pressure. In a gas compressor or engine, this loading increases over 90 to 180° of crank rotation before maximum loading is reached. In a power pump, the maximum loading is reached in less than 30° of crank rotation due to the relative incompressibility of the pumpage. Since this loading cycle is repeated with each plunger stroke, the actual loading resembles a shock load more than a simple cyclic fatigue load. The design criteria for the stressed components of the power end must therefore include the material's capability to absorb shock loads and overall safety factors greater than 3:1.

Another critical factor in the capability of the power-end components to handle the rod load is how the load is applied. Figures 11 and 12 show the direction of the load on the

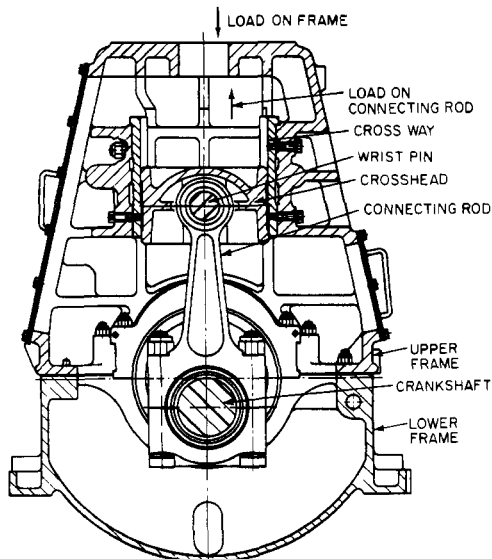


FIGURE 12 Power end, vertical pump (Flowserve Corporation)

frame and connecting rod for two different power end configurations. On vertical pumps with outboard packed stuffing boxes, the frame is in compression and the crosshead and connection rod are in tension. With horizontal, single-acting pumps, the frame is in tension, and the crosshead and connecting rod are in compression. The material selection and factors of safety must be altered accordingly.

Liquid Separation from the Plunger It has already been explained that fluid flows through the suction valve (and suction manifold and related piping) to fill the increasing cylinder volume caused by the withdrawal of the plunger. If the plunger accelerates faster than the flow of incoming liquid, the liquid will lose contact or separate from the plunger. The void that forms will be at a pressure lower than anywhere else in the cylinder. If the pumpage contains entrained gas, the gas may come out of a solution in this low-pressure area. The gas bubbles, when recompressed later in the plunger stroke, can cause cavitation damage to the plunger and surrounding cylinder.

The geometry of the slider-crank mechanism affects the point at which liquid separation occurs. As the ratio of the connecting rod length to the crank radius increases, the pump speed at which liquid separation occurs will decrease. Since liquid separation can be the determining factor in a pump's NPSHR, the pump designer must carefully evaluate the slider-crank geometry in order to optimize the pump's hydraulic performance. The pump speed at which water will separate from the end of the plunger can be calculated from the following:

$$\text{In USCS units:} \quad \text{rpm} = 54.5 \sqrt{\frac{(34 - h_s - h_f)A_s}{LR[l - (l/LR)]A_p}}$$

$$\text{In SI units:} \quad \text{rpm} = 16.6 \sqrt{\frac{(10.36 - h_s - h_f)A_s}{LR[l - (l/LR)]A_p}}$$

where h_s = suction head, ft (m)

h_f = piping frictional loss, ft (m)

A_s = area of suction pipe, in² (mm²)

L = length of connecting rod, centerline to centerline, ft (m)

R = crank radius, ft (m)

l = length of pipe where resistance to flow is to be measured, ft (m)

A_p = area of plunger, in² (mm²)

Unbalanced Forces Due to the relatively slow speed of a power pump, the inertia loads of the rotating/reciprocating parts are low enough to avoid the vibration problems associated with centrifugal pumps. For that reason, power pump crankshafts are not normally balanced. However, when the power pump is coupled to a drive containing a high-speed motor and gear reducer, a torsional analysis of the pump/drive unit may be required. For this analysis, the unbalanced forces of the rotating and reciprocating pump components can be calculated as follows:

Unbalanced Reciprocating Parts Force (F_{rec}) Parts are typically about one-third of the connecting rod weight (the crosshead, the crosshead bearing, the wrist pin, the pony rod, and the plunger). Additional parts on vertical pumps include the pull rods, yoke, and plunger nut.

$$F_{rec} = \frac{W}{g} \omega^2 R \left(\cos \theta + \frac{R}{L} \cos \theta \right)$$

where F_{rec} = reciprocating parts force, lb (N)

W = weight of all reciprocating or rotating parts, lb (N)

$$g = 32.2 \text{ ft/s}^2 (9.81 \text{ m/s}^2)$$

$$\omega = (2\pi/60) \times n \text{ (n = pump rpm)}$$

$$R = \text{one-half of stroke, ft (m)}$$

$$L = \text{length of connecting rod, centerline to centerline, ft (m)}$$

$$\theta = \text{crank angle; usually maximum force is at } \theta = 0^\circ, \cos \theta = 1$$

Unbalanced Rotating Parts Force (F_{rot}) Parts are typically about two-thirds of the connecting rod weight, crank end bearing, and crankpin, where the variables are as above.

$$F_{rot} = \frac{W}{g} \omega^2 R$$

Mechanical Efficiency The mechanical efficiency of a single-acting power pump without internal gears is typically 90 to 92%. Over half of the mechanical losses are due to the frictional drag of the plungers through the packing. The remaining losses are from the bearings, the crosshead-to-crosshead guide friction, and the extension rod-to-gland seal friction. If these components are properly lubricated, very few power-end design options are available that will produce a measurable increase in mechanical efficiency. Decreasing the diameter of the plungers and minimizing the number of packing rings will result in a small efficiency increase.

FURTHER READING

American National Standard for Reciprocating Power Pumps for Nomenclature, Definitions, Application and Operation, ANSI/HI 6.1-6.5-2000, Hydraulic Institute, Parsippany, NJ. www.pumps.org

Positive Displacement Pumps—Reciprocating, API Standard 674. 2nd Edition, 1994, The American Petroleum Institute, Washington, D.C. www.api.org

Henshaw, T. L. *Reciprocating Pumps*. Van Norstrand Reinhold Company, Inc., New York, 1987.

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SECTION 3.2

POWER PUMP DESIGN AND CONSTRUCTION

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The power pump, as a piece of rotating equipment, would seem to have a great deal in common with a reciprocating gas compressor or diesel or gas engine. All have crankshafts, bearings, connecting rods, and lubricating oil systems that use the pump frame as an oil reservoir. All have a fluid or gas end that contains relatively high pressure and includes some type of valving that enables the equipment to perform its required task.

Due to the unique operating characteristics of the power pump, very little design criteria or construction details can be directly transferred from these other mechanisms. Because of the high compressibility of most gases, the resulting cyclic loadings are applied to the various rotating, reciprocating, and pressure-retaining components in a smoother manner. Therefore, gas-driven or gas-pumping mechanisms can operate at speeds three or four times higher than a reciprocating pump with good reliability and low vibration levels.

It has been said that a power pump has more in common with a jackhammer because of the constant high-cycle shock loading that it must be designed to withstand. Therefore, very conservative material selections, stress limits, and factors of safety have been established by the pump designers for all continuous duty applications. Intermittent duty applications can be met using less stringent design criteria.

Seldom is a power pump designed for one specific application having one specific set of operating conditions. Consequently, few pump designs are truly optimized. The best economical approach has been to design a power end capable of accepting a number of fluid ends, which in turn allows the pump to operate over a wide range of hydraulic conditions. By using different plunger diameters for a specific operating pressure, the designer tries to "load up" the power to its maximum continuous rod load rating while maintaining the pump speed at a level just sufficient to provide the required flow rate.

A power pump has its cylinders operating in parallel with interconnecting suction and discharge manifolds. Additional cylinders are sometimes added to a basic power-end design in order to obtain higher flow rates. The most common examples of this approach are the quintuplex pumps that have an 80% component interchangeability with triplex

pumps. Triplex (three cylinder), quintuplex (five cylinder), septuplex (seven cylinder), and nonoplex (nine cylinder) pumps have been designed that have over a 90% interchangeability of parts. Only the power frame and crankshaft are unique to a specific pump size.

Factors of safety are used in power pump design and application. A connecting rod assembly, for instance, normally has a safety factor in excess of a 3:1 ratio because if a connecting rod fails in operation, it normally destroys many other power-end parts. A nodular iron frame will have a high safety factor also, primarily due to the requirement for thick wall sections to mount interfacing parts and to reduce operating noise levels. Some fluid-end expandable parts in the stuffing box and valves have safety factors below a 1.5:1 ratio because these parts are normally replaced before they would fail due to fatigue.

There has been a trend in past years in most industries to purchase power pumps that operate at higher and higher speeds. This has been driven by the economics of buying smaller, higher speed equipment versus the older, large, slower speed pumps. This trend seems to be losing some momentum, especially in critical applications. It remains for the pump purchaser and pump supplier to jointly evaluate each application and try to arrive at a pump selection that will provide the most reliable performance for the lowest installed cost and operating cost.

LIQUID END

The liquid end consists of the cylinder, the plunger or piston, the valves, the stuffing box, the manifolds, and the access covers. Figure 1 shows the liquid end of a horizontal pump, and Figure 2 shows the same for a vertical pump.

Cylinder (Working Barrel) The cylinder is the body where the pump pressure to overcome discharge pressure is developed. It is continuously under fatigue. Cylinders on many horizontal pumps have suction and discharge manifolds integral with the cylinder. Vertical pumps have separate manifolds.

A cylinder containing the passages for more than one plunger is referred to as a *single cylinder*. When the cylinder is used for one plunger, it is called an *individual cylinder*. Individual cylinders are used where developed stresses or replacement costs are high.

Cast cylinders are usually limited to the developed pressures listed in Table 1. Forged cylinders are made from 1020 and 4140 carbon steel; 304L, 316L, 17-4 PH, and 15-5 PH stainless steels; and nickel aluminum bronze. In recent years, duplex stainless steels have seen an increase in usage when higher strength and corrosion resistance is required.

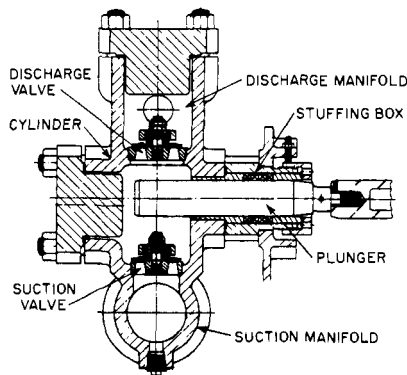


FIGURE 1 A liquid-end horizontal pump (Flowserve Corporation)

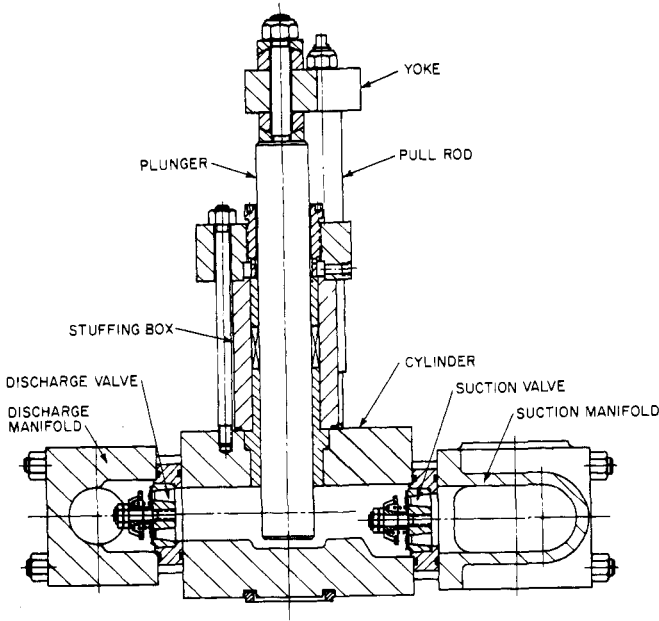


FIGURE 2 A liquid-end vertical pump (Flowserve Corporation)

Forged cylinders have a 4:1 to 6:1 forging reduction on all sides of the cylinder to obtain a homogeneous internal structure. This type of cylinder requires heat treatment after forging to eliminate the stresses that occur during the forming operation. The pump designer must minimize the section thickness of the cylinder or residual stresses may remain after machining the internal bores.

The highest stresses occur at the intersection of the horizontal and vertical internal bores. Unlike a simple cylinder where the internal stresses are analyzed as a single hoop stress, the stress at the intersecting bores of a power pump cylinder are based on a double hoop stress. In Figures 3 and 4, the distances b and d are the internal diameters, a and c are the diameters to the nearest obstruction of the solid material of the cylinder, P is the developed pressure, and S is the resulting stress. Stress concentration factors at the intersecting bores can be omitted if the radius is not less than 0.25 in (1.5 mm). Above 3000 lb/in² (207 bar), the internal bores should have a surface finish of 63 rms minimum.

During each plunger cycle, the developed pressure goes from suction pressure to discharge pressure and back to suction. For a pump operating at 360 strokes per minute, over six million fatigue cycles will occur in less than 12 days. For this reason, very conservative allowable stress limits are used. Normally, these limits range from 10,000 to 25,000 lb/in² (69 to 172 Mpa), depending on the material of the cylinder, duty cycle, and liquid being

TABLE 1 Maximum pressure in cast cylinders

Material	lb/in ² (bar)
Cast iron	2000 (137)
Aluminum bronze	2500 to 3000 (172 to 207)
Steel	3000 (207)
Ductile iron	3000 (207)

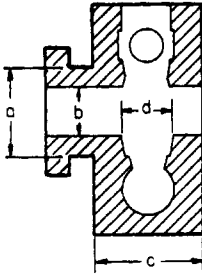
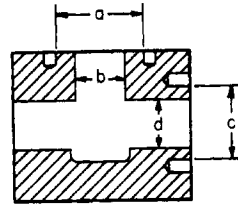


FIGURE 3 Stress dimensions for a horizontal liquid end (Flowserve Corporation)



$$S = P \left[\frac{a^2 + b^2}{a^2 - b^2} \right] + \left(\frac{c^2 + d^2}{c^2 - d^2} \right)$$

FIGURE 4 Stress dimensions for a vertical liquid end (Flowserve Corporation)

pumped. The allowable stress is a function of the fatigue stress of the material for the liquid being pumped and the life cycles required.

Due to improper suction system design, misapplications, or process upsets, instantaneous pressure in the cylinder may be much higher than design pressure. When the liquid contains entrained gas that can be released because of inadequate suction pressure, the resulting cavitation can cause instantaneous pressure four to five times the design pressure. This results in reducing the life of the cylinder and other liquid-end components, as well as damaging pressure pulsations in the suction and discharge piping systems.

For continuous duty pumps operating above 10,000 lb/in² (690 bar) of discharge pressure, special liquid-end designs have been developed. In Figure 5, the intersecting bores have been eliminated by arranging the suction and discharge valves on the same axis as the plunger. This means the stresses in the cylinder will be only half of those in a comparable T-block design.

Plungers The plunger transmits the force that develops the pressure. It is normally a solid construction of up to 5 in (127 mm) in diameter. Above that dimension, it may be made hollow to reduce its weight. Small-diameter plungers used for 6000 psi (414 bar) and above should be reviewed for possible buckling. Plunger speeds range from 150 to 350 fpm (46 to 107 m/s). The surface finish normally is between 14 and 20 rms. A finish below 8 rms should be avoided because excessive packing leakage may occur due to the inability of the packing to seal properly on the smooth surface.

Although some plungers are made of heat-treated or case-hardened steel, the most common are the hard-coated or solid ceramic. The hard coatings are normally flame sprayed powders of Colmonoy or tungsten carbide, or ceramic oxides. These can be applied over base materials of 1020 carbon steel or 316L stainless steel. Ceramic oxide is normally limited to 200°F (93°C) and is used for soft water, crude oil, mild acids, and mild alkalis.

The porosity and bond strength of coatings must be carefully evaluated for use with higher operating pressures. Under those conditions, the liquid may penetrate the pores of the coating and lift the coating off the base material. Ceramic plungers also have special requirements and limitations. This type of plunger is often constructed as a solid bar, closed-end tube of ceramic, bonded to a metal end cap or plug. A vent must be provided to allow the pressure inside the plunger to equal the atmospheric pressure or the ceramic-to-metal bond may fail or the plunger may explode. In addition to the normally fragile nature of any ceramic plunger, the solid ceramic plunger is susceptible to failure due to thermal shock.

Pistons Pistons are used for water pressures up to 2000 lb/in² (138 bar). For higher pressures, a plunger is usually used. Pistons are cast iron, bronze, or steel with reinforced elastomer sealing rings (see Figure 6). They are most frequently used in duplex, double-acting pumps. The latest trend is to use pistons in single-acting triplex pumps.



FIGURE 5 Special high-pressure liquid-end designs arrange the suction and discharge valves on the same axis as the plunger, thus eliminating intersecting bores (Flowserve Corporation).

Stuffing Box The stuffing box assembly consists of a stuffing box, upper and lower bushings, packing, and a gland. For ease of maintenance, the stuffing box assembly is usually removable (see Figure 7).

The stuffing box bore is machined to a 63 rms finish to ensure packing sealing and life. A single hoop stress equation is used to determine the wall thickness with an allowable stress limit of 10,000 to 20,000 lb/in² (69 to 138 MPa).

The bushings that guide the plunger have a 63 rms finish with approximately 0.001 to 0.002 in (0.02 to 0.05 mm) of diametrical clearance per inch of the plunger diameter. The

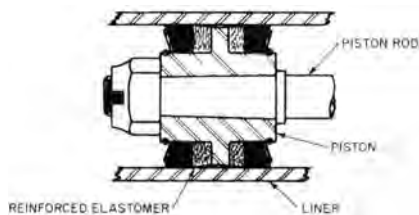


FIGURE 6 Elastomer face piston (FWI)

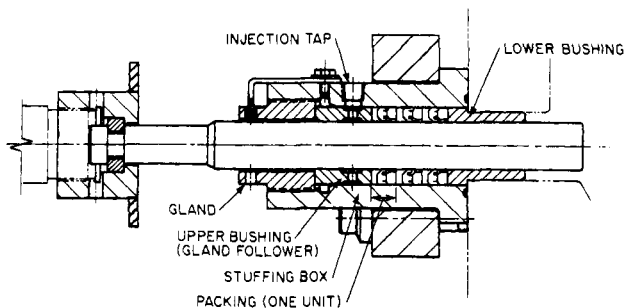


FIGURE 7 A stuffing box (Flowserve Corporation)

lower bushing is sometimes secured in an axial position to prevent movement of the packing. Bushings are made of bearing bronze, Ni-resist, or cast iron. Stainless steel can be used if its antigalling properties with the plunger coating can be verified.

Packing is either square cross-section, woven construction, or molded V or chevron-shaped (see Figure 8). Some types use metal backup adapters. A packing set consists of top and bottom adapters and one or more packing sealing rings. A stuffing box can use two to five rings of packing, depending on the pressure and the fluid being pumped. Packing rings are usually made up of a number of composite materials selected for their strength, wear resistance, and lubricity, including neoprene, Teflon, cotton duct, and Kevlar.

Packing can be made self-adjusting by installing a spring between the bottom of the packing set and the lower bushing or bottom of the stuffing box. This arrangement eliminates overtightening the packing and enables a more uniform break-in. Packing can be lubricated through a grease fitting or by an auxiliary lubricator driven through a take-off on the crankshaft.

For chemical or slurry service, a lower injection ring is used for flushing. This prevents concentrated pumped fluid from impinging directly on the packing. This injection can be continuous or synchronized to inject only on the suction stroke. Flush glands at the out-board end of the stuffing box are employed when a toxic vapor is present or when the leakage may flash.

The stuffing box for the piston rod of a double-acting pump is similar in construction to that of a stuffing box for a plunger. The primary difference is that it must seal against pressure being developed while the rod moves back through the packing. Single-acting pistons do not employ a stuffing box. Leakage past the piston-sealing rings goes into the frame extension to mix with the continuously circulating lubricant.

Cylinder Liner The cylinder wear liner (refer to Figure 6 and see Figure 9) is usually of Ni-resist material. Its length is slightly longer than the stroke of the pump to enable an assembly entrance taper of the piston into the liner. In double-acting pumps, the liner has packing to prevent leakage from the high-pressure side to the low pressure side of the

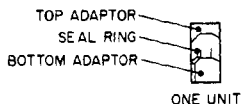


FIGURE 8 Chevron packing (Flowserve Corporation)

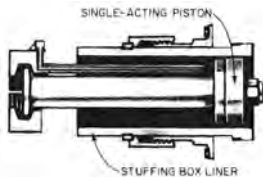


FIGURE 9 Single-acting piston stuffing box (Flowserve Corporation)

cylinder. Because of the brittleness of the liner, the construction should be such that the liner is not compressed. The finish of the bore of the liner is approximately 16 rms.

Valves Many types of valve designs exist. Which type is used depends on the application. The main parts of a valve assembly are the seat and sealing member, usually a disc, ball, or plate. The plate movement is controlled by a spring or retainer. The seat usually uses a taper where it fits into the cylinder or manifold. The taper not only gives a positive fit but permits easy replacement of the seat.

Some pumps use the same size suction and discharge valve for interchangeability (refer to Figure 2). Some use larger suction valves than discharge valves for improved NPSH reasons. Others use larger discharge valves than suction valves because the cylinder configuration requires the suction valve to be removed through the location of the discharge valve (refer to Figure 1). Because of space considerations, valves are sometimes used in clusters on each side of the plunger to obtain the required total valve area that reduces valve velocity.

Table 2 shows seat and plate hardness for some valve materials. Seats and plates made of 316 stainless steel material are usually chrome-plated or flame-sprayed to give them the desired surface hardness.

The flow area of the valve must be large enough to prevent significant pressure drop or restriction to flow. Normally, suction valves are sized for 6 to 8 ft/s velocity, and the discharge valves and sized for 8 to 12 ft/s.

The valve springs must be made of corrosion-resistant material and designed to withstand high-cycle fatigue stresses. Under no situation should the valve operation enable the spring to go to a solid stack height or wire-to-wire condition. The ends of the spring must be ground flat and square with a maximum perpendicularity of 3 degrees.

Manifolds These are the chambers where the liquid is disbursed or collected for distribution before or after passing through the cylinder. In horizontal pumps and some vertical pumps, the manifolds are cast or machined integral with the fluid cylinder. Most vertical pumps have the suction and discharge manifolds separate from the cylinder (refer to Figure 2).

Suction manifolds are designed to eliminate air pockets from the flange to the valve entrance (see Figure 10). Separate suction manifolds are cast iron, cast bronze, or fabricated steel. Discharge manifolds are steel forgings or fabricated steel. The manifolds have a minimum deflection to prevent gasket shifts when subjected to maximum operating pressures.

The velocity through the manifolds of a clean liquid is 3 to 5 ft/s (0.9 to 1.5 m/s) at the suction and 6 to 16 ft/s (1.8 to 4.9 m/s) at the discharge. Suction and discharge manifold velocities in a slurry service are 6 to 10 ft/s (1.8 to 3 m/s). Slurry services have a minimum velocity of 6 ft/s (1.8 m/s) to prevent the heavier slurry particles from falling out of solution.

In USCS units, $V \text{ (ft/s)} = \text{pump gpm} \times 0.321 / \text{cross-sectional area of the manifold, in}^2$

In SI units, $V \text{ (m/s)} = \text{pump m}^3/\text{h} \times 277.8 / \text{cross-sectional area of the manifold, mm}^2$

TABLE 2 Recommended material hardness for valve plate and seat

Material	Plate	Seat
Rockwell C hardness		
329	30 to 35	38 to 43
440	44 to 48	52 to 56
17-4 PH	35 to 40	40 to 45
15-5 PH	35 to 40	40 to 45
Brinell hardness		
316	150 to 180	150 to 180

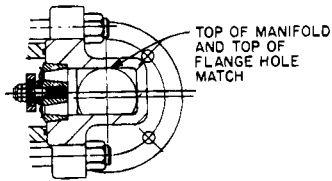


FIGURE 10 A suction manifold construction to eliminate air pockets (Flowserve Corporation)

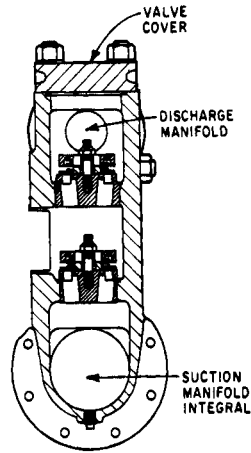


FIGURE 11 An integral suction and discharge manifold (Gardner-Denver)

A water hammer creates an additional pressure (which is added to the rated pump pressure), as does hydraulic shock loading. The discharge manifold rating is then made equal to or greater than the sum of these pressures.

Valve Covers Valve covers are used to provide accessibility to the valves without disturbing the cylinder, manifold, or related piping (see Figures 11 and 12).

Plunger Covers and/or Cylinder Heads These are used on horizontal pumps to provide accessibility to the plunger, piston, and cylinder liner (refer to Figures 11 and 12 and see Figure 13).

POWER END

The power end contains the crankshaft, connecting rods, crossheads, pony rod, bearings, and frame (see Figures 14 and 15). Basic designs are horizontal and vertical with sleeve or antifriction bearings. Some have integral drive gears.

Frame The frame absorbs the plunger load and torque. Vertical pumps with outboard packed stuffing boxes (refer to Figure 2) have the frame in compression (refer to Figure 14). With horizontal single-acting pumps, the frame is in tension (refer to Figure 15). Frames are usually close-grain cast iron with relatively thick cross-sections. This combination provides superior dampening for the normal low-frequency cyclic loading. Some intermittent duty pumps, specifically designed for mobile services, are constructed of lighter weight, fabricated steel plate.

The frame is usually vented to the atmosphere. However, when the atmosphere is detrimental to the working parts of the frame, such as an ammonia attack on bronze bearings, the frame can be purged continuously with nitrogen gas.

Crankshaft The crankshaft (see Figures 16, 17, and 18) varies in construction, depending on the design and power output of the pump. In horizontal pumps, the crankshafts are usually of nodular iron or cast steel. Vertical pumps use forged steel or machined billet crankshafts. Because the crankshafts have a low mass and operate at relatively low

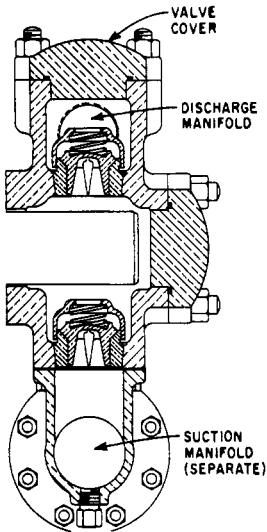


FIGURE 12 Separate suction with an integral discharge manifold (Gardner-Denver)

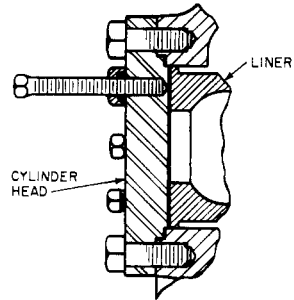


FIGURE 13 A plunger cover (FWI)

speeds, counterweights and flywheels are not used. Except for duplex pumps, the crankshaft always has an odd number of throws to obtain the best flow pulsation characteristics. The firing order in a revolution depends on the number of throws on the crankshaft (see Table 3).

The crankshaft main and rod-bearing surfaces are ground to a 16 rms surface finish or better. Large radii are used where journal diameters intersect with the adjacent cheeks to

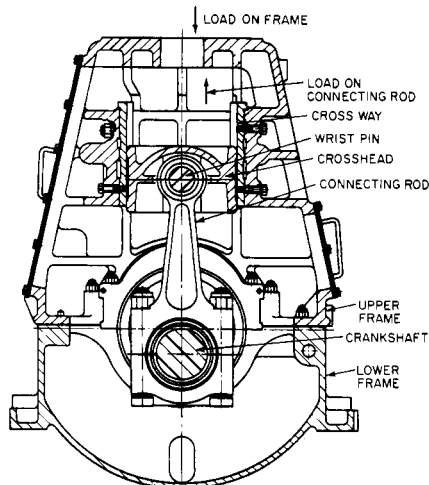


FIGURE 14 A vertical pump power end (Flowsolve Corporation)

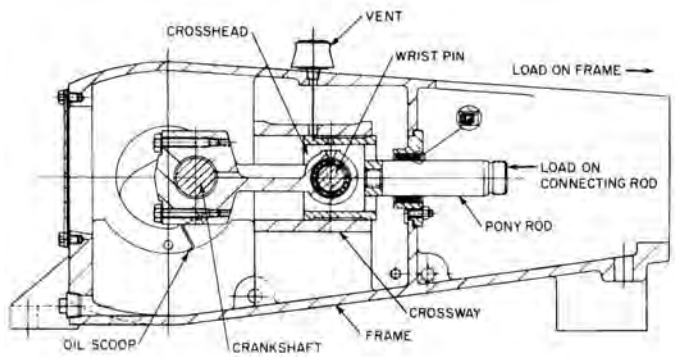


FIGURE 15 A horizontal pump power end (Flowserve Corporation)

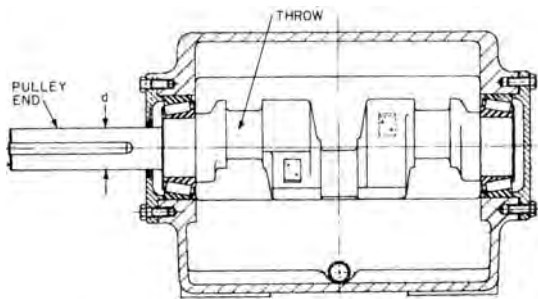


FIGURE 16 A cast crankshaft (Flowserve Corporation)



FIGURE 17 A crankshaft machined from a billet (Flowserve Corporation)



FIGURE 18 A cast crankshaft with integral gear (Continental-EMSCO)

TABLE 3 Order of Pressure Build-up or Firing Order

Throw from pulley end	No. of plungers or pistons	Pressure build-up order								
		1	2	3	4	5	6	7	8	9
Duplex	2	1	2							
Triplex	3	1	3	2						
Quintaplex	5	1	3	5	2	4				
Septuplex	7	1	4	7	3	6	2	5		
Nonuplex	9	1	5	9	4	8	3	7	2	6

minimize stress concentrations in these fatigue-prone areas. Cross-drilling of the main to rod journals is often provided to furnish lubrication to the crankpin bearings.

Many horizontal pumps under 250 hp (186 kW) are driven by means of a sheave or pulley and belt arrangement. Increasing the diameter of the pulley for a given horsepower will increase the stress in the crankshaft. The maximum allowable crankshaft stress will often limit the maximum diameter of the pulley that can be used. Stress levels of 2000 to 3000 lb/in² (14 to 20.7 MPa) are common, where d is the pulley end diameter of the crankshaft in inches (mm) and n is the pump speed in revolutions per minute.

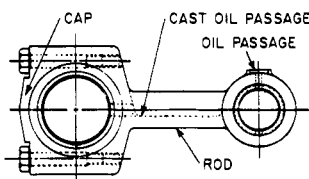
Connecting Rods and Eccentric Straps The connecting rods (see Figure 19) transfer the rotating force of the crank pin to an oscillating force on the wrist pin. Most connecting rods are split perpendicular to their centerline at the crankpin end for ease of assembly onto the crankshaft.

The cap and rod are aligned with close-tolerance bushings or body-bound bolts. The rods either are rifle-drilled or have cast passages to transfer oil from the wrist pin to the crank pin. A connecting rod with a tension load is made of forged steel, cast steel, or fabricated steel. Rods with a compressive loading are cast steel, nodular iron, or aluminum alloy. The connecting rod finish, where the bearings are mounted, is 32 to 63 rms.

The ratio of the distance between the centerlines of the wrist pin and crank pin bearings to half the length of the stroke is referred to as the L/R . The ratio directly affects the pressure pulsations, volumetric efficiency, size of the pulsation dampener, speed of liquid separation, acceleration head, moments of inertia forces, and size of the frame. Low L/R results in higher fluid pulsations. A high L/R reduces pulsations but may result in a large and uneconomical power frame. The common industrial L/R range is 4:1 to 6:1.

The eccentric strap (see Figure 20) performs the same function as a connecting rod, except that the former usually is not split. The eccentric strap is furnished with rolling element bearings, whereas connecting rods are furnished with sleeve bearings. Eccentric straps are applied to mud pumps and some slurry pumps, which are started against a full load without requiring a bypass line.

Wrist Pin Located in the crosshead, the wrist pin transforms the oscillating motion of the connecting rod to a reciprocating motion. The maximum stress in the wrist pin from deflection should not exceed 10,000 lb/in² (69 MPa):

**FIGURE 19** The connecting rod (Flowserve Corporation)

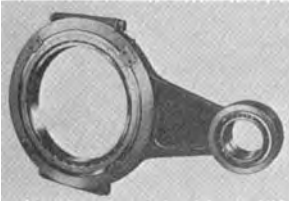


FIGURE 20 An eccentric strap (Gardner-Denver)

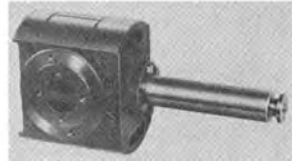


FIGURE 21 A crosshead (Gardner-Denver)

In USCS units,

$$S = \frac{PL \times \ell}{8 \times 0.098d^3}$$

In SI units,

$$S = 1.25 \frac{PL \times \ell}{d^3}$$

where PL = plunger load, lb (N)

ℓ = length under load, in (mm)

d = diameter of pin, in (mm)

Large pins are hollow to reduce the oscillating mass and assembly weight. Depending on the design, pins can have a straight fit, taper fit, or loose fit in the crosshead. The pin is case-hardened and has a surface finish of 16 rms or better. When needle or roller bearings are used for the wrist pin bearing, the pin serves as the inner race.

Crosshead The crosshead (see Figure 21) moves in a reciprocating motion and transfers the plunger load to the wrist pin. The crosshead is designed to absorb the side or radial load from the plunger as the crosshead moves linearly on the crossway. The side load is approximately 25% of the plunger load. For cast-iron crossheads, the allowable bearing load is 80 to 125 lb/in² (551 to 862 kPa). Crossheads are grooved for oil lubrication and have a 63 rms surface finish. Crossheads are piston types (full round) or partial-contact types. The piston type should be open or vented to prevent air compression at the end of the stroke.

In vertical pumps, the pull rods, or side rods, go through the crosshead so that the crosshead is under compression when the plunger load is applied (refer to Figure 14).

Crosshead Guides (Crossways) The crossway (see Figure 22) is the surface on which the crosshead reciprocates. In horizontal pumps, it is cast integral with the frame (refer to Figure 15). In large frames, it is usually replaceable and is shimmed to effect proper running clearances (refer to Figures 14 and 22). The crossway finish is 63 rms.

Pony Rod (Intermediate Rod or Extension Rod) The pony rod is an extension of the crosshead on horizontal pumps (refer to Figures 21 and 22). It is screwed or bolted to the



FIGURE 22 Crossways (Gardner-Denver)

crosshead and extends through the frame (refer to Figure 15). A seal on the frame and against the pony rod prevents oil from leaking out of the frame. A baffle is attached to the plunger end of the pony rod to prevent stuffing box leakage from entering the frame.

Pull Rod (Side Rod) In vertical pumps, two pull rods are connected to the crosshead. The rods are secured by a shoulder and nut so that the cast iron is in compression when the plunger load (rod load) is applied. The rods extend out through the top of the frame and fasten to a yoke (upper crosshead). The plunger is attached to the middle of the yoke with an aligning feature for the plunger (refer to Figure 2). The nuts fastening the pull rod to the yoke and crosshead must be torqued to a high enough level to prevent the bolted joint from seeing cyclic loading.

Bearings Both sleeve and rolling element bearings are used in power pumps. Some power ends use all sleeve, others use all rolling element, and others use a combination of both.

SLEEVE BEARINGS

When properly installed and lubricated, sleeve bearings are considered to have an infinite life. They are inexpensive and can usually be replaced without special tools and with minimum pump disassembly. The sleeve bearing is designed to operate within a certain speed range, and too high or too low a speed will upset the oil film lubrication. With standard sleeve-bearing designs, pumps cannot be operated at speeds below 20 rpm with the plunger fully loaded using standard lubrication. Below this speed, the oil film in the wrist pin bearing starts to break down and the increased friction causes excessive heat and leads to bearing failures. Recent developments of improved bearing materials, along with special grooving and other aids to lubrication, now allow for fully loaded pump operations at speeds as low as 10 rpm. The finish on the sleeve bearing is 16 rms. Clearances are approximately 0.001 in per in (0.001 mm per mm) of the diameter of the bearing.

Wrist Pin Bearings These bearings are normally the highest loaded bearings in the pump. They can only perform an oscillating motion and therefore cannot develop the dynamic oil film common to fully rotational bearings. In single-acting pumps with low suction pressure, adequate reversals in loading take place on the bearing to permit replenishment of the oil film. High suction pressure in a horizontal pump increases the reverse loading. In vertical pumps, high suction pressure can produce a condition of no reverse loading, and in this case, special bearing materials, grooving, and pump derating may be required. Increasing oil pressure by itself will not solve this problem. Allowable projected area loading is 1200 to 1500 lb/in² (8.27 to 10.3 MPa) with bearing bronze.

Crank Pin Bearings The crank pin bearing is a rotating split bearing and develops a better oil film than the wrist pin bearing. It is clamped between the connecting rod and cap. The bearing is a bronze-backed babbitt metal or a tri-metal automotive type bearing. The allowable projected area load is 1200 to 1600 lb/in² (8.27 to 11.08 MPa).

Main Bearings The main bearings absorb the plunger load and gear load. The total plunger load varies during each revolution of the crankshaft. The triplex main bearings receive the greatest variations in loading because the crankshaft has the greatest relative span between bearings. Sleeve bearings are flanged to locate them in an axial position, and the flange absorbs residual axial thrust. In long-stroke vertical pumps, a main bearing can be found between every connecting rod. Split bearings are a bronze-backed babbitt metal with an allowable loaded area of 750 lb/in² (5.1 MPa).

ROLLING ELEMENT BEARINGS

A pump with rolling element bearings can be started under a full plunger load without a bypass line. Rolling element bearings enable the pump to operate continuously at low speeds with full plunger loads. They are selected for 30,000 to 50,000 hours of L_{10h} life. Slurry pipeline applications may require 100,000 hours. Eccentric straps are used in place of connecting rods.

Wrist Pin Bearings These are needle or roller bearings. The outer race is a tight fit into the strap, and the wrist pin is used as the inner race. This reduces the size of the bearing and strap. The wrist pin is held in the crosshead with a taper or keeper plate (see Figure 23).

Main Bearing The main bearings used in conjunction with sleeve wrist pin bearings and sleeve crank pin bearings are usually tapered roller bearings, as is the case with a horizontal triplex plunger pump. The main bearings are usually mounted directly into the frame (see Figure 24).

The main bearings used with full rolling element bearing designs are self-aligning spherical roller bearings. These bearings compensate for axial and radial movements of the crankshaft and are usually mounted in bearing holders, which in turn are mounted on the frame (refer to Figure 23). This type of design is mostly used in mud pumps and in some slurry pumps.

LUBRICATION

Non-detergent SAE 30 to 50 oil is used for bearing lubrication. In splash lubrication, the cheeks of the crankshaft or oil scoops (refer to Figure 15) throw oil by centrifugal force against the frame wall. This oil is then distributed by gravity to the crosshead, wrist pin, and crank pin. At low speeds, there is not enough centrifugal force for proper oil distribution, so a partial force feed is then employed to maintain proper lubrication. Force-feed lubrication requires 0.5 to 1 gpm (1.9 to 3.8 l/m) per bearing with an oil pressure of 28 to 40 lb/in² (1.7 to 2.8 bar).

APPLICATIONS

Power pumps are used in a variety of applications where high-pressure and low-flow requirements exist. They are also widely used when a high-volumetric efficiency is

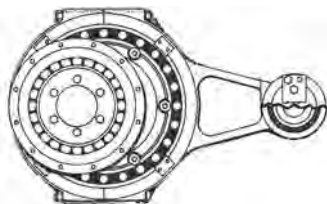


FIGURE 23 The main bearing mounted in a bearing holder (Gardner-Denver)

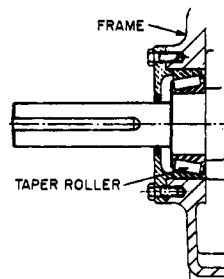


FIGURE 24 The main bearing mounted directly in the frame (Flowserve Corporation)

required. For these applications, other types of pumps have design or operating deficiencies that make their lower initial cost or lower maintenance costs less attractive.

Some of the common applications for power pumps include the following:

Ammonia service	Liquid petroleum gas
Carbamate service	Liquid pipeline
Chemicals	Power oil
Crude-oil pipeline	Power press
Cryogenic service	Salt-water injection for water flood
Fertilizer plants	Slurry pipeline (up to 70% by weight)
High-pressure water cutting	Slush ash service
Hydro forming	Steel mill descaling
Hydrostatic testing	Water blast service

The following applications may require power pumps with special modifications, and the exact requirements should be reviewed in detail with the pump supplier:

Cryogenic service	Low-speed operation
Highly compressible liquids	Slurry pipeline
Liquids over 250°F (121°C)	Special fluid-end materials
Liquids with entrained gas	Viscosities over 500 SSU

Duty Service To make an economical pump selection, the type of full-load service that you need should be considered. For instance, a small, lighter weight pump operating at higher speeds may be suitable for an intermittent duty application such as hydrostatic testing. The different types of duty services are as follows:

- *Continuous*: 8 to 24 hours per day
- *Light*: 3 to 8 hours per day
- *Intermittent*: Up to 3 hours per day
- *Cyclic*: 30 seconds loaded out of every 3 minutes

RELIEF VALVES

Relief valves should be rated for the full flow of the pump. The *set pressure* or *cracking pressure* must be sufficiently above the operating pressure to prevent normal pressure pulsations from activating the relief valve. The following values can be used as guidelines, but the final selection should be made only in agreement with the pump supplier and system designer:

- Duplex double-acting: 25%
- Triplex and above: 10%

Bypass Relief Starting a power pump under a full load requires a high starting torque. Also, a pump with sleeve-bearing construction when starting under a full load may not have adequate bearing lubrication to reach the operating speed.

The starting torque on the driver and plunger load on the bearings may be reduced by performing one of two steps:

- Install a bypass line from the discharge back to the source or to a drain (see Figure 25). This line is located between the pump discharge and a check valve before the discharge piping system. The bypass valve is operated manually or automatically. The bypass line

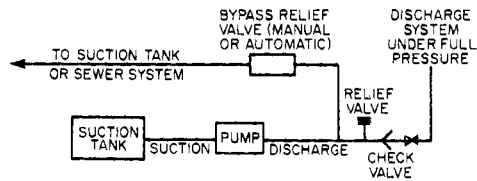


FIGURE 25 A bypass relief system (Flowserve Corporation)

reduces the starting torque from the mechanical losses and the liquid inertia of the suction and bypass systems. After the pump is up to speed, the bypass is closed and the pump goes into normal operation.

- Use suction valve unloaders. The unloader is a mechanism that mechanically holds the suction valves off of their seats before and during startup, which stops the pumping action. Only mechanical losses must be overcome. After the pump is up to speed, the suction valves are closed in unison. In large-flow pumps, an electronic distributor is used to synchronize actuation of the suction valves with the plunger pressure build-up order (or firing order).

PULSATION DAMPENERS

The performance of a power pump, or its interaction with the suction and discharge piping systems, can often be improved by adding suction and discharge dampeners. Dampeners are used to reduce suction and discharge pressure pulsations. A properly sized and located suction dampener can reduce the system pulsations to an equivalent pipe length of 5 to 15 pipe diameters. If the dampener is the flow-through type, it should be located on the same side as the pipe, not at the dead end of the manifold.

The most commonly used dampeners are the gas-bladder types. These units are normally charged with nitrogen gas at 50 to 66% of the system pressure. Although these units are relatively inexpensive, they can attenuate up to 90% of the pressure pulsations, below 50 Hz, in the interconnecting piping systems.

SECTION 3.3

STEAM PUMPS

ROBERT M. FREEBOROUGH

BASIC THEORY

A reciprocating positive displacement pump is one in which a plunger or piston displaces a given volume of fluid for each stroke. The basic principle of a reciprocating pump is that a solid will displace an equal volume of liquid. For example, when an ice cube is dropped into a glass of water, the volume of water that spills out of the glass is equal to the submerged volume of the ice cube.

In Figure 1, a cylindrical solid, a plunger, has displaced its volume from the large container to the small container. The volume of the displaced fluid (B) is equal to the plunger volume (A). The volume of the displaced fluid equals the product of the cross-sectional area of the plunger and the depth of submergence.

All reciprocating pumps have a fluid-handling portion, commonly called the *liquid end*, that has

1. A displacing solid called a *plunger or piston*
2. A container to hold the liquid, called the *liquid cylinder*
3. A suction check valve to admit fluid from the suction pipe into the liquid cylinder
4. A discharge check valve to admit flow from the liquid cylinder into the discharge pipe
5. Packing to seal the joint between the plunger and the liquid cylinder tightly to prevent liquid from leaking out of the cylinder and air from leaking into the cylinder

These basic components are identified on the rudimentary liquid cylinder illustrated in Figure 2. To pump the liquid through the liquid end, the plunger must be moved. When the plunger is moved out of the liquid cylinder as shown in Figure 2, the pressure of the fluid in the cylinder is reduced. When the pressure becomes less than that in the suction pipe, the suction check valve opens and liquid flows into the cylinder to fill the volume being

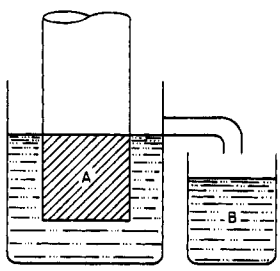


FIGURE 1 The volume of liquid displaced by a solid equals the volume of the solid

vacated by withdrawal of the plunger. During this phase of operation, the discharge check valve is held closed by the higher pressure in the discharged pipe. This portion of the pumping action of a reciprocating positive displacement pump is called the *suction stroke*.

The withdrawal movement must be stopped before the end of the plunger gets to the packing. The plunger movement is then reversed, and the *discharge stroke* portion of the pumping action is started, as illustrated in Figure 3.

Movement of the plunger into the cylinder causes an increase in pressure of the liquid contained therein. This pressure immediately becomes higher than suction pipe pressure and causes the suction valve to close. With further plunger movement, the liquid pressure continues to rise. When the liquid pressure in the cylinder reaches that in the discharge pipe, the discharge check valve is forced open and liquid flows into the discharge pipe. The volume forced into the discharge pipe is equal to the plunger displacement less very small losses. The plunger displacement is the product of its cross-sectional area and the length of stroke. The plunger must be stopped before it hits the bottom of the cylinder. The motion is then reversed, and the plunger again goes on suction stroke as previously described.

The pumping cycle just described is that of a *single-acting* reciprocating pump. It is called single-acting because it makes only one suction stroke and only one discharge stroke in one reciprocating cycle.

Many reciprocating pumps are *double-acting*; that is, they make two suction and two discharge strokes for one complete reciprocating cycle (Figure 4). Most double-acting pumps use as the displacing solid a piston that is sealed to a bore in the liquid cylinder or

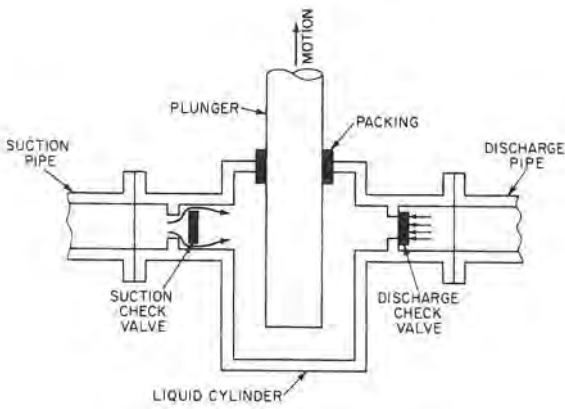


FIGURE 2 Schematic of the liquid end of reciprocating pump during the suction stroke

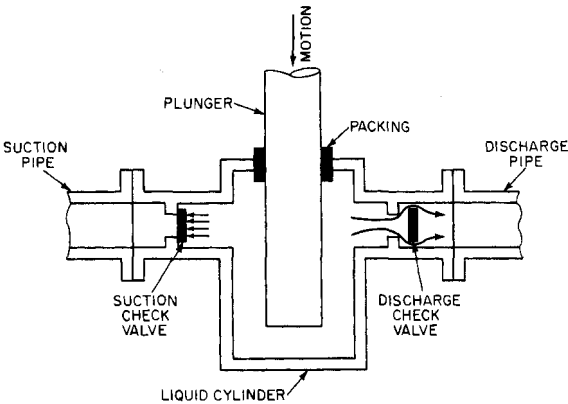


FIGURE 3 Schematic of the liquid end of a reciprocating pump during the discharge stroke

to a liquid cylinder liner by pistons with packing. It has two suction and two discharge valves, one of each on each side of the piston. The piston is moved by a piston rod. The piston rod packing prevents liquid from leaking out of the cylinder. When the piston rod and piston are moved in the direction shown, the right side of the piston is on a *discharge stroke* and the left side of the piston is simultaneously on a *suction stroke*. The piston packing must seal tightly to the cylinder liner to prevent leakage of liquid from the high-pressure right side to the low-pressure left side.

The piston must be stopped before it hits the right side of the cylinder. The motion of the piston is then reversed so the left side of the piston begins its discharge stroke and the right side begins its suction stroke.

A reciprocating pump is not complete with a liquid end only; it must also have a driving mechanism to provide motion and force to the plunger or piston. The two most common driving mechanisms are a reciprocating steam engine and a crank-and-throw device. Those pumps using the steam engine are called *direct-acting steam pumps*. Those pumps using the crank-and-throw device are called *power pumps*. Power pumps must be connected to an external rotating driving force, such as an electric motor, steam turbine, or internal combustion engine.

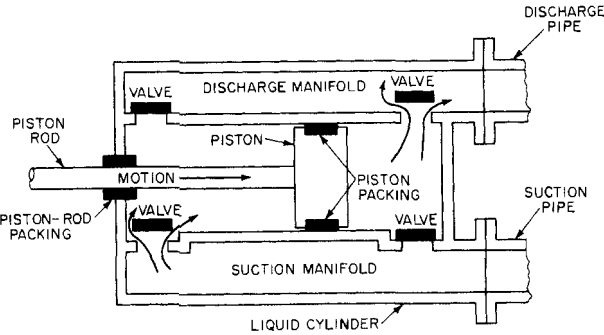


FIGURE 4 Schematic of a double-acting liquid end

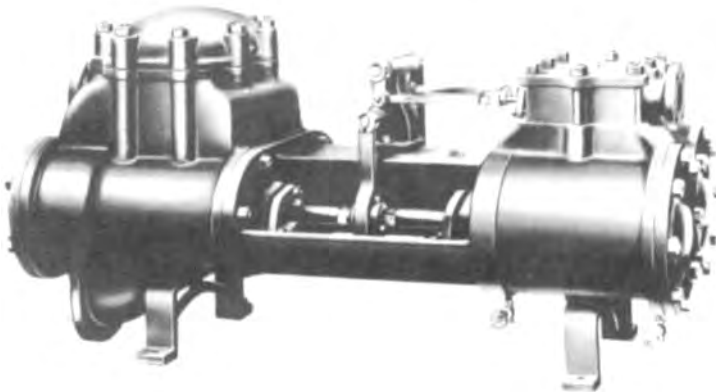


FIGURE 5 Duplex steam pump (Flowserve Corporation)

DIRECT-ACTING STEAM PUMPS

Direct-acting steam pumps are mainly classified by the number of working combinations of cylinders. For example, a duplex pump (Figure 5) has two steam and two liquid cylinders mounted side by side, and a simplex pump (Figure 6) has one steam and one liquid cylinder.

Additionally, simplex and duplex pumps may be further defined by (1) cylinder arrangement, whether horizontal or vertical; (2) number of steam expansions in the power end; (3) liquid end arrangement, whether piston or plunger; and (4) valve arrangement, that is, cap and valve plate, side pot, turret type, and so on.

Although this section will refer to steam as the driving medium, compressed gases such as air or natural gas can be used to drive a steam pump. These gases should have oil or mist added to them prior to entering the pump to prevent wear of the steam end parts.

STEAM END CONSTRUCTION AND OPERATION

The driving mechanism, or *steam end*, of a direct-acting steam pump includes the following components, as illustrated in Figure 7:

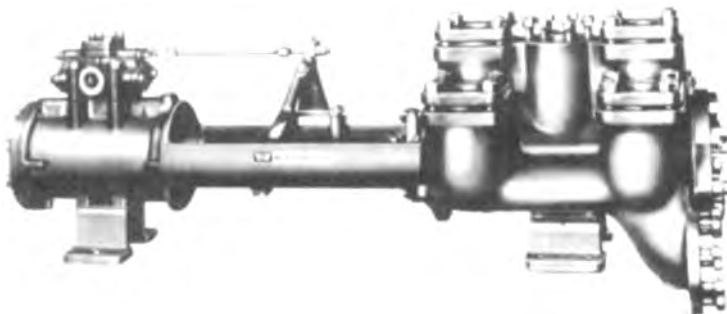


FIGURE 6 Simplex steam pump (Flowserve Corporation)

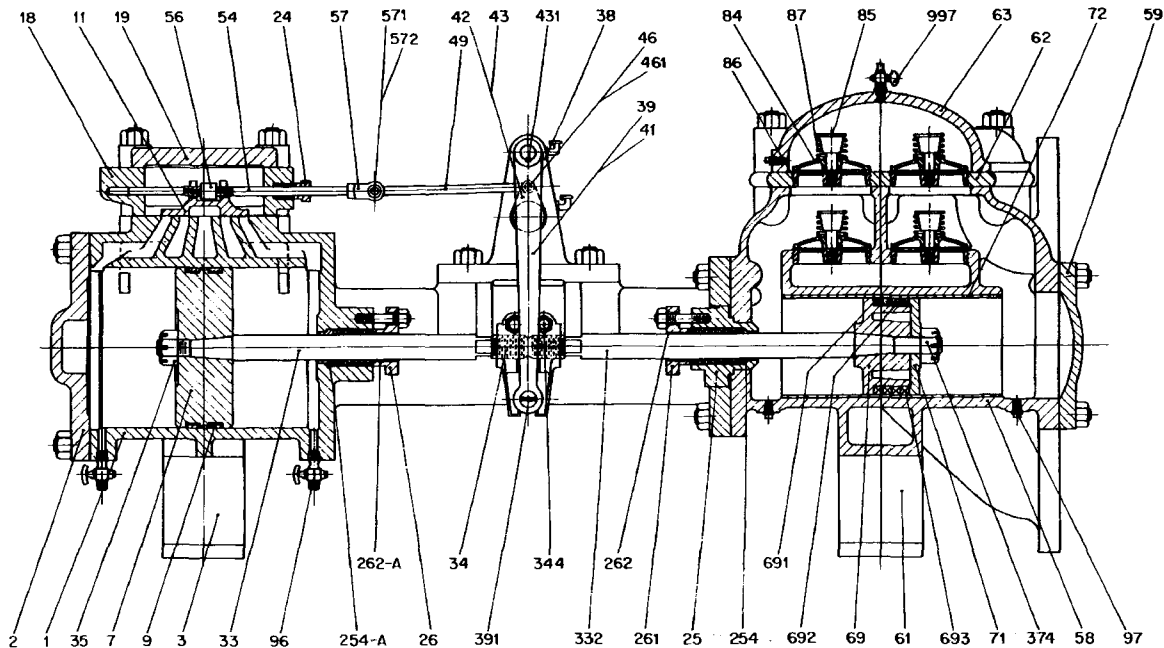


FIGURE 7 Typical section of duplex steam pump: (1) Steam cylinder with cradle, (2) steam cylinder head, (3) steam cylinder foot, (7) steam piston, (9) steam piston rings, (11) slide valve, (18) steam chest, (19) steam chest cover, (24) valve rod stuffing box gland, (25) piston rod stuffing box, liquid, (26) piston rod stuffing box gland, steam, (33) steam piston rod, (34) steam piston spool, (35) steam piston nut, (38) cross stand, (39) long lever, (41) short lever, (42) upper rock shaft, long crank, (43) lower rock shaft, short crank, (46) crank pin, (49) valve rod link, (54) valve rod, (56) valve rod nut, (57) valve rod head, (58) liquid cylinder, (59) liquid cylinder head, (61) liquid cylinder foot, (62) valve plate, (63) force chamber, (69) liquid piston body, (71) liquid piston follower, (72) liquid cylinder lining, (84) metal valve, (85) valve guard, (86) valve seat, (87) valve spring, (96) drain valve for steam end, (97) drain plug for liquid end, (254) liquid piston rod stuffing box bushing, (332) liquid piston rod, (344) piston rod spool bolt, (374) liquid piston rod nut, (391) lever pin, (431) lever key, (461) crank pin nut, (571) valve rod head pin, (572) valve rod head pin nut, (691) liquid piston snap rings, (692) liquid piston bull rings, (693) liquid piston fibrous packing rings, (997) air cock, (251) liquid piston rod stuffing box, (254A) steam piston rod stuffing box bushing, (261) piston rod stuffing box gland, liquid, (262) piston rod stuffing box gland lining, liquid, (262A) piston rod stuffing box gland lining, steam (Flowserve Corporation)

1. One or more steam cylinders with suitable steam inlet and exhaust connections
2. Steam piston with rings
3. Steam piston rods directly connected to liquid piston rods
4. Steam valves that direct steam into and exhaust steam from the steam cylinder
5. A steam valve actuating mechanism that moves the steam valve in proper sequence to produce reciprocating motion

The operation of a steam pump is quite simple. The motion of the piston is obtained by admitting steam of sufficient pressure to one side of the steam piston while simultaneously exhausting steam from the other side of the piston. There is very little expansion of the steam because it is admitted at a constant rate throughout the stroke. The moving parts, that is, the steam piston, the liquid piston, and the piston rod or rods, are cushioned and brought to rest by exhaust steam trapped in the end of the steam cylinder at the end of each stroke. After a brief pause at the end of the stroke, steam is admitted to the opposite side of the piston and the pump strokes in the opposite direction.

Steam Valves Because the steam valve and its actuating mechanism control the reciprocating motion, any detailed description of the construction of the direct-acting steam pump should rightfully begin with a discussion of steam valve types, operation, and construction.

Duplex Steam Valves The steam valves in a duplex steam pump are less complicated than those in a simplex pump and will be described first. As previously stated, the duplex steam pump can be considered as two simplex pumps arranged side by side and combined to operate as a single unit. The piston rod of one pump, in making its stroke, actuates the steam valve and thereby controls the admission or exhaust of steam in the second pump. A valve gear cross stand assembly is shown in Figure 8. The wishbone-shaped piston rod lever of one side is connected by a shaft to the valve rod crank of the opposite side. The steam valve is connected to the valve rod crank by the steam valve rod and steam valve link. Through this assembly, the piston rod of one side moves the steam valve of the opposite side in the same direction. When the first pump has completed its stroke, it must pause until its own steam valve is actuated by the movement of the second pump before it can make its return stroke. Because one or the other steam cylinder port is always open, there is no "dead center" condition; hence, the pump is always ready to start when steam

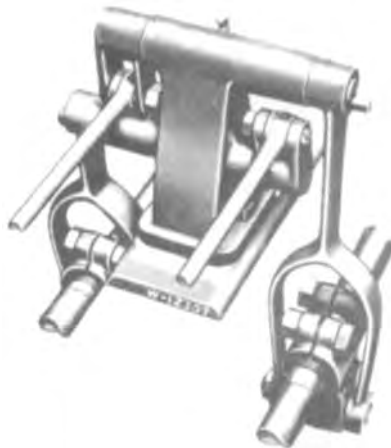


FIGURE 8 Steam valve actuating mechanism of a duplex pump (Flowserve Corporation)

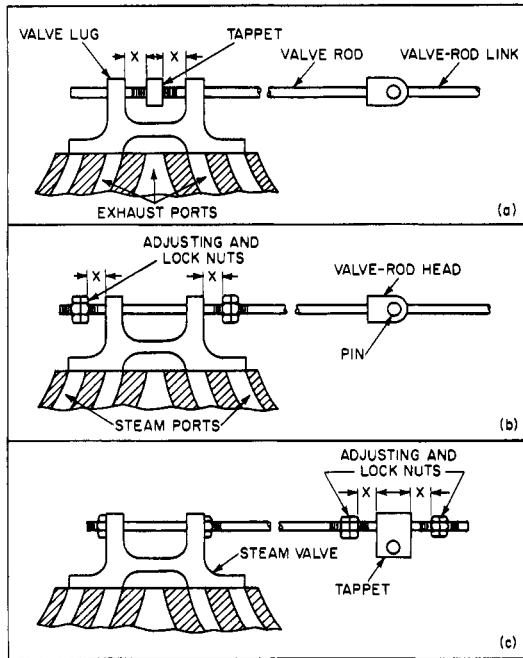


FIGURE 9A through C Duplex pump steam valve lost-motion arrangements

is admitted to the steam chest. The movements of both pistons are synchronized to produce a well-regulated flow of liquid free of excessive pulsations and interruptions.

Flat Slide Steam Valves Steam enters the pump from the steam pipe into the steam chest on top of the steam cylinder. Exhaust steam leaves the pump through the center port of five ports, as shown in Figure 9. Most duplex pumps use a flat slide valve that is held against its seat by steam pressure acting upon its entire top area; this is called an *unbalanced* valve. The flat or D valve, as it is often called, is satisfactory for steam pressures up to approximately 250 lb/in² (17 bar*) and has a reasonable service life, particularly where steam end lubrication is permissible. On large pumps, the force required to move an unbalanced valve is considerable, and so a balanced piston valve, which will be discussed later, is used.

The slide valve shown in Figure 9 is positioned on dead center over the five valve ports. A movement of the valve to the right uncovers the left-side steam port and the right-side exhaust port, which is connected through the slide valve to the center exhaust port. The main steam piston will be moved from left to right by the admitted steam. Movement of the slide valve from dead center to the left would, of course, cause opposite movement of the steam piston.

The steam valves of a duplex pump are mechanically operated, and their movements are dependent upon the motion of the piston rod and the linkage of the valve gear. In order to ensure that one piston will always be in motion when the other piston is reversing at the end of its stroke, lost motion is introduced into the valve gear. Lost motion is a means by which the piston can move during a portion of its stroke without moving the steam valve. Several lost motion arrangements are shown in Figure 9.

*1 bar = 10⁵ Pa.

If the steam valves are out of adjustment, the pump will have a tendency not to operate through its designed stroke. Increasing the lost motion lengthens the stroke; if this is excessive, the piston will strike the cylinder head. Reducing the lost motion shortens the stroke; if this is excessive, the pump will short-stroke, with a resulting loss in capacity.

The first step in adjusting the valves is to have both steam pistons in a central position in the cylinder. To accomplish this, the piston is moved toward the steam end until the piston strikes the cylinder head. With the piston rod in this position, a mark is made on the rod flush with the steam end stuffing box gland. Next, the piston rod is moved toward the liquid end until the piston strikes, and then another mark is placed on the rod halfway between the first mark and the steam end stuffing box gland. After this, the piston rod is returned toward the steam end until the second mark is flush with the stuffing box gland. The steam piston is now in central position. This procedure is repeated for the opposite piston rod assembly.

The next step is to see that both steam valves are in a central position with equal amounts of lost motion on each side, indicated by distance *X* in Figure 9.

Most small steam pumps are fitted with a fixed amount of lost motion, as shown in Figure 9a. With the slide valve centered over the valve ports, a properly adjusted pump will have the tappet exactly centered in the space between the valve lugs. The lost motion (*X*) on each side of the tappet will be equal.

Larger pumps are fitted with adjustable lost motions, such as are shown in Figure 9b. The amount of lost motion (*X*) can be changed by moving the locknuts. Manufacturers provide specific instructions for setting proper lost motion. However, one rule of thumb is to allow half the width of the steam port on each side for lost motion. A method to provide equality of lost motion is to move the valve each way until it strikes the nut and then note if both port openings are the same.

In some cases, it is desirable to be able to adjust the steam valves while the pump is in motion. With the arrangements previously mentioned, this cannot be done because the steam chest head must be removed. In a pump equipped with a lost-motion mechanism such as that shown in Figure 9c, all adjustments are external and can therefore be made while the pump is in operation.

Balanced Piston Steam Valve The balanced piston steam valve (Figure 10) is used on duplex steam pumps when the slide-type valve cannot be used because of size. The balanced piston valve can also be used without lubrication at pressure above 250 lb/in² (17 bar) and temperatures above 500°F (260°C). At higher pressures, wire drawing or steam cutting can occur as the piston slowly crosses the steam ports. To prevent wear and permanent damage to the steam chest and piston, piston rings are used on the steam valve and a steam chest liner is pressed into the steam chest to protect it.

Cushion Valves Steam cushion valves are usually furnished on larger pumps to act as an added control to prevent the steam piston from striking the cylinder heads when the pump operates at high speeds. As previously shown, the steam end has five ports, the outside ports are for steam admission, and the inside ports are for steam exhaust. As the steam piston approaches the end of the cylinder, it covers the exhaust port, trapping a

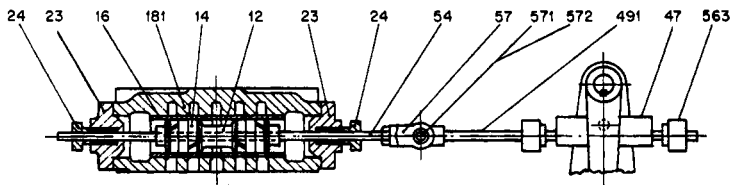


FIGURE 10 Balanced piston steam valve: (12) piston valve, (181) steam chest, (23) valve rod stuffing box, (24) valve rod stuffing box gland, (54) valve rod, complete, (57) valve rod head, (14) piston valve ring, (16) piston valve lining, (47) lost-motion block tappet, (491) valve rod link, (563) valve rod collar (Flowserve Corporation)

volume of steam in the end of the cylinder. This steam acts as a cushion and prevents the piston from striking the cylinder head. The cushion valve is simply a bypass valve between the steam and exhaust ports; by opening or closing this valve, the amount of cushion steam can be controlled.

If the pump is running at low speed or working under heavy load, the cushion valve should be opened as much as possible without allowing the piston to strike the cylinder head. If the pump is running at high speed or working under light load, the cushion valve should be closed. The amount of steam cushion and, consequently, the length of stroke can be properly regulated for different operating conditions by the adjustment of this valve.

Simplex Steam Valves The simplex pump steam valve is steam-operated, not mechanically operated as duplex steam valves are. The reason for this is that the piston rod assembly must operate its own steam valve. Consequently the travel of the valve cannot be controlled directly by means of the piston rod motion. Instead, the piston rod operates a pilot valve by means of a linkage similar to that used with a duplex pump. This controls the flow of steam to each end of the main valve, shuttling the steam back and forth. The arrangement illustrated in Figure 11 is one of the designs available to produce this motion.

With the pilot valve in the position shown in Figure 11, steam from the live steam space flows through the pilot valve steam port into the steam space at the left-hand end of the main valve (balanced piston type). Simultaneously, the D section of the pilot valve connects the steam space at the right-hand end of the main valve with the exhaust port, thereby releasing the trapped steam. The main valve has moved completely across to the right end of the chest. The main valve in this position permits steam to flow from the chest to the left steam cylinder port and, at the same time, connects the right steam cylinder port with the exhaust port.

The steam piston now moves to the right, and after the lost motion is taken up in the valve gear, the pilot valve moves to the left. In this position, the cycle previously described now takes place at the opposite end of the steam chest. Because the main valve is steam-operated, it can be in only two positions, either at the left-hand or at the right-hand end of the chest. Hence, it is impossible to have it at dead center. In other words, steam can always flow either to one side or to the other of the steam piston, regardless of the position of the steam piston.

For the valve to operate smoothly and quietly, an arrangement must be provided to create a cushioning effect on the valve travel. The steam piston, as it approaches the end of its travel, cuts off the exhaust port and traps a certain amount of steam, which acts as a cushion and stops the steam piston.

All valve adjustments are outside of the steam chest, and so it is possible to adjust the valve while the pump is in operation. The effect of decreasing or increasing the lost motion is the same as that described for duplex pumps. The lost-motion arrangement is the same as that shown in Figure 9c.

Steam End Materials For most services, cast iron is an excellent material for the steam cylinder and it is the major element of the steam end. It is readily cast in the complicated shape required to provide the steam porting. It possesses good wearing qualities, largely because of its free graphite content. This is required in the piston bores, which are continuously being rubbed by the piston rings. At high steam temperatures and pressures, ductile iron or steel is used. In the latter case, however, cast iron steam cylinder liners are frequently used because of their better wear resistance.

Counterbores are provided at each end of a steam cylinder so the leading piston ring can override, for a part of its width, the end of the cylinder bore to prevent the wearing of a shoulder on the bore.

The cylinder heads and steam pistons are also usually made of cast iron. The cylinder head has a pocket cast in it to receive the piston rod nut at the end of the stroke. Most steam pistons are made in one piece, usually with two piston ring slots machined into the outside circumference.

The relatively wide piston rings are usually made from hammered iron. They are split so they can be expanded to fit over the piston and snapped into the grooves in the piston.

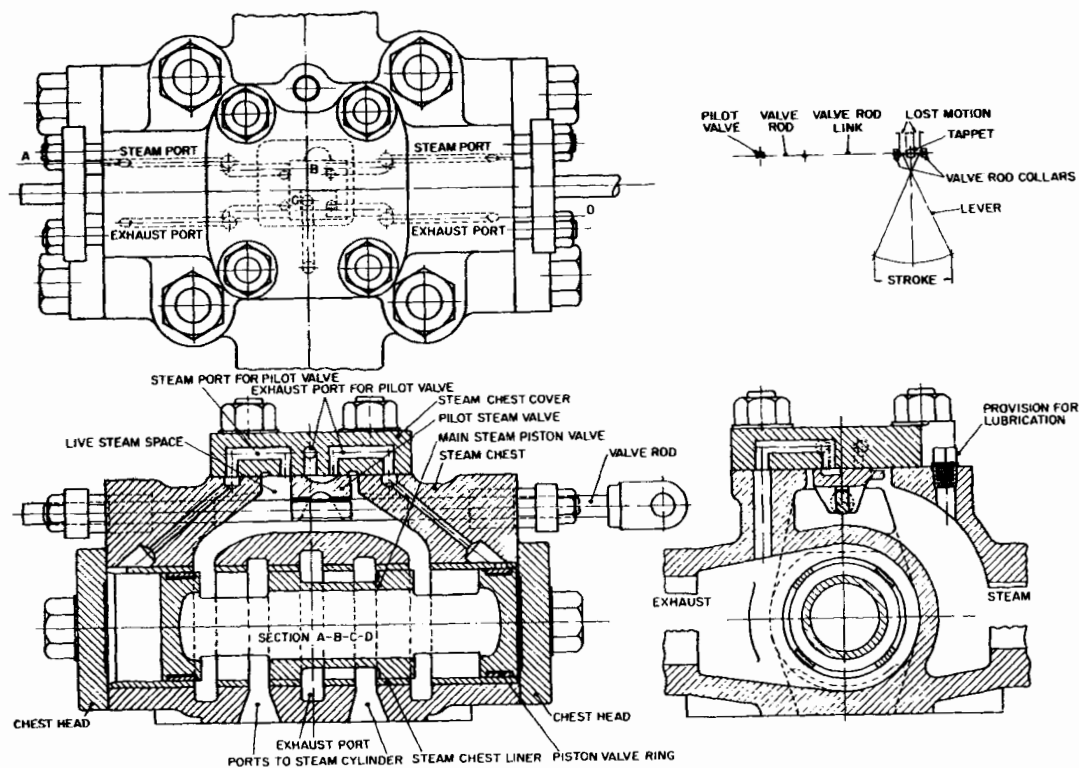


FIGURE 11 Simplex-type steam valve (Flowserve Corporation)

They must be compressed slightly to fit into the bores in the cylinder. This ensures a tight seal with the cylinder bores even as the rings wear during operation. In services where steam cylinder lubrication is not permissible, a combination two-piece ring of iron and bronze is used to obtain longer life than is obtained with the hammered iron rings.

The piston and valve rods are generally made from steel, but stainless steel and Monel are also commonly used. Packing for the rods is usually a braided graphited asbestos.

Drain cocks and valves are always provided to permit drainage of condensation, which forms in the cylinder when a pump is stopped and cools down. On each start-up these must be cracked open until all liquid is drained and only steam comes out; they are then closed.

The steam end and liquid end are joined by a cradle. On most small pumps, the cradle is cast integrally with the steam end. On large pumps, it is a separate casting or fabricated weldment.

LIQUID END CONSTRUCTION

Steam pumps are equipped with many types of liquid ends, each being designed for a particular service condition. However, they can all be classified into two basic types, the piston, or inside-packed, type and the plunger, or outside-packed, type.

The piston pump (Figure 7) is generally used for low and moderate pressures. Because the piston packing is located internally, the operator cannot see the leakage past it or make adjustments that could make the difference between good operation and packing failure. Generally, piston pumps can be used at higher pressures with noncorrosive liquids having good lubricating properties, such as oil, than with corrosive liquids, such as water.

Plunger pumps, illustrated in Figure 12, are usually favored for high-pressure and heavy-duty service. Plunger pumps have stuffing box packing and glands of the same type as those on the piston rods of piston pumps. All packing leakage is external, where it is a guide to adjustments that control the leakage and extend packing and plunger life. During operation, lubrication can be supplied to the external plunger packing to extend its life. Lubrication cannot be supplied to the piston packing rings on a piston pump.

Piston-Type Liquid Ends The most generally used piston pump is the cap-and-valve plate design, illustrated in Figure 7. This is usually built for low pressures and temperatures, although some designs are used at up to 350 lb/in² (246 bar) of discharge pressure and 350°F (177°C). The discharge valve units are mounted on a plate separate from the cylinder and have a port leading to the discharge connection. A dome-shaped cap, subject to discharge pressure, covers the discharge valve plate. The suction valve units are mounted in the cylinder directly below their respective discharge valves. A passage in the liquid cylinder leads from below the suction valves down between the cylinders of a duplex pump to the suction connection.

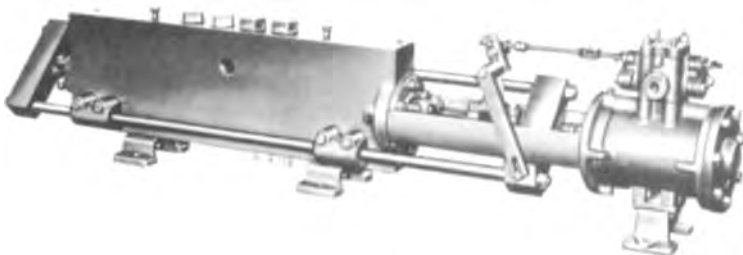


FIGURE 12 Simplex-type plunger pump (Flowserve Corporation)

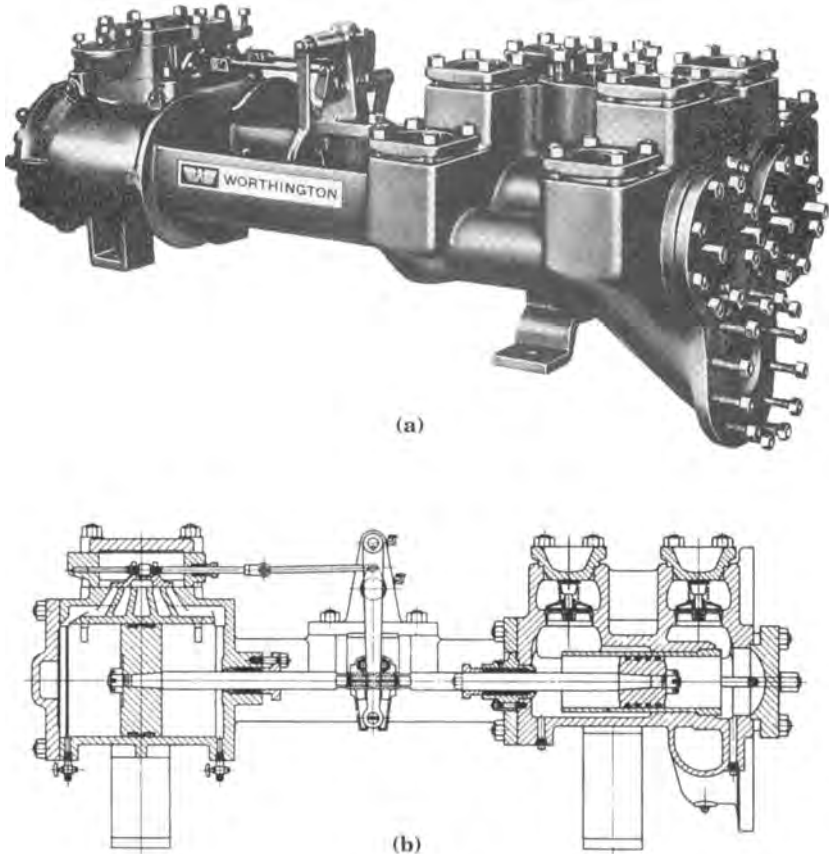


FIGURE 13A and B Side-pot piston pump (Flowsolve Corporation)

Side-pot liquid ends are used where the operating pressures are beyond the limitations of the cap-and-valve-plate pump. Figure 13 illustrates this design. Suction valves are placed in individual pots on the side of the cylinders and discharge valves in the pots above the cylinders. Each valve can be serviced individually by removing its cover. The small area of the valve covers exposed to discharge pressure makes the sealing much simpler than is the case in the cap-and-valve design. Side-pot liquid ends are widely used in refinery and oil field applications. This design is commonly employed to the maximum pressure practicable for a piston pump.

There are several specially designed piston-type liquid ends that have been developed for specific applications. One of these is the close-clearance design illustrated in Figures 14 and 15. This pump can handle volatile liquids, such as propane or butane, or a liquid that may contain entrained vapors.

The close-clearance cylinder is designed to minimize the dead space when the piston is at each end of its stroke. The liquid valves are placed as close as possible to the pump chamber to keep clearance to a minimum. The suction valves are positioned below the cylinder at the highest points in the suction manifold to ensure that all the gases are passed into the pump chamber. Although these pumps are of close-clearance design, they are not compressors and can vapor-bind; that is, a large amount of gas trapped below the

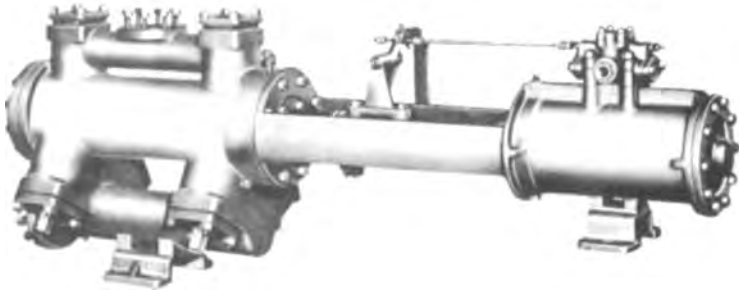


FIGURE 14 Close-clearance liquid end pump (Flowserve Corporation)

discharge valve will compress and absorb the entire displacement of the pump. When this occurs, the discharge valve will not open and this will cause a loss of flow. Hand-operated bypass or priming valves are provided to bypass the discharge valve and permit the trapped gases to escape to the discharge manifold. When the pump is free of vapors, the valves are closed.

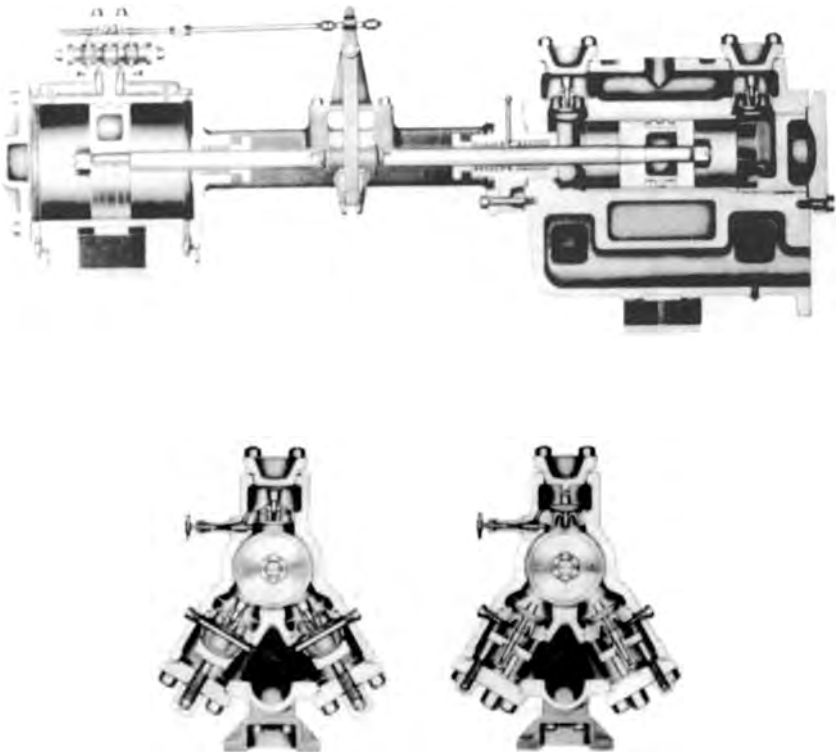


FIGURE 15 End views of close-clearance liquid end pump, showing disk valve assembly and wing valve assembly (Flowserve Corporation)

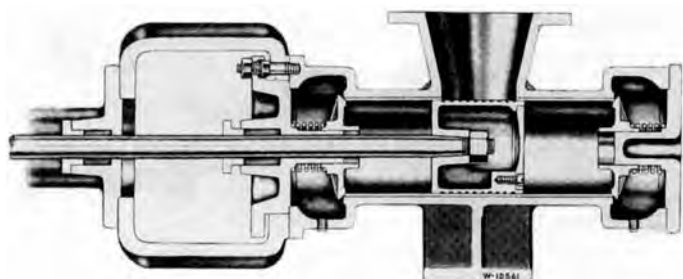


FIGURE 16 A pump for handling viscous liquid (Flowserve Corporation)

There are a number of other special piston pump designs for certain services in addition to the most common types just described. One of these special designs is the wet vacuum pump, which features tight-sealing rubber valves that permit the pump to handle liquid and air or non-condensable vapors. Another special design is made of hard, wear-resistant materials to pump cement grout on construction projects. Another design, shown in Figure 16, has no suction valves and is made for handling viscous products such as sugarcane pulp, soap, white lead, printer's ink, and tar. The liquid flows into the cylinder from above through a suction port that is cut off as the piston moves back and forth.

Piston Pump Liquid End Materials and Construction The materials used for piston pump liquid ends vary widely with the liquids handled. Most of the services to which these pumps are applied use one of the common material combinations listed in Table 1.

TABLE 1 Material and service specifications for pump liquid ends

Part	Regular Fitted (RF)	Bronze Fitted (BF)	Fully Bronze Fitted (FBB)	All Iron Fitted (AIF)	All Bronze (AB)
Cylinders	Cast iron	Cast iron	Cast iron	Cast iron	Bronze
Cylinder liners	Bronze	Bronze	Bronze	Cast iron	Bronze
Piston	Cast iron	Cast iron	Bronze	Cast iron	Bronze
Piston packing	Fibrous	Fibrous	Fibrous	Cast-iron, 3-ring	Fibrous
Stuffing box	Cast iron, bronze bushed	Cast iron, bronze bushed	Cast iron, bronze bushed	Cast iron	Bronze
Piston rod	Steel	Bronze	Bronze	Steel	Bronze
Valve	Bronze	Bronze	Bronze	Steel	Bronze
service					
Services for which most often used	Cold water; other cold liquids not corrosive to iron and bronze	Same as for RF, with reduced maintenance; continuous hot water	Boiler feed; intermittent hot-water; sodium chloride, brines	Oils and other hydrocarbons not corrosive to iron or steel; caustic solutions	Mild acids that would attack iron cylinders but not acid-resisting bronze

The liquid cylinder, the largest liquid end component, is most frequently made from cast iron or bronze. However, other materials are also used. Cast steel cylinders are used in refineries and chemical plants for high-pressure and high-temperature applications. Nickel cast steels are used for low-temperature services. Ni-Resist cast iron, chrome-alloy steels, and stainless steels are occasionally used for certain corrosive and abrasive applications, but tend to make pump cost very high. The liquid cylinder heads and valve covers are usually made from the same material as the liquid cylinder.

As was the case in the steam end, a liquid cylinder liner is used to prevent wear and permanent damage to the liquid cylinder. Liners must be replaced periodically when worn by the piston packing to the point that too much fluid leaks from one side of the piston to the other. The liners may be either of a driven-in (or pressed-in) type or of a removable type, which is bolted or clamped in position in the cylinder bore.

The pressed-in type (Figure 7) derives its entire support from the drive fit in the cylinder bore. As a rule, such a liner is relatively thin and is commonly made from a centrifugal casting or a cold-drawn brass tube. After a driven liner is worn to the point where it must be replaced, it is usually removed by chipping a narrow groove along its entire length. This groove is cut as closely as possible through the liner without damaging the wall of the cylinder bore. A flange on the liner fits into a recess at the beginning of the cylinder bore. This flange is held in contact with a shoulder by jack bolts or a spacer between the cylinder head and the end of the liner. Sometimes a packing ring is used between the flange and shoulder for a positive seal. Removable liners are heavier than pressed-in ones.

There are several designs of pistons and piston packings used for various applications. The three most common are as follows:

1. The body-and-follower type of piston with soft fibrous packing or hard-formed composition rings (Figure 17). The packing is installed in the packing space on the piston with a clearance in both length and depth. This clearance permits fluid pressure to act on one end and the inside of the packing to hold and seal it against the other end of the packing space and the cylinder liner bore.
2. The solid piston or, as shown in Figure 18, a body and follower with rings of cast iron or other materials. This type is commonly used in pumps handling oil or other hydrocarbons. The metal rings are split with an angle or step-cut joint. Their natural tension keeps them in contact with the cylinder liner, assisted by fluid pressure under the ring.
3. The cup piston (Figure 19), which consists of a body-and-follower type of piston with molded cups of materials such as rubber reinforced with fabric. Fluid pressure on the inside of the cup presses the lip out against the cylinder bore, forming a tight seal.

The piston rod stuffing boxes are usually made separate from, but of the same material as, the liquid cylinder. When handling liquids with good lubrication properties, the stuffing boxes are usually packed full with a soft, square, braided packing that is compatible with the liquid. When the liquid has poor lubricating properties, a lantern ring is

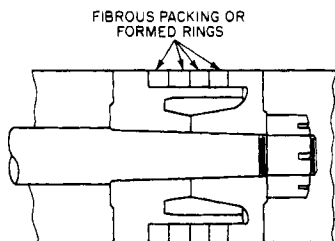


FIGURE 17 Body-and-follower piston

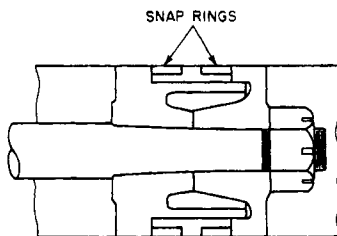


FIGURE 18 Body-and-follower piston with snap rings

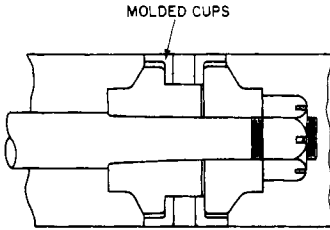


FIGURE 19 Body-and-follower piston with molded cups

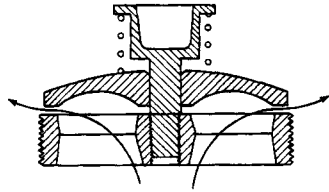


FIGURE 20 Stem-guided disk valve

installed in the center of the stuffing box with packing rings on both sides of it. A drilled hole is provided through which a lubricant, grease or oil, can be injected into the lantern ring from the outside of the stuffing box. At higher temperatures, approximately 500°F (260°C) or higher, a cooling water jacket is added to the outside of the stuffing box or as a spacer between the stuffing box and the liquid cylinder. The purpose of the cooling water jacket is to extend packing life by keeping the packing cool.

The liquid end valves of all direct-acting steam pumps are self-acting, in contrast to the mechanically operated slide valves in the steam end. The liquid end valves act like check valves; they are opened by the liquid passing through and are closed by a spring plus their weight.

Liquid end valves are roughly divided into three types: the disk valve for general service and thin liquids, the wing-guided valves for high pressures, and either the ball or the semispherical valve for abrasive and viscous liquids.

The valve shown in Figure 20 is typical of the disk type. This stem-guided design is commonly used in the cap-and-valve plate design. For hot-water boiler-feed and general service, the disk, seat, and stem are usually made of bronze, although other alloys may also be used. For lower temperatures and pressures, the disk may be made of rubber, which has the advantage of always forming a tight seal with the valve seat.

The wing-guided valve shown in Figure 21 is typical of the design use for high pressures. It derives its name from the wings on the bottom of the valve, which guide it in its seat. The beveled seating surfaces on the valve and seat tend to form a tighter seal than the flat seating surfaces on a disk valve. There is also less danger that a solid foreign particle in the liquid will be trapped between the seat and the valve. This type of valve is commonly made from a heat-treated chrome-alloy-steel forging, although a cast hard bronze and other materials may be used.

The ball valve, as its name suggests, is a ball that acts like a check valve. It is usually not spring-loaded, but guides and lift stops are provided as necessary to control its operation. The ball may be made of rubber, bronze, stainless steel, or other materials as service conditions require. The semispherical valve (Figure 22) is spring-loaded and can therefore be operated at higher speeds than the ball valve. Both the ball and the semispherical type

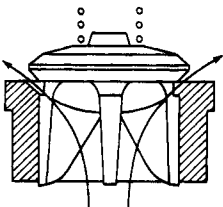


FIGURE 21 Wing-guided valve

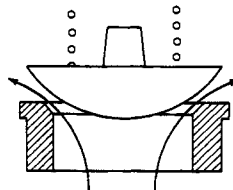


FIGURE 22 Semispherical valve

have the advantage of having no obstructions to flow in the valve seat (the disk valve seat has ribs and the wing-guided valve has vanes which obstruct the flow). The one large opening in the seat and the smooth spherical surface of ball and semispherical valves minimize the resistance to flow of viscous liquids. These types are also used for liquids with suspended solids because their rolling seating action prevents trapping of the solids between the seat and valve.

Plunger-Type Liquid Ends As mentioned previously, plunger-type pumps are used where dependability is of prime importance, even when the pump is operated continuously for long periods and where the pressure is very high. Cast liquid end plunger pumps are used for low and moderate pressures. Forged liquid end pumps (Figure 23), which are the most common plunger types, are used for high pressures and have been built to handle pressures in excess of 10,000 lb/in² (69 MPa).

Most of these designs have opposed plungers; that is, one plunger operating into the inboard end of the liquid cylinder and one into the outboard end. The plungers are solidly secured to inboard and outboard plunger crossheads. The inboard and outboard plunger crossheads are joined by side rods positioned on each side of the cylinder. With this arrangement, each plunger is single-acting; that is, it makes only one pressure stroke for each complete reciprocating cycle. The pump, however, is double-acting because the plungers are connected by the side rods.

PLUNGER PUMP LIQUID END MATERIALS The liquid cylinder of a forged liquid end plunger-type pump is most commonly made from forged steel, although bronze, Monel, chrome alloy, and stainless steels are also used. The stuffing boxes and valve chambers are usually integral with the cylinder (Figure 23), which is desirable for higher temperatures and pressures because high-temperature joints are minimized.

The liquid plungers may be made of a number of materials. The plungers must be as hard and smooth as possible to reduce friction and to resist wear by the plunger packing. Hardened chrome-alloy steels and steel coated with hard-metal alloys or ceramics are most commonly used.

The stuffing box packing used will vary widely depending upon service conditions. A soft, square packing cut to size may be used. However, solid molded rings of square, V-lip, or U-lip design are commonly used at higher pressures. Oil or grease is frequently injected into a lantern ring in the center of the stuffing box to reduce friction and reduce packing and plunger wear.

The liquid valves may be of any of the types or materials described above. However, the wing-guided valve with beveled seating surfaces is the most common because it is most suitable for high pressures.

DIRECT-ACTING STEAM PUMP PERFORMANCE

The direct-acting steam pump is a very flexible machine. It can operate at any point of pressure and flow within the limitations of the particular design. The speed of, and therefore the flow from, the pump can be controlled from stop to maximum by throttling the steam supply. This can be done by either a manual or an automatically operated valve in the steam supply line. The maximum speed of a particular design is primarily limited by the frequency with which the liquid valves will open and close smoothly. The pump will operate against any pressure imposed upon it by the system it is serving, from zero to its maximum pressure rating. The maximum pressure rating of a particular design is determined by the strength of the liquid end. In a particular application, the maximum liquid pressure developed may be limited by the available steam pressure and by the ratio of the steam piston and liquid piston areas.

STEAM PUMP CAPACITY

The flow to the discharge system is termed the pump *capacity*. The capacity, usually expressed in U.S. gallons per minute (cubic meters per minute), is somewhat less than the

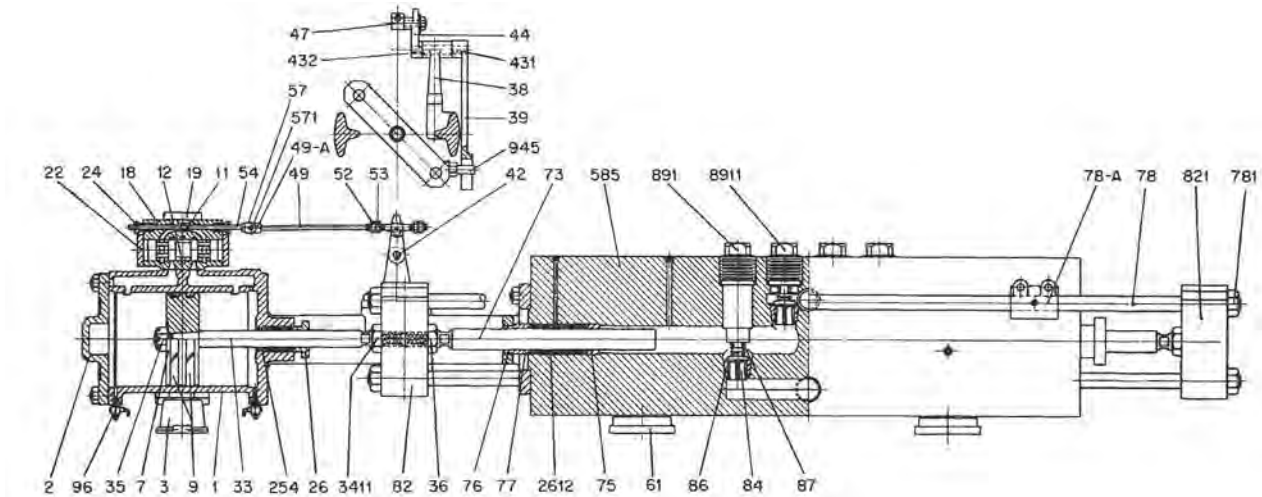


FIGURE 23 Simplex plunger pump with forged steel cylinder: (1) steam cylinder with cradle, (2) steam cylinder head, (3) steam cylinder foot, (7) steam piston, (9) steam piston rings, (11) slide valve, (12) piston valve, (18) steam chest, (19) steam chest cover, (22) steam chest head, (24) valve rod stuffing box gland, (26) steam piston rod stuffing box gland, (33) steam piston rod, (35) steam piston rod nut, (36) plunger nut, (38) cross stand, (39) lever, (42) fulcrum pin, (44) crank, (47) tappet, (49) valve rod link, (49A) valve rod link head, (52) lost-motion adjusting nut, (53) lost-motion locknut, (54) valve rod, (57) valve rod head, (61) liquid cylinder foot, (73) plunger, (75) plunger lining, (76) plunger gland flange, (77) plunger gland lining, (78) side rod, (78A) side rod guide, (82) plunger crosshead, front; (84) metal valve, (86) valve seat, (87) valve spring, (96) steam cylinder drain valve, (254) piston rod stuffing box bushing, (431) lever key, (432) crank key, (571) valve rod head pin, (585) liquid cylinder, (781) side rod nut, (821) plunger crosshead, rear, (891) suction valve plug, (945) crosshead pin, (2612) lantern gland, (8911) discharge valve plug, (3411) steam piston rod jam nut (Flowserve Corporation)

theoretical *displacement* of the pump. The difference between displacement and capacity is called *slip*. The displacement is a function of the area of the liquid piston and the speed at which the piston is moving.

The displacement of a single double-acting piston can be calculated from the formula

$$\text{in USCS units} \quad D = \frac{12 \times AS}{231} \text{ or } 0.0408 d^2 S$$

$$\text{in SI units} \quad D = 1 \times 10^{-6} \times AS$$

where D = displacement, gpm (m^3/min)

A = area of piston or plunger, in^2 (mm^2)

S = piston speed, ft/min (m/min)

d = diameter of the liquid piston or plunger, in (mm)

For a duplex double-acting pump, D is multiplied by 2. This formula neglects the area of the piston rod. For very accurate calculations, it is necessary to deduct the rod area from the piston area. This is normally not done, and the resultant loss is usually considered part of the slip.

The slip also includes losses due to leakage from the stuffing boxes, leakage across the piston on packed-piston pumps, and leakage back into the cylinder from the discharge side while the discharge valves are closing. Slip for a given pump is determined by a test. For a properly packed pump, slip is usually 3 to 5%. As a pump wears, slip will increase, but this can be compensated for by increasing the pump speed to maintain the desired capacity.

PISTON SPEED

Although piston speed in feet per minute (meters per minute) is the accepted term used to express steam pump speed, it cannot easily be measured directly and is usually calculated by measuring the revolutions per minute of the pump and converting this to piston speed. One revolution of a steam pump is defined as one complete forward and reverse stroke of the piston. The relationship between piston speed and rpm is

$$\text{in USCS units} \quad S = \frac{\text{rpm} \times \text{stroke}}{6}$$

$$\text{in SI units} \quad S = 0.002 \times \text{rpm} \times \text{stroke}$$

where S = piston speed, ft/min (m/min)

rpm = revolutions per minute

stroke = stroke of pump, in (mm)

A steam pump must fill with liquid from the suction supply on each stroke, or it will not perform properly. If the pump runs too fast, the liquid cannot flow through the suction line, pump passageways, and valves fast enough to follow the piston. On the basis of experience and hydraulic formulas, maximum piston speeds that vary with the length of stroke and the liquid handled can be established.

Table 2 shows general averages of maximum speed ratings for pumps of specified stroke handling various liquids. Some pumps, by reason of exceptionally large valve areas or other design features, may be perfectly suitable for speeds higher than shown. From the table, it should be noted that piston speed should be reduced for viscous liquids. Unless the net positive suction head is proportionately high, viscous liquids will not follow the piston at high speeds because frictional resistance in suction lines and in the pump increases with viscosity and rate of flow. Pumps handling hot water are run more slowly to prevent boiling of the liquid as it flows into the low-pressure area behind the piston.

TABLE 2 Average maximum speed ratings

Stroke Length, in*	Piston speed, ft/min*					Boiler Feed 212°F (100°C)
	Cold Water; Oil to 250 SSU	Oil, 250–500 SSU	Oil, 500–1000 SSU	Oil, 1000–2500 SSU	Oil, 2500–5000 SSU	
3	37	35	33	29	24	22
4	47	45	42	36	31	28
5	53	51	47	41	35	32
6	60	57	53	46	39	36
7	64	61	57	49	42	39
8	68	65	61	53	45	41
10	75	72	67	58	49	45
12	80	77	71	62	52	48
15	90	86	80	69	57	54
18	95	91	85	73	62	57
24	105	100	94	81	68	63
36	120	115	107	92	78	72

*SI conversion factors: in × 25.4 = mm; ft/min × 0.3048 = m/min.

SIZE OF LIQUID END

The size of a steam pump is always designated as follows:

Steam piston diameter × liquid piston diameter × stroke

For example, a $\frac{1}{2} \times 5 \times 6$ steam pump has a $7\frac{1}{2}$ -in (191-mm) diameter steam piston, a 5-in (127-mm) diameter liquid piston, and a 6-in (152-mm) stroke.

To determine the liquid piston diameter for a specified capacity, the following procedure is used. First, a reasonable stroke length is assumed and the maximum piston speed for this stroke and the type of liquid pumped is selected from Table 2. Then the desired capacity is increased by 3 to 5% to account for slip. The result is the desired displacement. Then for either a simplex or duplex pump, the liquid piston diameter can be calculated as follows:

In USCS units, for simplex pumps: $d_l = 4.95 \left(\frac{D}{S} \right)^{1/2}$

For duplex pumps: $d_l = 3.5 \left(\frac{D}{S} \right)^{1/2}$

In SI units, for simplex pumps: $d_l = 1128.4 \left(\frac{D}{S} \right)^{1/2}$

For duplex pumps: $d_l = 797.9 \left(\frac{D}{S} \right)^{1/2}$

where d_l = liquid piston diameter, in (mm)

D = displacement, gpm (m^3/min)

S = piston speed, ft/min (m/min)

Using the resultant liquid piston diameter, the next larger standard piston size is selected.

SIZE OF STEAM END

To calculate the size of the steam end required for a specific application, the basic principle of steam pump operation should be considered. A simple schematic of a steam pump is shown in Figure 24.

In order for the pump to move, the force exerted on the steam piston must exceed the force on the liquid piston that is opposing it. The force on the steam piston is the product of the net steam pressure and the steam piston area. The *net* steam pressure is the steam inlet pressure minus the exhaust pressure. The force acting on the liquid piston is the product of the net liquid pressure and the liquid piston area. The *net* liquid pressure is the pump discharge pressure minus the suction pressure or *plus* the suction lift. This may be expressed algebraically as follows:

$$P_s A_s > P_l A_l$$

where P_s = net steam pressure, lb/in² (bar)

A_s = steam piston area, in² (mm²)

P_l = net liquid pressure, lb/in² (bar)

A_l = liquid piston area, in² (mm²)

Because the pistons are circular, the squares of their diameters are directly proportional to their areas, and the above formula can be rewritten as

$$P_s d_s^2 > P_l d_l^2$$

where d_s = steam piston diameter, in (mm)

d_l = liquid piston diameter, in (mm)

In practice, it is necessary for the force on the steam piston to exceed the force opposing it on the liquid piston by a considerable amount. This is because of mechanical losses, which include stuffing box friction, friction between piston rings and cylinder of both liquid and steam ends, and the operation of the valve gear. These losses are determined by testing and are accounted for in size calculations by introduction of a mechanical efficiency figure. Mechanical efficiencies are expressed as a percentage, with 100% being a perfect balance of forces acting on the steam and liquid pistons as expressed in the previous formula. Because the efficiencies of two identical pumps may vary with stuffing box and piston ring packing tightness, the efficiencies published by manufacturers tend to be conservative.

With the mechanical efficiency factor inserted, the formula of forces becomes

$$P_s d_s^2 E_m = P_l d_l^2$$

where P_s = net steam pressure, lb/in² (bar)

d_s = steam piston diameter, in (mm)

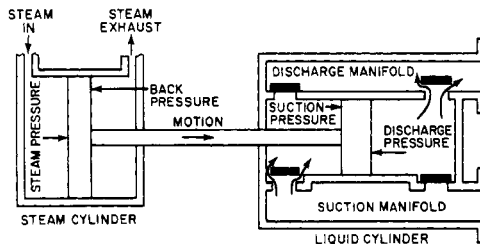


FIGURE 24 Schematic showing direction of forces on pistons

TABLE 3 Typical mechanical efficiency values

Stroke length, in (mm)	Mechanical efficiency, %	
	Piston Pump	Plunger pump
3 (76)	50	47
4 (102)	55	52
5 (127)	60	57
6 (152)	65	61
8 (201)	65	61
10 (254)	70	66
12 (305)	70	66
18 (457)	73	69
24 (610)	75	71

P_l = net liquid pressure, lb/in² (bar)

d_l = liquid piston diameter, in (mm)

E_m = mechanical efficiency, expressed as a decimal

This formula is commonly used to determine the *minimum* size of steam piston required when the liquid piston size has already been selected and the net steam and net liquid pressures are known. For this calculation, the formula is rearranged to the form

$$d_s = d_l \left(\frac{P_l}{P_s E_m} \right)^{1/2}$$

The efficiency of a long-stroke pump is greater than that of a short-stroke pump. Although mechanical efficiency varies with stroke length, any two pumps of the same size are capable of the same efficiency.

Table 3 shows typical mechanical efficiencies that can be used to determine required steam end size.

STEAM CONSUMPTION AND WATER HORSEPOWER

After determining the proper size of other pump types, the next concern usually is to calculate the maximum brake horsepower so the proper size of driver can be selected. With a steam pump, the next step is usually to determine the steam consumption. This must be known to ensure that the boiler generating the steam is large enough to supply the steam required by the pump as well as that required for all its other services.

To determine the steam consumption, it is necessary first to calculate the water horsepower as follows:

$$\text{In USCS units} \quad whp = \frac{Q \times P_l}{1715}$$

$$\text{In SI units} \quad kW = \frac{Q \times P_l}{36}$$

where whp = water horsepower

Q = pump capacity, gpm (m³/h)

P_l = net liquid pressure, lb/in² (bar)

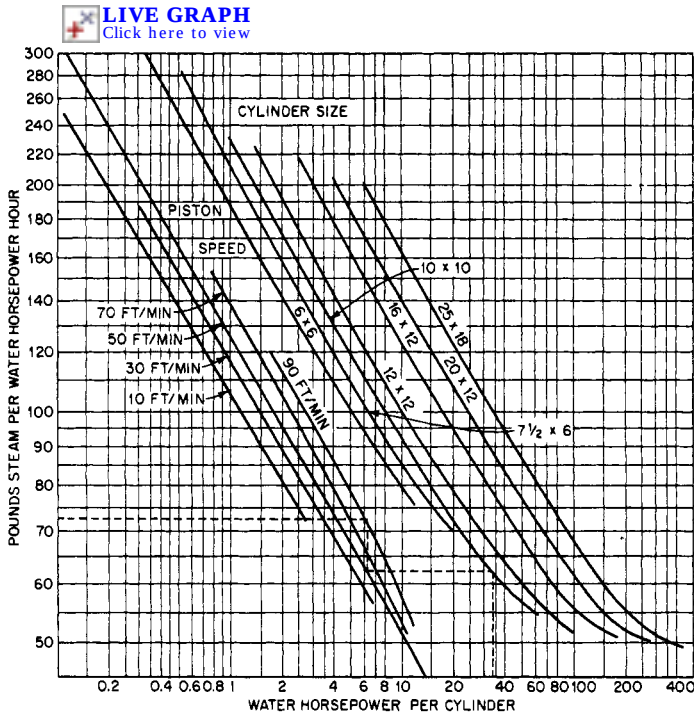


FIGURE 25 Approximate steam consumption for steam pumps (pounds of steam per water horsepower-hour = 1.644 kg of steam per water kilowatt-hour; water horsepower per cylinder = 1.341 kW per cylinder; ft/s = 0.3047 m/s; in = 25.44 mm) (Hydraulic Institute Standards, 12th Edition, 1969—out of print)

A steam consumption chart (Figure 25) affords a means of quickly obtaining an approximate figure for the steam rate of direct-acting steam pumps. For duplex pumps, divide the water horsepower by 2 before applying it to the curves. These curves were made up on the basis of water horsepower per cylinder; if the above procedure is not followed, the results will be inaccurate.

Starting with the water horsepower per cylinder:

1. Move vertically to the curve for steam cylinder size.
2. Move horizontally to the curve for 50-ft/min (15.15-m/min) piston speed. This is the basic curve from which the other curves were plotted.
3. Move vertically to the piston speed at which the pump will run.
4. Move horizontally to the steam rate scale and read it in pounds per water horsepower-hour (kilograms per kilowatt-hour).
5. Multiply the result by total water horsepower to obtain the steam rate in pounds per hour (kilograms per hour).

For steam cylinders with diameters as shown, but with longer stroke, deduct 1% from the steam rate for each 20% of additional stroke. Thus, a 12×24 steam end will have a steam consumption about 5% less than a 12×12 steam end. For $5\frac{1}{4} \times 5$ and $4\frac{1}{2} \times 4$ steam ends, the 6×6 curve will give approximate figures. For cylinders of intermediate diameters, interpolate between the curves.

To correct for superheated steam, deduct 1% for each 10° of superheat. To correct for back pressure, multiply the steam rate by a correction factor equal to

$$\left(\frac{P + BP}{P} \right)^{1/2}$$

where P = net steam pressure to drive pump, lb/in² (bar)

BP = back pressure, lb/in² (bar)

Direct-acting steam pumps have inherently high steam consumption. This is not necessarily a disadvantage, however, when the exhaust steam can be used for heating the boiler-feed water or for building heating or process work. Because these pumps can operate with a considerable range of back pressure, it is possible to recover nearly all the heat in the steam required to operate them. Because they do not use steam expansively, they are actually metering devices rather than heat engines and as such consume heat from the steam only as the heat is lost via radiation from the steam end of the pump. These pumps act, in effect, like a reducing valve to deliver lower pressure steam that contains nearly all its initial heat.

SUCTION SYSTEMS AND NET POSITIVE SUCTION HEAD

A majority of pump engineers agree that most operating problems with pumps of all types are caused by failure to supply adequate suction pressure to fill the pump properly.

The steam pump industry uses the term *net positive suction head required* (NPSHR) to define the head, or pressure, required by the pump over its datum, usually the discharge valve level. This pressure is needed to (1) overcome frictional losses in the pump, (2) overcome the weight and spring loadings of the suction valves, and (3) create the desired velocity in the suction opening and through the suction valves. The NPSHR of a steam pump will increase as the piston speed and capacity are increased. The average steam pump will have valves designed to limit the NPSHR to 5 lb/in² (0.3 bar) (34.5 MPa) or less at maximum piston speed.

If the absolute pressure in the suction system minus the vapor pressure is inadequate to meet or exceed the NPSHR, the pump will cavitate. Cavitation is the change of a portion of the liquid to vapor, and it causes a reduction in delivered capacity, erratic discharge pressure, and noisy operation. Even minor cavitation will require frequent refacing of the valves, and severe cavitation can cause cracked cylinders or pistons or failure of other major parts.

FLOW CHARACTERISTICS

The flow characteristics of duplex and simplex pumps are illustrated in Figure 26. The flow from a simplex pump is fairly constant except when the pump is at rest. However,

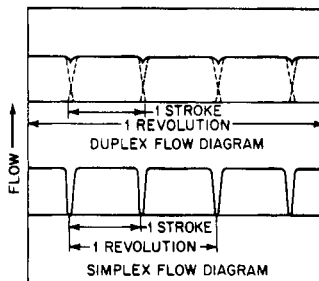


FIGURE 26 Flow characteristics of simplex and duplex pumps

because the flow must stop for the valves to close and for the forces on both sides of the steam and liquid pistons to reverse, there is uneven and pulsating flow. This can be compensated for, in part, by installing a pulsation-dampening device on the discharge side of the pump or in the discharge line.

In a duplex pump, one piston starts up just before the other piston completes its stroke, and the overlapping of the two strokes eliminates the sharp capacity drop.

FURTHER READING

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SECTION 3.4

DISPLACEMENT PUMP PERFORMANCE, INSTRUMENTATION, AND DIAGNOSTICS

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The most common operational and reliability problems in reciprocating positive displacement pump systems are characterized by

- Low net positive suction head (NPSH), pulsations, pressure surge, cavitation, waterhammer
- Vibrations of pump or piping
- Mechanical failures, wear, erosion, alignment
- High horsepower requirements, high motor current, torsional oscillations
- Temperature extremes, thermal cycling
- Harsh liquids: corrosive, caustic, colloidal suspensions, precipitates (slurries)

Although any component in a pump system may be defective, most operational problems are caused by liquid transient interaction of the piping system and pump system or by purely mechanical interaction of the pump, drive system, foundation, and so on. This section discusses hydraulic and mechanical problems and suggests measurement and diagnostic procedures for determining the sources of these problems.

HYDRAULIC AND MECHANICAL PUMP PROBLEMS

Inadequate NPSH Net positive suction head available (NPSHA) is the static head plus atmospheric head minus lift loss, frictional loss, vapor pressure, and acceleration head available at the suction connection centerline.

Acceleration head can be the highest factor of NPSHA. In some cases, it is 10 times the total of all the other losses. Data from both the pump and the suction system are required

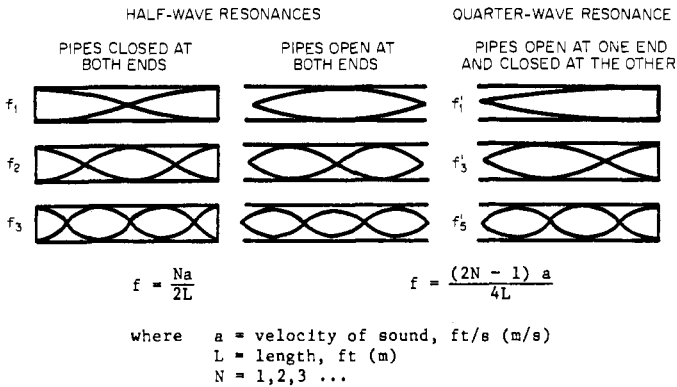


FIGURE 1 Organ pipe resonant mode shapes

to determine acceleration head; its value cannot be calculated until these data have been established. Inadequate NPSH can cause cavitation, the rapid collapse of vapor bubbles, which can result in a variety of pump problems, including noise, vibration, loss of head and capacity, and severe erosion of the valves and surfaces in the adjacent inlet areas. To avoid cavitation of liquid in the pump or piping, the absolute liquid static pressure at pumping temperatures must always exceed the vapor pressure of the liquid. The pressure at the pump suction should include sufficient margin to allow for the presence of pulsations as well as pressure losses due to flow.

Positive Displacement Pump Pulsations The intermittent flow of a liquid through pump internal valves generates liquid pulsations at integral multiples of the pump operating speed. For example, a 120-rpm triplex pump generates pulsations at all multiples of pump speed (2 Hz, 4 Hz, and so on); however, the most significant components will usually be multiples of the number of plungers (6 Hz, 12 Hz, 18 Hz, and so on). Resultant pulsation pressures in the piping system are determined by the interaction of the generated pulsation spectrum from the pump and the acoustic length resonances of liquid in the piping. For variable-speed units, the discrete frequency components change in frequency as a function of operating speed and the measured amplitude of any pulsation harmonic can vary substantially with changes in the location of the measurement point relative to the pressure nodes and antinodes of the standing wave pattern.

Piping System Pulsation Response Because acoustic liquid resonances occur in piping systems of finite length, these resonances will selectively amplify some pulsation frequencies and attenuate others. Resonances of individual piping segments can be described from organ pipe acoustic theory. The resonant frequencies of standing pressure waves depend upon the velocity of sound in the liquid being pumped, pipe length, and end conditions. The equations for calculating these frequencies are shown in Figure 1. All of the integral multiples (N) of a resonance can occur, and it is desirable to mismatch the excitation frequencies from any acoustical resonances. A 2:1 diameter increase or greater would represent an open end for the smaller pipe. Closed valves, pumps, or a 2:1 diameter reduction represent closed ends. For example, a 2-in (51-mm) diameter pipe that connects radially into two 8-in (203-mm) diameter volumes would respond acoustically as an open-end pipe.

Complex piping system responses depend upon the termination impedances and interaction of acoustical resonances and cannot be handled with simplified equations. An electroacoustic analog¹ or digital computer can be used for the more complex systems.

Velocity of Sound in Liquid Piping Systems The acoustic velocity of liquids can be determined by the following equation:

$$a = C_1 \sqrt{\frac{K_s}{\text{sp. gr.}}} \quad (1)$$

where a = velocity of sound, ft/s (m/s)

C_1 = 8.615 for USCS units, 1.0 for SI units

K_s = isentropic bulk modulus, lb/in² (kPa)

sp. gr. = specific gravity

In liquid piping systems, the acoustic velocity can be significantly affected by pipe wall flexibility. The acoustic velocity can be adjusted by the following equation:

$$a_{\text{adjusted}} = a \sqrt{\frac{1}{1 + \frac{DK_s}{tE}}} \quad (2)$$

where D = pipe diameter, in (mm)

t = pipe wall thickness, in (mm)

E = elastic modulus of pipe material, lb/in² (kPa)

Pipe wall radial compliance can reduce the velocity of sound in liquid in a pipe as shown in Figure 2.²

The bulk modulus of water can be calculated with the following equation³ for temperatures from 0 to 212°F (0 to 100°C) and pressures from 0 to 4.4×10^4 lb/in² (0 to 3 kbar*):

$$K_s = K_0 + 3.4P \quad (3)$$

where K_s = isentropic bulk modulus, kbar

K_0 = constant from Table 1, kbar

P = pressure, kbar

(1 kbar = 10^5 kPa = 14,700 lb/in²)

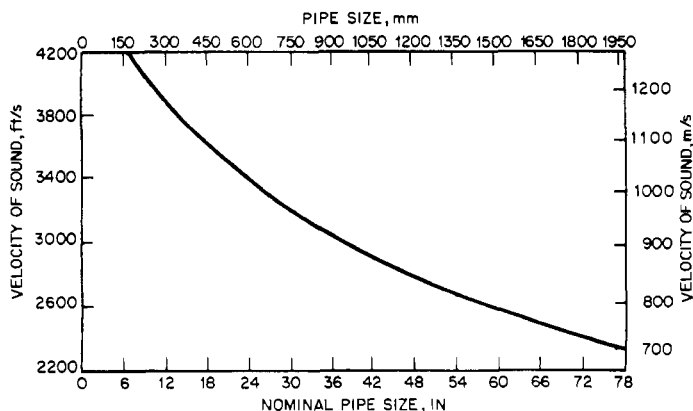


FIGURE 2 Velocity of sound in water at 14.6 lb/in² (1 bar), 60°F (15.6°C) versus nominal pipe size with 0.25-in (6.35-mm) wall thickness

*1 bar = 10^5 Pa.

TABLE 1 Constant K_0 for evaluation of isentropic bulk modulus of water from 0 to 3 kbar

Temperature, °C (°F)		Isentropic constant K_0 , kbar ^a
0	(32)	19.7
10	(50)	21.0
20	(68)	22.0
30	(86)	22.7
40	(104)	23.2
50	(122)	23.5
60	(140)	23.7
70	(158)	23.7
80	(176)	23.5
90	(194)	23.3
100	(212)	22.9

^a1 kbar = 14,700 lb/in².
Source: Reference 3.

The calculation of the isentropic bulk modulus of water is accurate to $\pm 0.5\%$ at 68°F (20°C) and lower pressures.⁴ At elevated pressures (greater than 3 kbar) and temperatures (greater than 100°C), the error should not exceed $\pm 3\%$.

The bulk modulus for petroleum oils (hydraulic fluids) can be obtained at various temperatures and pressures by using Figures 3 and 4, which were developed by the American

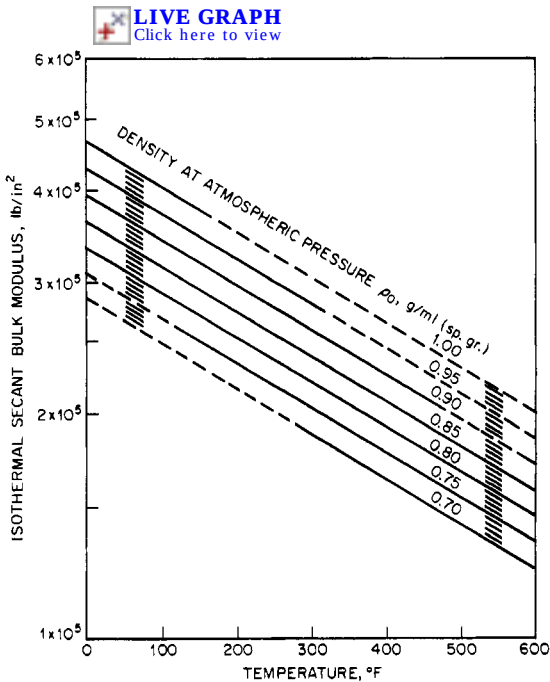


FIGURE 3 Isothermal secant bulk modulus at 20,000 lb/in² gage for petroleum oils (1 lb/in² = 6.895 kPa; °C = (°F - 32)/1.8).

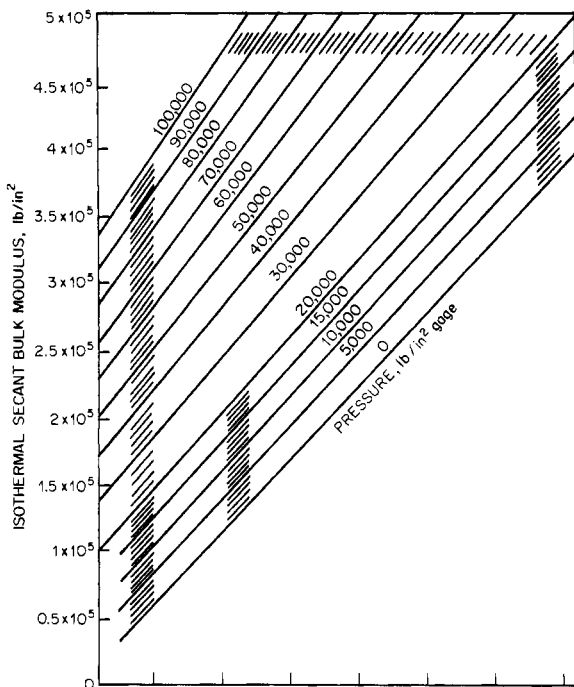


FIGURE 4 Pressure correction for isothermal secant bulk modulus for petroleum oils (1 lb/in² = 6.895 kPa)

Petroleum Institute (API).⁴⁻⁷ Figure 3 relates density (mass per unit volume) and temperature to the isothermal secant bulk modulus at 20,000 lb/in² (137,900 kPa), and Figure 4 corrects for various pressures.

The isentropic tangent bulk modulus is needed to calculate the speed of sound in hydraulic fluids and can be readily obtained from Figures 3 and 4 as follows:

1. Read the isothermal secant bulk modulus at the desired temperature from Figure 3.
2. Using the value for isothermal secant bulk modulus obtained from Figure 3, go to Figure 4 and locate the intersection of the pressure line with that value. Move vertically to the pressure line representing twice the normal pressure. Read the adjusted isothermal secant bulk modulus for the double value of pressure.
3. Multiply the adjusted isothermal secant bulk modulus by 1.15 to obtain the value of the isentropic tangent bulk modulus (compensation for the ratio of specific heats).

The isothermal tangent bulk modulus has been shown to be approximately equal to the secant bulk modulus at twice the pressure⁸ within $\pm 1\%$. The relationship between isothermal bulk modulus K_t and isentropic bulk modulus K_s is

$$K_s = K_t c_p / c_v \quad (4)$$

The value of c_p / c_v for most hydraulic fluids is approximately 1.15.

Pulsation Control Pulsation control can be achieved by judicious use of acoustic filters and side branch accumulators. Acoustic filters are liquid-filled devices consisting of volumes and chokes that use reactive filtering techniques to attenuate pulsations. Side

branch resonators are of two types: quarter-wavelength resonant stubs and gas-charged accumulators. (The terms *accumulators*, *dampeners*, and *dampers* are used interchangeably in the liquid filter industry.)

Acoustic Filters An acoustic filter consisting of two volumes connected by a small-diameter choke can significantly reduce the transmission of pulsations from the pump into the suction and discharge piping systems. The equations given in Figure 5 can be used to calculate the resonant frequency for a simple volume-choke-volume filter. The filter should be designed to have a resonant frequency no more than one-half the lowest frequency desired to be reduced, referred to as the cutoff frequency. Such a filter is called a low-pass filter because it attenuates frequencies above the cutoff frequency.

A special case for symmetric liquid-filled filters can be obtained by choosing equal chamber and choke lengths. This reduces the equation in Figure 5 to

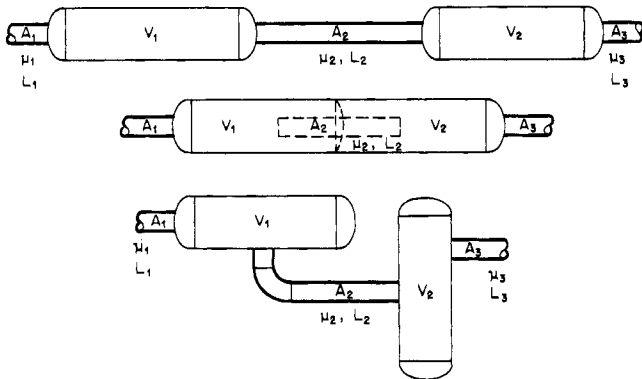
$$f = \frac{ad}{\pi \sqrt{2CD}} \quad (5)$$

where d = choke diameter, in (mm)

L = chamber and choke length, ft (m)

D = chamber diameter, in (mm)

Normally, a good filter design will have a resonant frequency less than one-half the plunger frequency and will have a minimal pressure drop. For example, a triplex pump running at 600 rpm generates pulsations at all multiples of 10 Hz. The largest amplitudes



$$\mu_j = \frac{A_j}{L_j + 1/2 \sqrt{\pi A_j}} \quad \text{for } j = 1, 2, 3$$

$$f = \frac{a}{2\pi} \sqrt{\frac{1}{2} \left[\frac{\mu_1 + \mu_2}{V_1} + \frac{\mu_2 + \mu_3}{V_2} \right] \pm \sqrt{\left(\frac{\mu_1 + \mu_2}{V_1} - \frac{\mu_2 + \mu_3}{V_2} \right)^2 + \frac{4\mu_2^2}{V_1 V_2}}}$$

FOR EQUAL VOLUMES, THE RESONANT FREQUENCY IS APPROXIMATELY:

$$f = \frac{a}{\sqrt{2} \pi} \sqrt{\frac{\mu_2}{V_1}}$$

FIGURE 5 Two-chamber resonator system with both ends open. V = volume, ft³ (m³); f = resonant frequency, Hz; L = choke tube length, ft (m); A = choke tube area, ft² (m²); a = acoustic velocity, ft/s (m/s); m = acoustic parameter

would normally be at 30 Hz, 60 Hz, 90 Hz, and so on. The filter resonant frequency should be set at 15 Hz or lower. For water, in which the velocity of sound is 3200 ft/s (976 m/s), and a volume bottle size inner diameter of 19 in (482.6 mm),

$$\text{in USCS units} \quad \frac{L}{d} = \frac{3200}{\pi \sqrt{2} (15)(19)} = 2.53 \frac{\text{ft}}{\text{in}}$$

$$\text{in SI units} \quad \frac{L}{d} = \frac{976}{\pi \sqrt{2} (15)(482.6)} = 0.03 \frac{\text{m}}{\text{mm}} \quad (6)$$

If the choke diameter is selected to be 1.049 in (26.64 mm), the length of each volume bottle and choke tube is 2.65 ft (0.81 m). See also Reference 8.

Side Branch Accumulators Liquid-filled, quarter-wavelength, side branch accumulators reduce pulsations in a narrow frequency band and can be effective on constant/speed positive displacement pumps. However, in variable-speed systems, accumulators with or without a bladder can be made more effective by partially charging them with a gas (nitrogen or air) because the gas charge cushions hydraulic shocks and pulsations. If properly selected, located, tuned, and charged, a wide variety of accumulators (weight- or spring-loaded; gas-charged) can be used in positive displacement pump systems to prevent cavitation and waterhammer, damp pulsations, and reduce pressure surges.^{9,10} Improper sizing or location can aggravate existing problems or cause additional ones. Typically, the best location for accumulators is as close to the pump as possible.

Gas-charged dampeners, or accumulators, such as those depicted in Figure 6, are most commonly used and can be quite effective in controlling pulsations. These devices are commercially available from several sources. Their location and volume and the pressure of the charge are important to their effectiveness. When gas-charged dampeners are used, the gas pressure must be monitored and maintained because the gas can be absorbed into the liquid. The system pressure can sometimes be lower than the gas charging pressure, such as on start-up; therefore, a valve should be installed to shut off the accumulator during start-up to eliminate gas leakage to the primary liquid. When the valve is closed, the accumulator is decoupled from the system and is not effective. Accumulators with integral check valves should be adjusted so pressure transients do not close the check valve and render the accumulator ineffective. Accumulators that have bladders (Figures 6b, d, and e) to separate the gas charge from the liquid have some distinct advantages, particularly if gas absorption is a problem. Accumulators with flexible bladders must be carefully maintained because failure of a bladder could release gas into the liquid system and could compromise the effectiveness of the dampener.

The in-line gas dampener (refer to Figure 6e) has a cylinder around the pipe containing a gas volume and bladder. The liquid enters the dampener through small holes in the circumference of the pipe and impinges upon the bladder, which produces the same acoustic effect as a side branch configuration.

It is not always possible to design effective pulsation control systems using simplified techniques. For complicated piping systems with multiple pumps, an electroacoustic

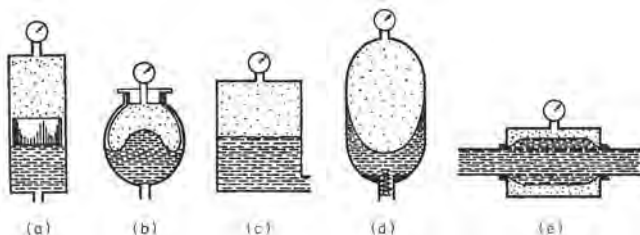


FIGURE 6 Types of accumulators: (a) piston, (b) diaphragm, (c) gas-charged, (d) bladder, (e) in-line



LIVE GRAPH
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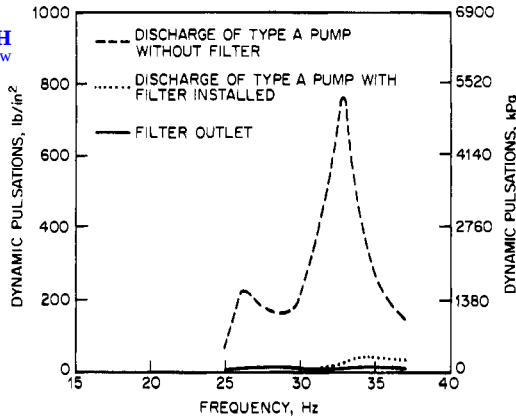


FIGURE 7 Effect of acoustic filter on pulsations

analog¹ is recommended for designing optimum filters or accumulators. This tool has become widely accepted for designing reliable piping systems for reciprocating liquid pumps and gas compressor units. The reduction in pulsations in a liquid pump system in a nuclear plant is shown in Figure 7. These results were obtained with a two-volume acoustic filter system designed with the electroacoustic analog. The volume diameter was 19.3 in (49 cm), and the length was 4 ft (1.2 m). The choke tube diameter was 0.8 in (2 cm), and its length was 7 ft (2.1 m). The speed of the triplex (2-cm) pump was 360 rpm, and the filter resonant frequency was set at 8.1 Hz for a velocity of sound of 4550 ft/s (1390 m/s).

Piping Vibrations When mechanical resonances are excited by pulsations, vibrations in the pump and piping can sometimes be 20 times higher than under off-resonant conditions. When the mechanical resonances coincide with the acoustic resonances, an additional amplification factor as high as 300 can be encountered.

Piping system mechanical natural frequencies can be calculated using simplified design procedures to provide effective detuning from known excitation sources. A nomogram (see Figure 8) for calculating the lowest natural frequency of uniform steel piping spans¹¹ can be used in designing piping systems and in diagnosing and solving vibration problems. For example, welded between two bottles, a 4-in (102-mm) pipe that is 10 ft (3 m) long and has an inner diameter of 3.826 in (97.18 mm) would have a mechanical frequency of 74 Hz. If the pipe was an equal-leg L bend ($L = 5$), the natural frequency would be 50 Hz.

To minimize piping vibration problems, all unnecessary bends (considering routing and thermal flexibility) should be eliminated because they provide a strong coupling point between pulsation excitation forces and the mechanical system. When bends are needed, use the largest enclosed angle possible and locate restraints near each bend. Piping should also have supports near all piping size reductions and at large masses (valves, accumulators, flanges, and so on). Small auxiliary piping connections (vents, drains, pressure test connections, and so on) should be designed so the mass of the valve and flange is effectively tied back to the main piping, thus eliminating relative vibration.

Diagnostics and Instrumentation The diagnosis of vibrations in positive displacement pumps should usually include dynamic pressure measurements in the cylinders and piping near the pump. These measurements can be obtained by the use of piezoelectric or strain gage pressure transducers. If cavitation or flashing is suspected, a pressure transducer capable of measuring static and large dynamic pressures should be used; even then, cavitation-produced pressure shocks may damage the transducer.

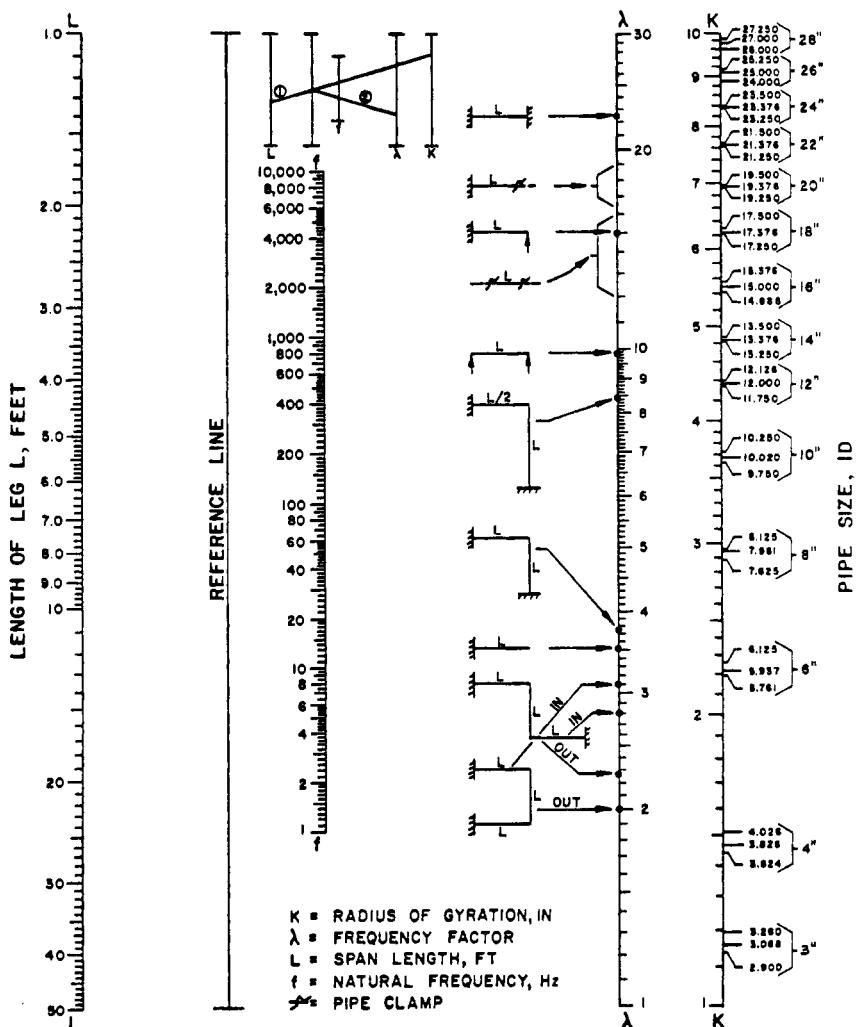


FIGURE 8 Natural frequency of uniform steel piping spans (1 ft = 0.3048 m; 1 in = 2.54 cm)

Accelerometers with low frequency characteristics may be used with electronic integrators to obtain accurate vibration displacement data from the pump case, cylinders, or piping. Similarly, velocity or seismic pickups may be employed. Maximum vibrations usually occur at the middle of piping spans and at unsupported elbows (out of plane).

Real-time analyzers and oscilloscopes may be used to display the resulting signals. A field example showing the diagnosis of cavitation at the pump suction using a strain gage diaphragm pressure transducer is given in the oscilloscope trace of Figure 9. The vapor pressure (gage) for this system was 25 lb/in² (172 kPa). Note that the negative half of the cycle is flattened when vapor pressure is reached and that very high amplitude pressure spikes are apparent. For liquids with dissolved gases, lower pulsation amplitudes can produce cavitation; however, the cavitation is usually less severe.

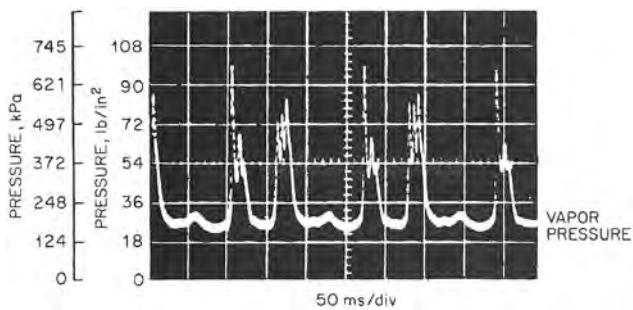


FIGURE 9 Complex wave data showing cavitation effects of pressure wave in a liquid piping system

Typical field pulsation data obtained on a three-plunger pump are shown in Figure 10, which is a frequency spectrum of the pulsations made by a real-time analyzer. Each spike represents a frequency multiple of running speed. Note that the third spike, representing the plunger frequency, has the largest amplitude; however, the components at one and two times pump speed are also significant, which means that these frequency components should be considered for pulsation control.

The vibration frequencies of the piping should be compared with pulsation frequencies to evaluate potential pulsation excitation of mechanical resonances. A check of the piping mechanical natural frequencies from the nomogram (refer to Figure 8) should be made to evaluate the possibility of a mechanical resonance. High vibrations produced by low-level pulsations at a particular frequency are indicative of a mechanical resonance, which can usually be corrected by additional piping restraints or snubbers. After the causes of the pulsations and vibrations are diagnosed, the techniques presented previously can be used to develop solutions.

Thermal Problems In pump systems with high thermal gradients, large forces and moments on the pump case can cause misalignment of the pump and its driver as well as pump case distortion resulting in vibrations, rubbing (wear), bearing failure, seal leakage, and so on. High stresses can be imposed on the piping, resulting in local yielding or damage to the piping restraints, snubbers, or support system. Misalignment problems commonly exhibit a high second-order component of the shaft vibrations. Proximity

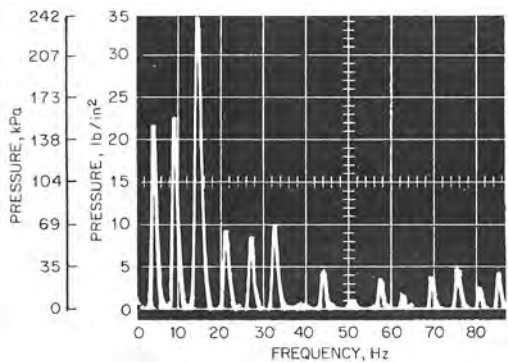


FIGURE 10 Typical field data recorded on a three-plunger pump

probes can be used at the bearings to measure movement of the shaft relative to the bearing centerline.

Diagnosis of Shaft Failures Pump and driver shafting can experience high stresses during start-up and normal operation because of the uneven torque loading of the positive displacement pumping action. Shaft failures are strongly influenced by the torsional resonances of the system, which are the angular natural frequencies of the system.

Torsional vibrations can be measured using velocity-type torsional transducers that mount on a stub shaft. Alternatively, they may be gauged by measuring the gear tooth passing frequency with a magnetic transducer or proximity probe and using frequency-to-voltage converters to give the change in tooth passing frequency (the torsional vibrational velocity). Spectral analysis of these signals defines the torsional amplitudes and natural frequencies. The stresses can be calculated by using the mode shape of the specific resonant natural frequency and combining all the torsional loads. Torsional natural frequencies, mode shapes, and stresses can be calculated by using either the Holzer technique or digital computer programs.¹²

Torsional problems can usually be solved by changing the coupling stiffness between the driver and pump or by using a flywheel in an effective location. The addition of a flywheel will tend to smooth the torque oscillations. Pumps with a greater number of cylinders and equal cylinder phasing usually operate more smoothly with lower shaft stresses.

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SECTION 3.5

DISPLACEMENT PUMP FLOW CONTROL

WILL SMITH

FLOW CONTROL IN INDIVIDUAL PUMPS

An inherent characteristic of positive displacement pumping of relatively incompressible liquids is that flow rate is proportional to displacement rate and independent of pressure levels. The capacity of a centrifugal pump operating at constant speed varies from a maximum flow at no developed pressure to zero flow at a definite limiting pressure known as shutoff head. The *average* capacity of positive displacement pumps at constant speed is, within design limits of pressure, practically constant, even though flow rate pulsations do occur as individual displacements are forced into the discharge pipe.

Flow control of positive displacement pumps is accomplished by

- Changing the displacement rate
- Changing the displacement volume
- Changing the proportion of the displacement delivered into the piping system

Throttle Control in Direct-Acting Steam Pumps Direct-acting pumps are controlled by speed change, which is effected by throttling the flow of steam (or other motive gas) to the drive cylinder. The magnitude of excess force on the drive piston over that required for the driven, or pumping, piston to force pumpage into the piping system dictates the stroke rate and therefore the capacity of the pump. A sensor that detects the desired result of pumping (pressure, level, flow rate, and so on) may be used to modulate the steam throttle valve.

Speed Control in Power-Driven Pumps Speed modulation is the most common means of flow control for power-driven pumps. The most rudimentary speed control is intermittent (start-stop) operation. The average capacity over relatively long time periods depends upon the percentage of time the pump operates at 100% versus the percentage of time it

operates at zero flow. Of course, consideration must be given to the frequency of starts because electric motors may overheat if there is insufficient time for cooling after the inrush of starting current.

When continuous modulation of flow is desired, means of varying the driver speed or the ratio of speed reduction between the driver and pump are employed.

DRIVER SPEED CONTROL

1. Multispeed constant-torque motors offer limited steps in speeds.
2. Variable-frequency motor controls permit a wide range of fully modulated speeds but, at present, are limited in maximum horsepower (Section 9.2.2).
3. Direct current and wound rotor alternating current motors permit modulation of speeds within definite limits of torque capability (Section 9.2.2).
4. Gasoline and diesel engines provides speed variation capabilities within the limits where driver torque is adequate to satisfy the constant-torque requirements of the pump (Section 9.1.3).
5. Steam or gas turbines can operate over a limited speed range within their particular output torque limits (Sections 9.1.2 and 9.1.5).

SPEED CHANGER CONTROL Because *most* positive displacement pumps operate at significantly lower speed than *most* drivers, a speed reduction unit is coupled between driver and pump. Capacity control by varying the ratio of the speed reduction is quite convenient. A number of methods are used.

1. *Variable-ratio belt drives* change the speed by changing the pitch diameter of driver and driven sheaves in response to capacity requirements.
2. *Hydraulic torque converters* (hydrodynamic drives) vary the speed by regulating the *amount* of active drive fluid in the coupling. Because these converters relate torque to speed, they are sensitive to the driven torque requirement change that may derive from system pressure changes (Section 9.2.3).
3. *Hydroviscous drives* vary the slip between input and output shafts by adjusting the *distance* between elements in a viscous fluid environment (Section 9.2.3).
4. *Eddy-current couplings* also vary the slip between driver and driven elements but do so by varying the strength of a magnetic field (Section 9.2.1).

Variable Displacement Flow Control Rather than by speed control (changes in the number of displacements in a given period of time), flow may be varied by changing the *volume* displaced per stroke.

Variable strokes are generally limited to small metering or “controlled volume” pumps. Although large-capacity variable-stroke pumps have been built, the complexity of the mechanism involved and the development of other effective and economical means of flow control have precluded their use.

Changing Delivery to the System Another means of capacity control is *not* to vary the pump capacity but to alter the amount of pumpage delivered to the system. There are two popular means of doing this.

SUCTION VALVE UNLOADING Suction valve unloaders cause the displacement to be ineffective during the strokes for which the unloaders are activated. They command either full or zero delivery while the pump is kept running. Thus, current inrush problems from frequent motor starts are avoided. Because discharge pressure is not developed when suction valves are unloaded, energy consumption is held to a minimum.

A pump valve is unloaded by mechanically preventing the valve plate from returning to its seat. If the suction valve is held away from its seat, liquid will ebb and flow through the valve from suction header to pump cylinder and from cylinder to header during the stroking of the plunger or piston.

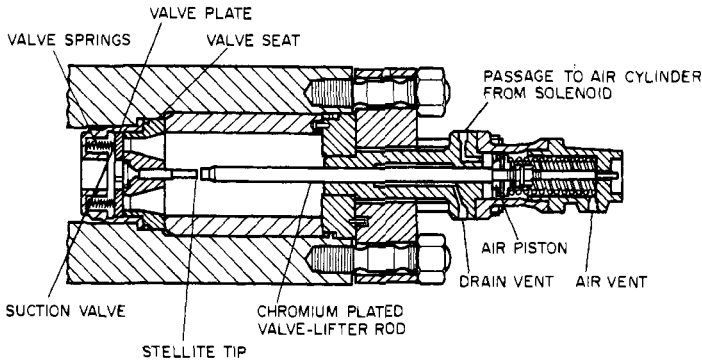


FIGURE 1 Synchronized unloading keeps the valve open through the use of a suction valve unloader

The unloader mechanism does not move the valve. The valve is lifted off its seat by the pressure differential created by normal pump operation. In synchronized unloading, the suction valve unloader (Figure 1) is retracted only when it is certain that the incoming flow will keep the valve open.

The unloader never resists pressure in the cylinder; it merely holds the valve plate against valve springs that had already been compressed and moves away from the valve while the springs are still compressed. This permits the valve to close when the plunger begins its discharge stroke.

For smooth transition between full flow and zero flow, the unloading and loading are accomplished in an immediate and identical sequence of the pumping order of the plungers. It is the function of the synchronized suction valve unloader distributor (Figure 2) to control unloading events in that proper and immediate sequence. Similarity can be drawn to the gasoline engine distributor in its function of assuring the proper and immediate sequence of firing of the spark plugs.

The distributor is driven directly by the crankshaft so its position is positively indexed to the stroking of the plungers. It provides an electrical signal to a solenoid valve, which

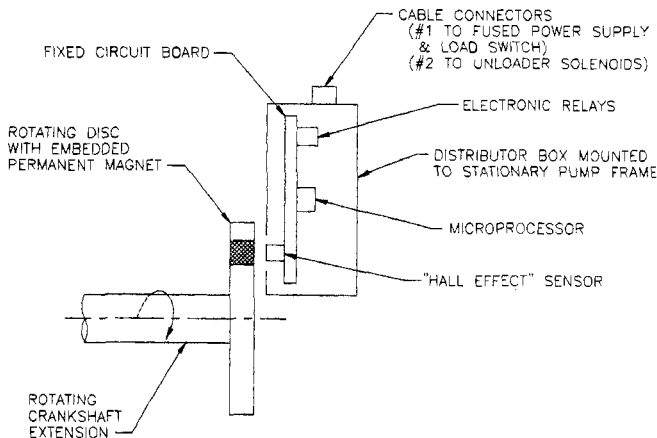


FIGURE 2 "Hall effect" microprocessor distributor system

admits motive fluid pressure to the unloader chamber when the suction valve to the particular cylinder has been opened. Then, in sequence, each of the remaining suction valves is retained so, in the first revolution of the pump after a control signals the distributor to initiate unloading, all suction valves are unloaded and the pump capacity is reduced to zero. When the control again signals for capacity, each suction valve is released, in sequence, during a single revolution of the crank.

While unloaded, the discharge valves do not function; they remain closed and serve as check valves against the pressurized discharge system. The suction valves remain open. Liquid enters from and is dispelled back into the suction manifold through the open suction valves. Therefore, no work is applied to the liquid, not even that required to open the suction and discharge valves. When the control calls for loading, the load is applied in steps equal to the number of plungers. Thus the pumping horsepower, as well as the liquid inertia in the pumping system, is changed in a number of steps equal to the number of plungers over the time span of one crank revolution.

The unloader consists of a spring-loaded pneumatic cylinder attached to the suction valve cover. The spring advances holding fingers that keep the valve away from its seat. Pneumatic pressure, applied through a solenoid valve in response to the distributor signal, creates a force on a piston that opposes the spring force and allows the suction valve to load (close). The system is normally arranged to fail in the unloaded position on either electrical or pneumatic pressure failure.

Distributor systems have evolved from a) direct mechanical linkages between reciprocating parts and the suction valves, to b) electromechanical interfacing, to c) electronic proximitors systems, to d) a "Hall effect" sensor signaling a microprocessor. The objective of each of these systems is to cause suction valve unloader solenoids to activate or deactivate in the appropriate sequence and at the appropriate position of the plunger during the suction stroke.

BYPASS CONTROL Ultimate delivery of pumpage to the system may be controlled by a discharge-to-suction bypass valve. Pump operation is continuous, with the possibility of modulating the flow to the system by regulating the portion "bypassed" to suction. A serious disadvantage is the full power utilization and full-load wear and tear even when at zero system capacity.

Evaluation of Flow Control Options Proper selection of the means of capacity control requires consideration of system control requirements and then evaluation of the options available to satisfy those requirements.

Precision, responsiveness, and degree of modulation required may limit options. First cost, maintenance requirements, and the effect on pump operating cost should then be weighed for viable options. To evaluate the effect of operating costs, it is necessary to determine (or reasonably assume) the expected pattern of operation throughout the capacity range. For example, if practically all operation except infrequent start-ups is to be at full pump capacity, little penalty should be assigned to the full power usage of a bypass valve or to the part-load losses inherent with hydroviscous or eddy-current drives.

If smooth modulation of flow is not essential, start-stop or suction valve unloading may be the preference. If smooth modulation *and* considerable operation at part capacity are necessary, variable-frequency motors or variable-pitch V-belt drives may be indicated.

Table 1 and Figure 3 provide some general guidelines. Each application should be evaluated on the merits of requirements and options.

FLOW CONTROL IN COMBINED DISPLACEMENT AND CENTRIFUGAL PUMPS

Systems that involve both centrifugal and reciprocating positive displacement pumps deserve some special consideration. Centrifugal pumps are often used as suction boosters to overcome acceleration head requirements peculiar to reciprocating pumps but are rarely used to supplement flow. Some unique characteristics of each type of pump that affect the other type must be considered in the design, operation, and control of the inter-related system.

TABLE 1 Comparison of several capacity control schemes for positive displacement pumps

Control Method	Degree of Modulation	First Cost	Operating Cost	Comments
<i>Directing-acting pumps</i>				
Steam throttle	Full: 0 to 100%	Low	Low	Steam pressure required to balance liquid piston force and overcome breakaway friction. Steam volume throttled to produce desired capacity.
<i>Power pumps</i>				
Start-stop	0 or 100%	Low	Low	Limited in frequency of starts because of temperature rise from inrush.
Multispeed motors	Steps dependent on motor winding	Medium	Low	Cost of motor controller switch gear motors must be assessed.
Variable frequency	Full: 0 to 100% +	High	Low	Limited by current-handling capacity of solid-state controller.
Direct current	Full: 0 to 100%	High	Medium	Drive is torque-speed sensitive, pump is torque-pressure sensitive at all speeds. Check drive for required torque at minimum and maximum speeds.
Wound rotor motors	Full: 0 to 100%	High	Low	Drive is torque-speed sensitive. Pump is torque-pressure sensitive at all speeds. Check drive for required torque at minimum and maximum speeds.
Combustion engines	Variable for limited range	High	Low	Torsional analysis required to avoid high torsional stresses.
Steam or gas turbines	Variable for limited range	Medium	Medium	Drive is torque-speed sensitive. Pump is torque-pressure sensitive at all speeds. Check drive for required torque at minimum and maximum speeds.
Hydraulic torque converter	Full: 0 to 100%	High	High	Low full speed efficiency.

TABLE 1 Comparison of several capacity control schemes for positive displacement pumps (Continued)

Control Method	Degree of Modulation	First Cost	Operating Cost	Comments
<i>Power pumps</i>				
Hydroviscous speed control	Full: 0 to 100%	High	High at part load	Efficiency proportional to capacity. Loss to heat exchanger.
Fluid coupling	Full: 0 to 100%	Medium	High	Full-speed slip loss greater than hydroviscous. Loss to heat exchanger.
Magnetic (eddy) coupling	Full: 0 to 100%	High	High	Full-speed slip loss.
Suction valve unloaders	0 or 100%	Low	Low	Synchronization with suction stroke avoids start-stop and shock problems.
Bypass valve	Limited	Low	High	Uses full power at zero capacity with full valve wear.
Variable-pitch belt	Limited	Low	Low	Limited to belt drive horsepower.

Pertinent Positive Displacement Pump Characteristics

- 1. The discharge pressure produced is a function of the system requirement only and is independent of pump capacity.
- 2. The flow rate pulses between maximum and minimum values for each revolution of the crankshaft.

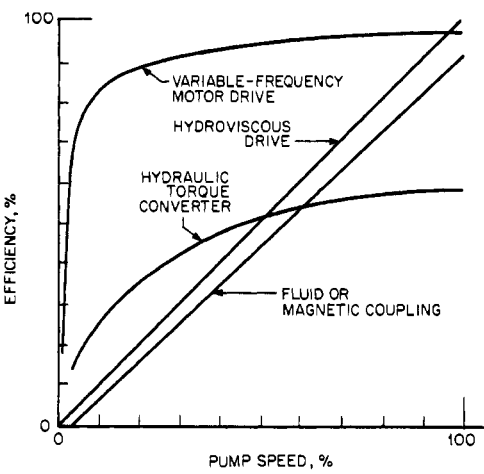


FIGURE 3 Relative efficiencies of speed control options

3. The pulsating flow imposes an acceleration head that adds to the net positive inlet pressure required (Section 3.1).
4. Being more energy-efficient, the positive displacement pump is normally the lead pump and the centrifugal pump is the supplement when operated in parallel.

Pertinent Centrifugal Pump Characteristics

1. The dynamic head (or pressure rise) produced is a function of pump capacity as well as of system requirements.
2. For satisfactory operation, the flow must be kept within a limited range of the best efficiency capacity (Section 2.1.3).
3. When used as a suction booster, the centrifugal pump must be designed so it cannot introduce air into the gas-intolerant reciprocating pump. (Gas in positive displacement pumps, like liquids in positive displacement compressors, may cause severe hydraulic and mechanical shock.)
4. Centrifugal pumps do not generate acceleration heads that impose on net positive suction head required. However, if a centrifugal pump is connected in parallel to a common suction line with a reciprocating pump, some of the system acceleration head loss from the positive displacement pump may affect the centrifugal pump.

Only two cases need to be considered: centrifugal pumps feeding (1) in series into reciprocating pumps to increase suction NPSHA to the reciprocating pump or (2) in parallel to augment the delivered capacity. It would be most unusual to encounter a positive displacement pump feeding into the suction of a centrifugal pump because of the high pressure that could be imposed on the centrifugal pump suction and because of the amplification of flow pulsations resulting from interaction of the characteristics of the two pumps, which could be deleterious to both pumps.

Series Operation, Suction Boost The flow rate is always determined by the positive displacement pump capacity and is independent of the centrifugal booster pump. The capacity is therefore controlled as with any positive displacement pump.

The purpose of the centrifugal pump is to supply sufficient pressure to satisfy the suction requirements of the positive displacement pump. The centrifugal pump must be sized to provide the maximum rate of flow of the positive displacement pump at a pressure high enough to meet the NPSH requirement of the latter.

With a flow-controlled positive displacement pump, speed control of the booster might be considered, as illustrated in Figure 4. The system pressure requirement can be satisfied by the positive displacement pump at all flow rates; therefore, total energy requirements may be reduced by using the less efficient centrifugal pump to develop no more than the required suction head.

If the centrifugal booster pump is run at constant speed, its share of the total head requirement will increase as system capacity is reduced.

Parallel Operation To increase the capacity of an existing pumping system, it is sometimes convenient to consider paralleling a positive displacement pump with a centrifugal pump. The interactions of the two pumps and the system must be considered and controlled for satisfactory operation.

Suction stabilizers and discharge dampeners, desirable for most reciprocating pump installations, become mandatory when a centrifugal pump and a positive displacement pump are connected in parallel. These devices are used to isolate, as much as possible, the effects of positive displacement pump flow pulsations from the sensitive centrifugal pump operating characteristics.

Centrifugal pumps in parallel with positive displacement pumps can supplement the system capacity only while they generate sufficient discharge pressure to satisfy the system requirement at the higher total flow. Undampened flow pulsations shift the centrifugal shutoff flow point relative to system flow at the frequency of the reciprocating

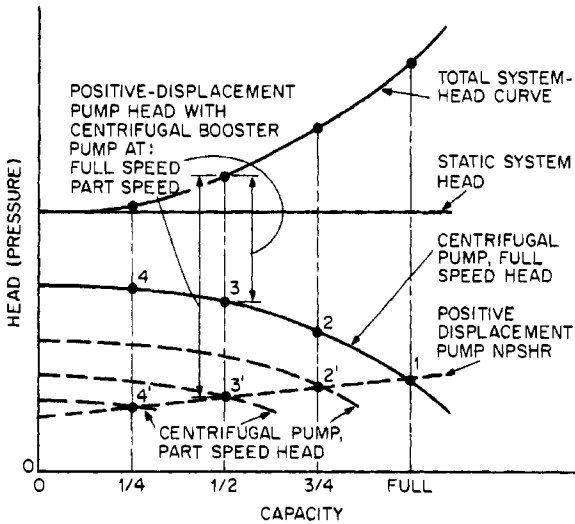


FIGURE 4 A centrifugal booster pump can satisfy the NPSH requirement of a positive displacement pump by operating either on its characteristic full speed curve 1-2-3-4 or with speed reduction along curve 1'-2'-3'-4'

pulsation. From the system standpoint, the total flow varies along curve 1-2-3-2-1 in Figure 5. From the centrifugal pump standpoint, it varies along its characteristic curve, 1'-2'-3'-2'-1'. The reciprocating pump operates along curve 1''-2''-3''-2''-1''.

For purely static system-head curves, the centrifugal pump theoretically would not “sense” the pulsating flow. For dynamic system heads, the centrifugal pump flow varies

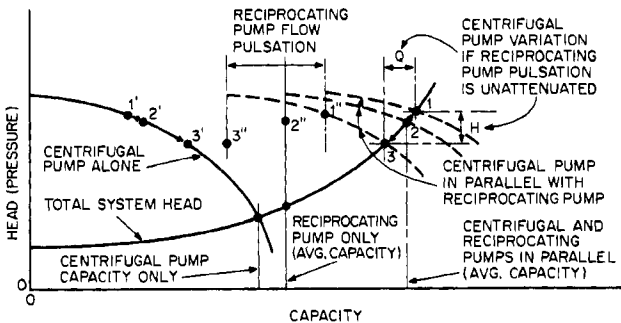


FIGURE 5 Parallel operation of undamped reciprocating and centrifugal pumps

1. The reciprocating pump will operate at a capacity corresponding to its speed regardless of its discharge pressure.
2. If the system head is only static (constant head), the centrifugal pump capacity is constant and the system flow variation corresponds to the reciprocating pump variation.
3. If the system head is partially dynamic (as illustrated), the centrifugal pump will experience flow variations.
 - a. The flatter the centrifugal pump curve, the greater the capacity and smaller the head variations on the centrifugal pump.
 - b. The flatter the system curve, the lower the capacity and greater the head variations on the centrifugal pump.

contracyclically with the reciprocating pump pulsations. The magnitude of the centrifugal pump flow variation due to the positive displacement pump pulsations is a function of the system head curve, the centrifugal pump head-capacity curve, the degree of flow pulse imposed on the centrifugal pump discharge header, and the liquid bulk modulus (or compressibility).

FLOW CONTROL IN POSITIVE DISPLACEMENT PUMPS IN SERIES _____

Whenever positive displacement pumps are installed in series (such as for multiple-station pipeline service), some variable-capacity capability is mandatory. Otherwise, any variation in capacity between series pumps or series groups of pumps would result in disastrous interstation pressures.

The flow is determined by the first pump in the line. Subsequent pumps should be equipped with capacity controls responsive to their inlet pressures. The preferred control is a continuous-modulation type, such as variable speed, unless large reservoir capacity is available between pumps.

Recirculation from discharge to suction through a throttling bypass valve may be used, at the expense of energy, to match the delivery from downstream pumps to upstream pumps. Excessive bypass flow in relation to through flow can result in significant temperature rise of the liquid.

When downstream pump capacity is less than the upstream delivery, the inter-pump pressure will rise, indicating a need for increased capacity. When the downstream pump capacity exceeds the upstream delivery, the downstream pump will "draw down" the inter-pump pressure to the point where net positive inlet pressure difficulties are ensured.

Reservoir accumulators between pumps permit a limited degree of mismatch in capacities, allowing the use of incremental capacity controls, such as suction valve unloading and intermittent pump bypass.

In any event, effective discharge dampeners on the upstream pump and suction stabilizers on the downstream pump are necessary to prevent the inherent pressure pulsations and control modulation variances from adversely affecting operation.

FLOW CONTROL IN POSITIVE DISPLACEMENT PUMPS IN PARALLEL _____

Because the capacity of positive displacement pumps is virtually independent of head, there is no problem from the head-capacity characteristics in parallel operation—even for pumps of different sizes.

Particular attention to suction acceleration head requirements is necessary when more than one pump is taking suction from a common suction pipe. Because it cannot be assured that two pumps will not, at some point, have simultaneous suction strokes, the acceleration head requirements of each pump must be added for that portion of the suction line that furnishes the total flow. This suggests extra large suction lines, individual ones from the liquid source, or suction stabilizers sized for the total flow with individual outlet connections for each pump.

SECTION 3.6

DIAPHRAGM PUMPS

STEPHEN D. ABLE
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Diaphragm pumps are a class of displacement pumps featuring flexible membranes in combination with check valves that are used to move fluids into and out of pumping chamber(s). These pumps are used extensively in transfer and metering applications requiring flows of up to 300 gallons per minute (1150 liters per minute). They are quite versatile, handling a wide variety of fluids including chemicals, dry powders, food additives, glues, paints, pharmaceutical products, slurries, tailings, and wastewater. A distinguishing feature of all diaphragm pumps is the absence of seals or packing, meaning they can be used in applications requiring zero leakage. There are three main categories of diaphragm pumps: 1) mechanically driven, 2) hydraulically driven, and 3) air-operated.

MECHANICALLY DRIVEN DIAPHRAGM PUMPS

Many industries are served by mechanically driven diaphragms pumps. They are used in construction, chemical, and water treatment applications.

Construction Industry Mechanically driven diaphragm pumps are widely used in the construction industry for dewatering applications where pumps may ingest rocks or other debris. A popular make of this type of pump contains a spring on the plunger rod (see Figures 1 and 2). If the operating pressure exceeds the maximum recommended pumping pressure, the spring compresses and does not move the diaphragm. The spring can compress and thus keep a rock from being pushed through the wall of the pumping chamber or cause the drive mechanism to fail.

In single-diaphragm pumps, the pumped liquid can have a lot of inertia if the suction and discharge lines are relatively long. A simple accumulator on the suction (inlet) side of the pump enables the pump to draw liquid from the accumulator while it simultaneously draws liquid through the suction line.

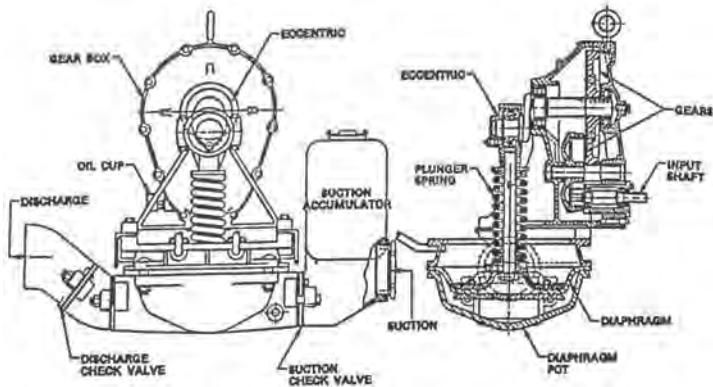


FIGURE 1 Cross-section of a mechanically driven single-diaphragm pump for the construction industry (Gorman-Rupp)

During the discharge stroke, the accumulator can refill with liquid from the suction line. If the discharge line from the pump is relatively long, the inertia of the liquid can be great, as mentioned earlier, and can impose severe loads on the diaphragm and drive mechanism. The spring on the plunger rod can absorb some of the drive energy early in the discharge stroke and “give it back” during the latter part of the discharge stroke, greatly reducing the inertia loading on the diaphragm and drive mechanism.

Mechanically driven diaphragm pumps in the construction industry operate by a reciprocating plunger, usually secured to plates on both sides of the diaphragm. The diaphragms are customarily fabric-reinforced elastomers (usually synthetic rubbers) similar in many ways to the fabric-reinforced materials used in pneumatic tires. The diaphragms are normally molded with a convoluted section between the central clamped area and the clamped periphery. This convoluted section permits longer strokes than would be possible otherwise.

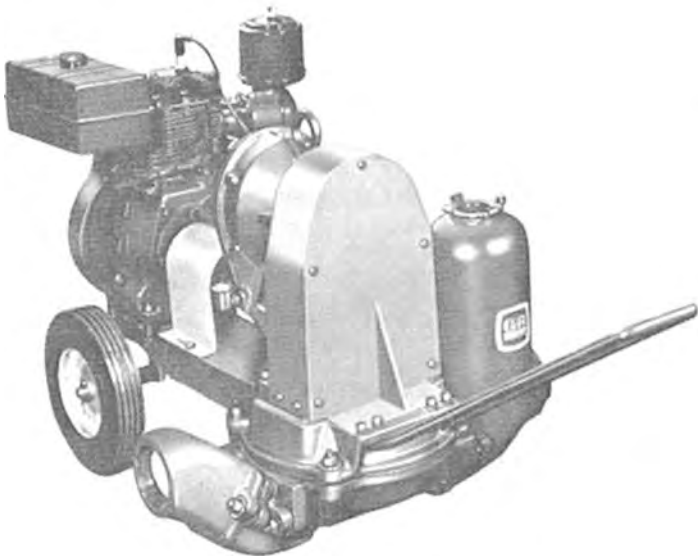


FIGURE 2 Mechanically driven single-diaphragm pump, engine-powered (Gorman-Rupp)

These pumps are sometimes duplexed so that the reciprocating means acts alternately on two diaphragms with one on a suction stroke, while the other is on a discharge stroke and vice versa. A connector called a *walking beam* is pivoted between two diaphragms. As one diaphragm is pushed down on a discharge stroke, the other diaphragm is simultaneously pulled up on a suction stroke. The pumping chambers with inlet and outlet check valves are manifolded together to a common inlet and a common outlet. The principle advantage of the duplex diaphragm pump is its more constant flow (two pressure pulsations per cycle).

Mechanically driven diaphragm pumps are used in the construction industry for dewatering foundations and cofferdams, as well as in sewage treatment plants for pumping lime slurries. They are normally limited to 50 ft (15.2 m) of differential head and are capable of suction lifts of as much as 25 ft (7.6 m).

Chemical and Water Treatment Industries Another type of mechanically driven diaphragm pump is used for the injection or transfer of chemicals into process streams at pressures up to 250 lb/in² (17 bar). These pumps are designed to enable the easy adjustment of their capacities, so precise volumes of chemicals can be injected. Typically, capacities can be adjusted through a 20:1 range. Injection repeatability is generally plus or minus 3%.

A wide range of chemicals can be handled. Wetted materials include PVC, PVDF, Polypropylene, 316SS, Alloy 20, and Alloy C22. Diaphragms are PTFE or PTFE with elastomeric backing. Ball type check valves are usually employed.

Applications for this type of pump include the injection of acids and bases for pH control, biocides, chlorination, coagulants, and fertilizers. There are two basic configurations for pumps in this class: *electromagnetic pumps* (solenoid) and *motor-driven pumps*.

Electromagnetic (electronic) pumps (see Figure 3) are used in a variety of low-power applications with flows from 0.026 to 26 gallons per hour (0.1 to 100 liters per hour) at pressures up to 250 lb/in² (17 bar). These metering pumps employ an electronic control circuit that pulses an electromagnet that, in turn, generates the linear motion of an armature-shaft-diaphragm assembly. Each electronic pulse results in one discharge stroke of the pump. At the end of the stroke, a set of springs returns the diaphragm assembly to its initial position, drawing more fluid into the pump chamber in preparation for the next stroke.

These pumps are inherently safe, as they can be run indefinitely in the stalled condition without damage to the pump or overpressuring most systems. An additional feature of certain electronic pumps is the regulation of pulse strength through electronic power control, which leads to smoother fluid injection. Capacity is usually controlled by adjusting

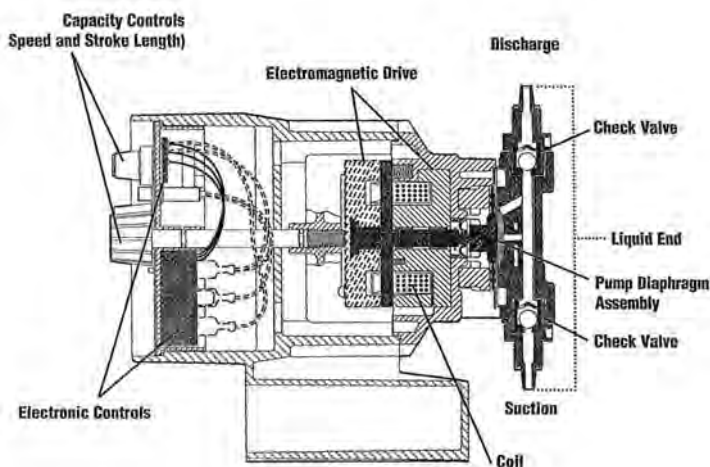


FIGURE 3 Electromagnetic diaphragm pump (Milton-Roy, subsidiary of Sundstrand Corp.)

the stroke rate, but the stroke length can also be adjusted. Combining these adjustments provides a wide range of outputs.

Advances in the electronic controls have led to the capability to control output manually from 4 to 20 mA process signals, digital pulses from external sources (such as flow meters), or serial data communications signals from computers. Yearly maintenance is recommended for low-pressure applications, but as pressures rise, diaphragms and check valves will need to be replaced more frequently.

Motor-driven, mechanically actuated diaphragm pumps are used in applications for flows from 2 to 300 gallons per hour (approximately 10 to 1000 liters per hour) again at pressures up to 250 lb/in² (17 bars). Since motor-driven, mechanically actuated diaphragm pumps are positive displacement pumps, capacities cannot be adjusted by the use of a throttling valve.

Three techniques are used in the industry to adjust motor-driven pump capacities. Some designs employ the use of a "mechanical lost motion" mechanism in which an adjustable mechanical stop interrupts the spring-loaded crosshead from following the cam for a portion of the stroke. This decreases the effective stroke length (see Figure 4). Other designs employ the use of an adjustable crank machined with a "variable eccentricity." The adjustment of the crank's position changes the pump's effective stroke length (see Figure 5). The third design technique is the use of AC or DC variable speed motors.

At higher flow rates, attention should be paid to the system design to ensure the proper handling of pressure pulsations, due to the acceleration and deceleration of the process fluid in the lines. This is especially true for pumps having a mechanical lost motion configuration, due to the diaphragm's rapid starts and stops. In the higher flow systems, pulsation dampeners are frequently employed in the discharge lines and occasionally need to be used in the suction lines to control pulsations. When discharge pressures are low, backpressure valves are employed to provide a system pressure sufficient to decelerate the fluid in the suction line at the end of every suction stroke. If the flow of the process fluid in the suction line has not stopped by the beginning of the discharge stroke, the accuracy of the injection is diminished. Yearly maintenance is recommended for low-pressure applications, but as pressures rise, diaphragms and check valves will need to be replaced more frequently.

Other Mechanically Driven Diaphragm Pumps Automotive fuel pumps are usually of the diaphragm type. The diaphragm is moved mechanically by a cam on a suction stroke

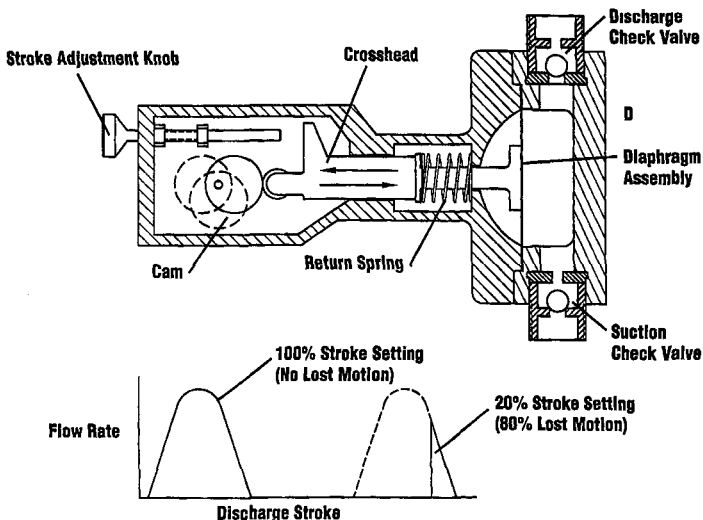


FIGURE 4 Diaphragm of a typical mechanical lost motion mechanism (Milton-Roy, subsidiary of Sundstrand Corp.)

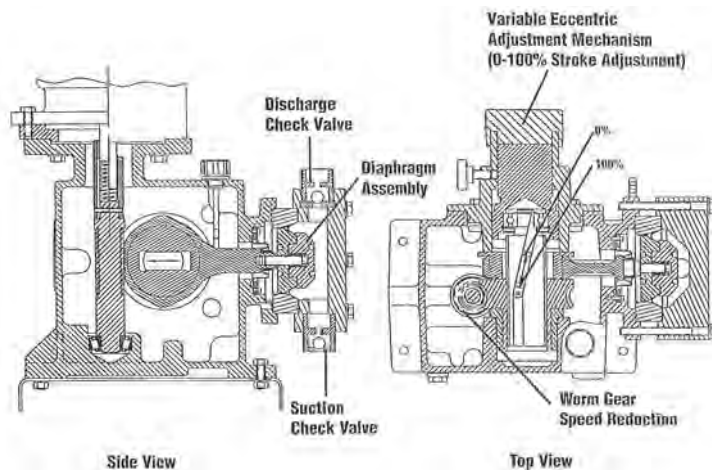


FIGURE 5 Cross-section of a variable eccentric mechanically driven diaphragm pump (Milton-Roy, subsidiary of Sundstrand Corp.)

and is returned by a spring for the discharge stroke. Thus, the spring determines the discharge pressure and provides a nearly constant pump pressure to the carburetor regardless of the pump (engine) speed or the rate of fuel consumption. Hand-operated diaphragm pumps are used extensively as bilge pumps on sailboats where the loss of power is a major concern and hand operation is essential.

HYDRAULICALLY DRIVEN DIAPHRAGM PUMPS

Hydraulically driven diaphragm pumps are used in applications for the transfer or injection of chemicals into process streams at pressures up to 7500 lb/in² (approximately 500 bar). Because the diaphragm is pressure-balanced, the stresses in the diaphragms are low. Therefore, these pumps tend to require minimal maintenance. The pump's capacities can be adjusted to match the specific process requirement by adjusting the effective stroke length or stroking speed of the pump. Effective stroke lengths are adjusted by either a hydraulic lost motion, a mechanical lost motion, or by varying the eccentric's offset. The repeatability of the injected flow is plus or minus 1% or better.

Applications range from 0.26 to 26,000 gallons per hour (1 to 100,000 liters per hour). At flows above 26 gallons per hour (100 liters per hour), most pump models employ capacity adjustments based on variable eccentric or variable speed technology to avoid significant pressure spikes due to the rapid acceleration and deceleration of the fluid in the pipes.

As with the mechanical diaphragm pumps, a wide range of chemicals can be handled. Wetted materials include PVC, Polypropylene, PVDF, 316 SS, Alloy 20, Alloy C-22, Titanium, and Inconel. Diaphragms for pressures up to 4350 lb/in² (300 bar) are typically composed of PTFE or PTFE with an elastomeric backing. Diaphragms above 4350 lb/in² (300 bar) are typically 316 SS, Alloy C, or PEEK. Optional features include fluid temperature control jackets, diaphragm rupture detection capabilities, and remote diaphragm head designs. Typical applications include the injection of acids and bases for pH control, corrosion inhibitors, methanol, coagulants, primary process blending, process slurries, and drag reducers. Three types of liquid ends are used: the disc diaphragm, shown in Figure 6, the tubular diaphragm in Figure 7, and the high performance diaphragm in Figure 8.

The disc diaphragm pump is equipped with process-side and suction-side restraining plates to prevent overdisplacement of the diaphragm during system upsets. When the

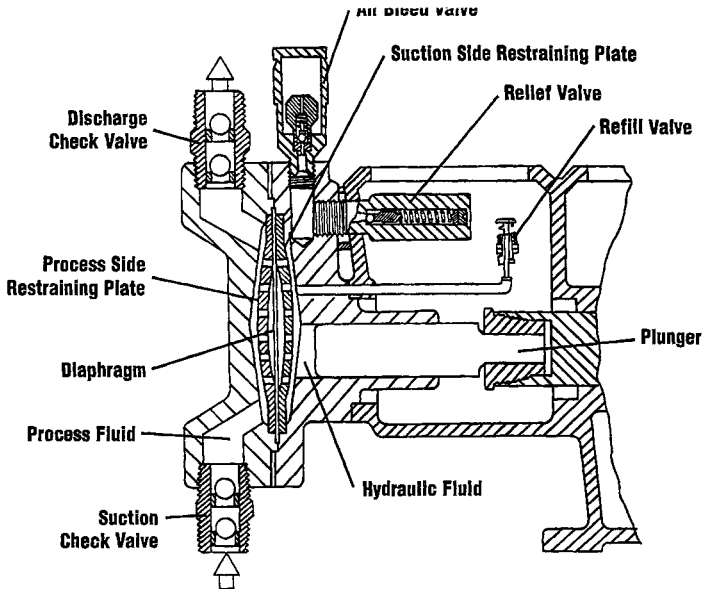


FIGURE 6 Diagram of a disc diaphragm pump (Milton-Roy, subsidiary of Sundstrand Corp.)

diaphragm reaches the suction-side restraining plate, the hydraulic oil pressure drops, causing the refill valve to open and replenish the oil. When the diaphragm hits the process-side restraining plate, the hydraulic pressure rises, causing the relief valve to open, venting some oil. The fluid volume between the restraining plates is typically 150% of the maximum displaced volume of the pump. Therefore, the diaphragm does not contact both restraining plates during the same stroke.

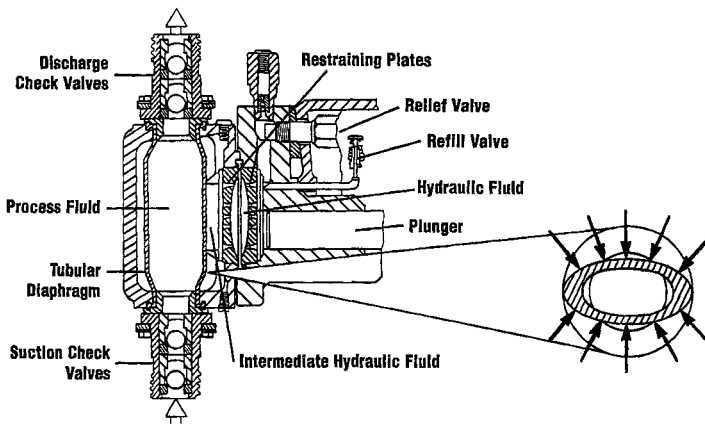


FIGURE 7 Diagram of a tubular diaphragm configuration (Milton-Roy, subsidiary of Sundstrand Corp.)

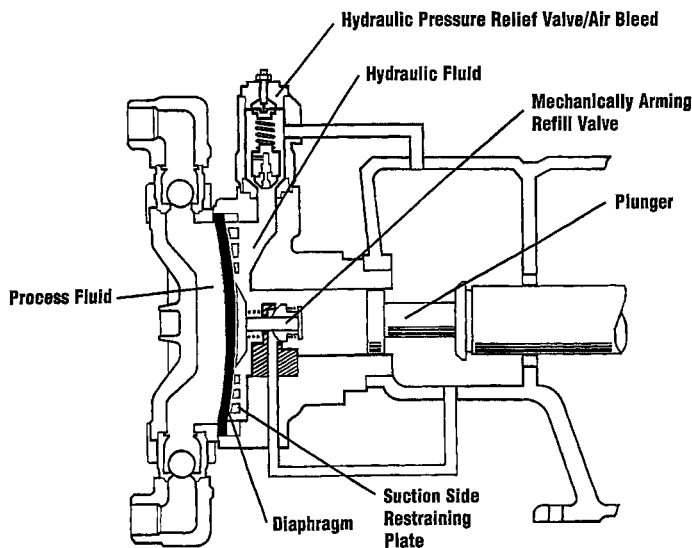


FIGURE 8 Diagram of a high performance diaphragm design (Milton-Roy, subsidiary of Sundstrand Corp.)

The tubular diaphragm configuration is a variation of the disc diaphragm design. A diaphragm shaped in the form of a tube is placed in a chamber in front of the disc diaphragm assembly. This design eliminates the process fluid flowing through the front-restraining plate, reducing viscous losses and wear in case of slurries. The chamber must be filled with a precise amount of hydraulic fluid to avoid overdisplacing the tube.

The high-performance diaphragm configuration eliminates the use of a process-side restraining plate providing the throughflow performance of a tubular design while eliminating the possibility of overdisplacing the tube during startup and maintenance. With a mechanically arming, pressure-sensitive refill valve, the hydraulic fluid can only be replenished when the diaphragm is in the most rearward position. This eliminates the possibility of overfilling the hydraulic chamber and therefore overdisplacing the diaphragm during system upsets (blocked suction or discharge lines).

Most problems with hydraulic diaphragm pumps occur due to incorrect system designs. Pressures above 9 lb/in² (0.6 bar) should be maintained in the pump diaphragm heads during the suction stroke to stop vapor buildups in the hydraulic or process-side cavities. Pressures at 3 lb/in² (0.2 bar) can be handled in some applications with modified designs and special hydraulic fluids. NPSH calculations should include viscous losses in the check valves and contour plates (if so equipped).

In addition, since hydraulic diaphragm pumps are reciprocating machines, acceleration losses also have to be considered. Peak acceleration/deceleration losses occur at the beginning and end of the stroke, while peak viscous losses occur at midstroke. The losses are not additive. The manufacturer should be contacted to provide guidance in performing these calculations.

As with mechanically driven diaphragm pumps, at higher flows, pulsation dampeners should be considered to ensure the proper handling of pressure pulsations due to the acceleration and deceleration of the process fluid in the lines. This is especially true for pumps having a mechanical lost motion configuration due to the diaphragm's rapid starts and stops. When discharge pressures are low, backpressure valves are employed to provide a system pressure sufficient to decelerate the fluid in the suction line at the end of every suction stroke. If the flow of the process fluid in the suction line has not stopped by the beginning of the discharge stroke, the accuracy of the injection is compromised.

AIR-OPERATED DIAPHRAGM PUMPS (AODPS)

In general, diaphragm pumps of all types are sealless, have no dynamic seals or packing, are self-priming, and have an infinitely variable flow rate and pressure rate within the pressure and capacity ranges of the pump. *Air-operated diaphragm pumps* (AODPs) can also run dry indefinitely, and the discharge can be throttled to zero flow indefinitely.

The most common types of AODPs are the double-diaphragm pumps (duplex pumps). These contain two diaphragm chambers and two flexible diaphragms. The diaphragms are connected to each other through a connecting rod and are clamped at the outer edges of the diaphragm. The shaft-connected diaphragms move in the same linear direction simultaneously. Compressed air directed to the back side of the left diaphragm moves both diaphragms to the left, while air is exhausted to the atmosphere from the back side of the right diaphragm. After completion of a stroke, an air distribution valve directs compressed air from the supply to the back side of the right diaphragm and exhausts air to the atmosphere from the left chamber. This continuous reciprocating motion, along with properly operating internal check valves, creates an alternating intake and discharge of pumped liquid into and out of each chamber that results in a nearly continuous pumping action from the combined chambers.

A diaphragm pump air motor contains an air distribution valve that shifts positions at the end of each stroke of the pump. The air distribution valve alternately directs supply air pressure to one chamber and exhausts the other. Air motors often use a two-stage valve to control the reciprocating motion of the pump (see Figure 11). The pilot valve supplies a pilot air pressure signal to the air distribution valve throughout the entire stroke of the pump, even though pressure oscillations in the pumping system may occur. The pilot valve is not connected directly to the diaphragm's connecting rod, which provides a "deadband" to prevent the power valve from erroneously shifting just after the end of each stroke. The two views contained in Figure 11 depict the position of the moving parts just before the pilot valve is moved by its contact with the diaphragm washer. Depending on its position, the pilot rod alternately pressurizes and exhausts the large end of the air distribution valve. Other valve design configurations pressurize and exhaust both ends of the air distribution valve.

The two common types of liquid check valves that are used in a diaphragm pump are the *flap valve* and the *ball valve*. A flap valve pump (see Figures 9 and 10) can handle nearly marble-sized solids. Because the discharge is from the bottom of the diaphragm chambers, the pump is ideally suited for pumping solids in suspension that may tend to settle out, particularly when the pumping rate is reduced or when the pump is shut down. The bottom outlet enables foreign matter to be easily pumped out of the chambers.

The popular air-operated, double-diaphragm pump with ball valves (see Figures 12 and 13) features the inlet at the bottom of the diaphragm chambers, and the outlet is at the top. The top discharge arrangement has the advantage of enabling air or vapors to be easily expelled from the chambers. Trapped air or vapors in pumps having bottom outlets can reduce the volumetric displacement of the pumps as the air or vapor is alternately compressed and expanded, instead of the liquid being displaced. This can be a concern in low-flow applications requiring relatively high pumping pressure and that handle viscous liquids. In higher flow applications, a sufficient turbulence is present and air or vapors mix with the pumped liquid to purge the pumping chambers of the gases.

The performance chart of a typical 2-in (51-mm) air-operated, double-diaphragm pump (see Figure 14) is similar to that of other pump types but contains air consumption rather than horsepower consumption. With a constant supply pressure of compressed air, work and energy relationships in the air chambers, as well as liquid flow losses within the pump, result in a downward sloping head-capacity curve, similar to a centrifugal pump. The following are noteworthy features of air-operated diaphragm pumps:

- With the pump shut off, there is no power consumption. Air consumption is approximately proportional to the flow rate; there is zero air consumption at a zero flow rate and maximum air consumption at a maximum flow rate. This feature enables diaphragm pumps to be used in applications where the flow rate must be varied over a wide range or from no flow to a high flow rate.

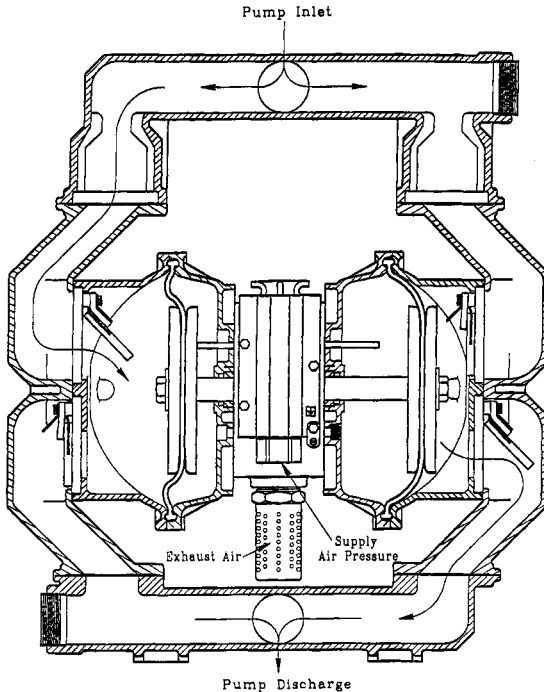


FIGURE 9 Cross-section of an air-operated double-diaphragm pump with flap check valves (Ingersoll-Rand Fluid Products)

- The pump discharge pressure remains the same for a given capacity and air inlet pressure regardless of the specific gravity of the liquid being pumped. The discharge head varies with the specific gravity of the pumped fluid, because pressure and not head is primarily maintained by the air pressure. For centrifugal pumps, the discharge pressure is directly proportional to the specific gravity of the liquid being pumped, while head is fixed at a given point on the performance curve. The supplied air pressure, pump flow rate, and NPSH set the discharge pressure for a given air operated pump, not the head, which is a characteristic of a centrifugal pump operated at a fixed speed.
- The NPSH required for an air-operated diaphragm pump is defined in the same manner as with any reciprocating pump. The NPSH required is determined to occur when the volumetric efficiency drops by a measurable amount (normally 3%) at a fixed speed or flow rate.

MATERIALS OF AIR-OPERATED DIAPHRAGM PUMPS

Check valves for diaphragm pumps are of three types: flap, ball, and poppet (see Figure 15). Flap valves preferably hang in a vertical position with a horizontal flow through the valves. Their principle advantage is the capability to handle large objects in suspension. Flap valves with elastomeric hinges are extensively used. Ball valves are arranged to provide a vertical flow through the valve seats. Poppet valves are usually guided by a valve stem and are spring-loaded. The poppet type of valve is not position-sensitive and can

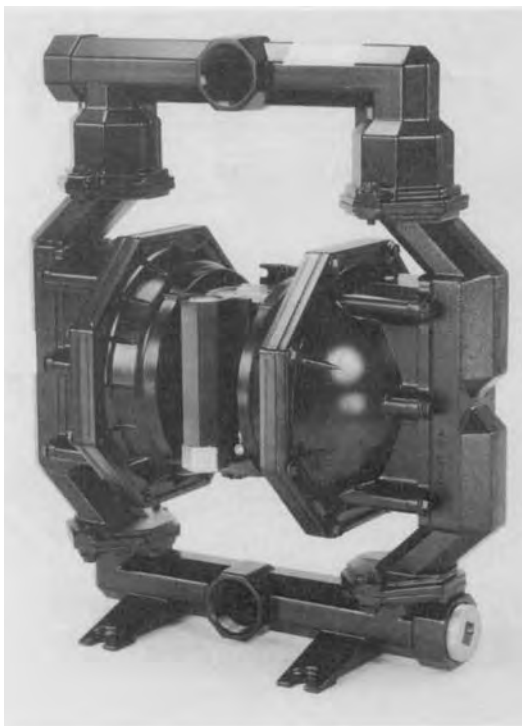


FIGURE 10 ARO air-operated double-diaphragm pump with flap check valves (Ingersoll-Rand Fluid Products)

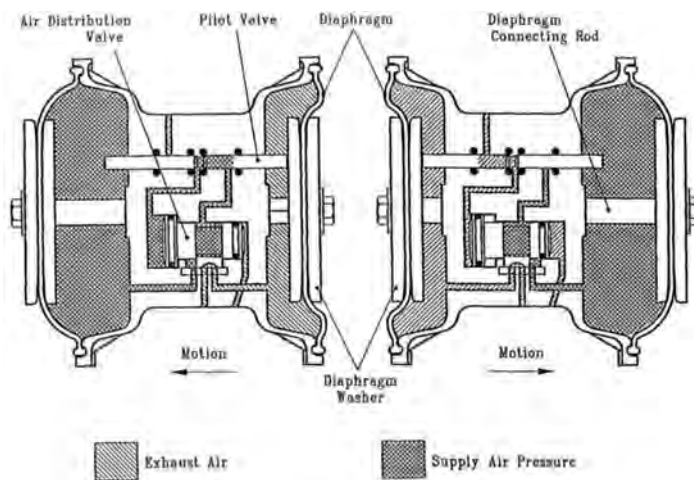


FIGURE 11 Cross-section of an air-operated double-diaphragm pump air motor operation (Ingersoll-Rand Fluid Products)

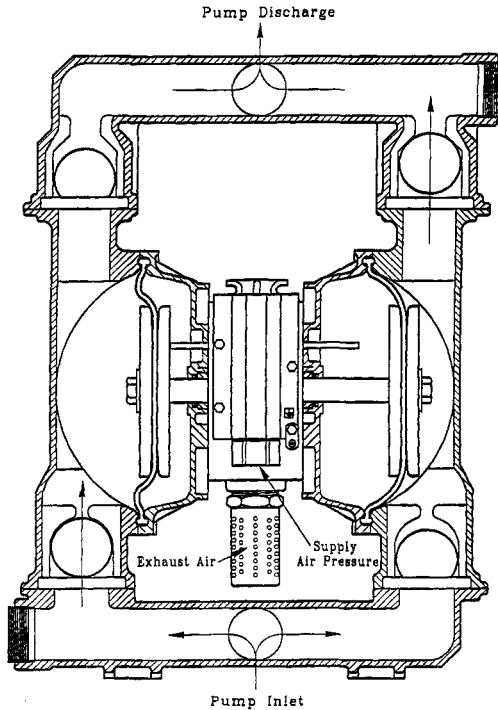


FIGURE 12 Cross-section of an air-operated double-diaphragm pump with ball check valves (Ingersoll-Rand Fluid Products)

operate in any orientation. Since many diaphragm pumps are used to pump abrasive slurries, the valves usually have elastomeric faces or are elastomeric balls. Large ball check valves may have metal cores covered by a thick wall of synthetic rubber. They may also be made of solid rubber, a thermoplastic elastomer, or Teflon®.

Diaphragms are customarily made of fabric-reinforced synthetic rubber, thermoplastic elastomers, or PTFE. Diaphragm materials include most of the synthetic rubbers: neoprene, Buna N, EP, Viton®, thermoplastic elastomers such as Hytrel® and Santoprene® as well as Teflon®. Many combinations of pump and diaphragm materials can cover a wide range of pumped products. For some solvents and aggressive acids or alkalis, Teflon® diaphragms can be used either directly or as an overlay on the conventional diaphragm.

The diaphragms in air-operated, double-diaphragm pumps are essentially balanced and act simply as membranes separating the compressed air from the product being pumped. The only unbalance occurs during the suction stroke of one diaphragm while it is being pulled by the shaft-connected other diaphragm. When the suction lift is minimal, the unbalance may be negligible.

Pump case materials include cast iron, aluminum, stainless steel, Carpenter 20, Hastelloy C, and plastics reinforced with glass or metal fibers.

APPLICATIONS OF AIR-OPERATED DIAPHRAGM PUMPS

General Diaphragm pumps are ideally suited for handling abrasive slurries because 1) the liquid velocity through the check valves and pumping chambers does not exceed the



FIGURE 13 ARO air-operated double-diaphragm pump with ball check valves (Ingersoll-Rand Fluid Products)

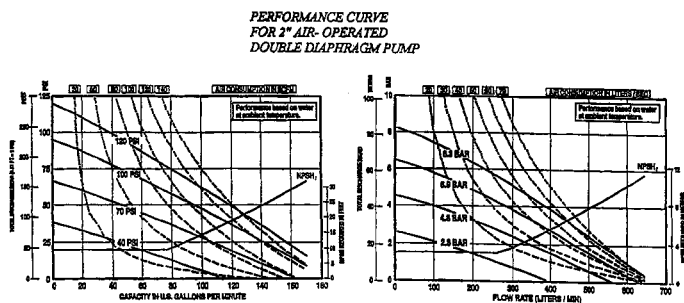


FIGURE 14 Performance curve for a 2-in air-operated double-diaphragm pump

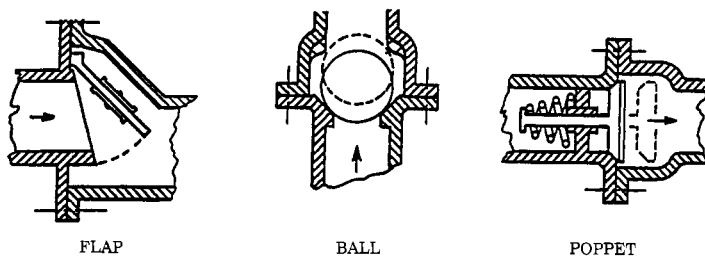


FIGURE 15 Three basic types of check valves

pipeline velocity and 2) the scouring and abrasions from the slurry are minimal. Because no close-fit sliding or rubbing parts exist and velocities are low, these pumps can be used for liquids with viscosities up to 50,000 SSU (11,000 cSt). Because the turbulence and mixing are minimal, they are ideally suited for shear-sensitive materials, such as latex.

Pumping Dry Powders Since diaphragm pumps can pump air as well as liquids, they are used successfully to pump dry powders. The air acts as a fluid medium for the powders in suspension, and the pump moves the air containing the suspended powder. Sometimes it is necessary to inject air into the powder to lower the apparent specific gravity and to get the powder into suspension.

Pump Controls An air pressure regulator in the compressed air supply line can control the pumping pressure. An air line valve can control the pumping rate. Thus, the pressure and capacity are easily controlled. Although air-operated diaphragm pumps are displacement pumps, they are not positive ones. The maximum pumping pressure cannot exceed the pressure of the compressed air powering the pumps. Table 1 describes the control of diaphragm pumps with the features inherent with each drive type.

Liquids Handled and Applications A variety of liquids can be handled by AODPs, yet the compatibility of pump case materials, valve materials, and diaphragms is the basic application limitation. Liquids and slurries handled by diaphragm pumps include ceramic slurry, paint, cement grout, chemicals, glue, resins, petroleum products, driller's mud, mill scale, ore concentrates, printer's ink, sewage, filter aids, latex, waste oils, wood preservatives, core washes, asphaltic coatings, bilge waste, radioactive waste, lapping compounds, porcelain frit, mine tailings, volatile solvents, coolants with metal fines, varnish, acids, coatings, soapstone slurries, explosives, lime slurries, yeast, chocolate, and wine. Typical applications of AODPs include tank and container loading and unloading, fluid filtration, spray painting, adhesive application, process mixing and batching, cutting oil, machine coolant, lubrication, sump pumping, dewatering, and waste water treatment.

LIMITATIONS AND ADVANTAGES OF AIR-OPERATED DIAPHRAGM PUMPS

This section outlines the pros and cons of AODPS. Table 1 also details the controls of the different types of pumps. The limitations of AODPs are as follows:

TABLE 1 Control of diaphragm pumps

Feature	Electro-Magnetically Driven	Mechanically Driven	Hydraulically Driven	Air-Operated
Can run dry:	Yes	Yes	Yes	Yes
Self-priming:	Yes	Yes	Yes	Yes
Discharge can be shutoff without damage:	Yes	No	Yes	Yes
Relief valve required:	No	Yes	Integral to pump	No
Flow controlled by:	Motor speed or stroke-length adjustment	Motor speed or stroke-length adjustment	Motor speed or stroke-length adjustment	Motor speed or discharge valve

- AODPs are not practical for pumping rates above about 300 gpm (1150 ℓ/m).
- AODPs are not manufactured for operating air pressures above 125 lb/in² gauge (8.6 bar). However, some versions are available that increase the pressure ratio by 2:1 or 3:1.
- Ice formation in air motors can occur (basic to the physics of operation), but the effect can be minimized by proper application and design.
- Diaphragms have a finite life. Fluids with abrasives and higher process temperatures can limit a diaphragm's life, but many material choices are available, including Teflon®, and several thermoplastic elastomers.

An AODP's advantages are as follows:

- They are self-priming from a dry start.
- The pumps have an infinitely variable flow rate and pressure-within-pressure and capacity ranges.
- AODPs have no dynamic seals or packing.
- They can run dry indefinitely.
- The discharge can be throttled to zero flow indefinitely.
- No air is used when the AODP is deadheaded. Electrically driven pumps will consume a significant portion of the rated power when no demand flow exists.
- AODPs are suited for use in hazardous environments (no electric power required).
- Power is used in proportion to the pumping rate.
- They can be used in confined areas without heat buildup.
- AODPs can pump abrasive slurries and solids in suspension.
- The pumps can handle viscous liquids up to 50,000 SSU (11,000 cSt).
- A minimal degradation of the viscosity of shear-sensitive materials occurs.
- They can pump dry powders in air suspension.
- No close-fit, sliding, or rotating parts come in contact with liquid.
- No bypass is required as in other displacement pumps.
- When properly maintained, there is zero leakage.
- They are simple to maintain and to repair.
- There is no bedplate and no coupling to align.
- AODPs can be used in handling aggressive chemical solutions.
- They can handle a wider range of materials than any other type of pump.

SECTION 3.7

SCREW PUMPS

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Screw pumps are a special type of rotary positive displacement pump in which the flow through the pumping elements is truly axial. The liquid is carried between screw threads on one or more rotors and is displaced axially as the screws rotate and mesh (see Figure 1). In all other rotary pumps, the liquid is forced to travel circumferentially, thus giving the screw pump with its unique axial flow pattern and low internal velocities a number of advantages in many applications where liquid agitation or churning is objectionable.

The applications of screw pumps cover a diversified range of markets including navy, marine, and utilities fuel oil services; marine cargo; industrial oil burners; lubricating oil services; chemical processes; petroleum and crude oil industries; power hydraulics for navy and machine tools; and many others. The screw pump can handle liquids in a range of viscosities, from molasses to gasoline, as well as synthetic liquids in a pressure range from 50 to 5000 lb/in² (3.5 to 350 bar) and flows up to 8000 gal/min (1820 m³/h).

Because of the relatively low inertia of their rotating parts, screw pumps are capable of operating at higher speeds than other rotary or reciprocating pumps of comparable displacement. Some turbine-attached lubricating oil pumps operate at 10,000 rpm and even higher. Screw pumps, like other rotary positive displacement pumps, are self-priming and have a delivery flow characteristic, which is essentially independent of pressure, provided there is sufficient viscosity in the liquid being pumped.

Screw pumps are generally classified into single- or multiple-rotor types. The latter is further divided into timed and untimed categories.

The single-screw or progressive cavity pump (see Figure 2) has a rotor thread that is eccentric to the axis of rotation and meshes with internal threads of the stator (rotor housing or body). Alternatively, the stator is made to wobble along the pump centerline.

Multiple-screw pumps are available in a variety of configurations and designs. All employ one driven rotor in a mesh and one or more sealing rotors. Several manufacturers have two basic configurations available: single-end (in Figure 3) and double-end (in Figure 4) construction, of which the latter is the better known.

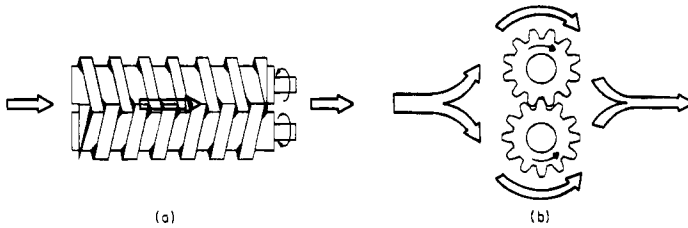


FIGURE 1 Diagrams of screw and gear elements, showing (a) axial and (b) circumferential flow

As with every pump type, certain advantages and disadvantages can be found in a screw pump design. These should be recognized when selecting the best pump for a particular application. The **advantages** of a screw pump design are as follows:

- A wide range of flows and pressures
- A wide range of liquids and viscosities
- High speed capability, allowing the freedom of driver selection
- Low internal velocities
- Self-priming, with good suction characteristics
- A high tolerance for entrained air and other gases
- Low velocities for minimum churning or foaming
- Low mechanical vibration, pulsation-free flow, and quiet operation
- A rugged, compact design that is easy to install and maintain
- High tolerance to contamination in comparison with other rotary pumps

The **disadvantages** are as follows:

- A relatively high cost because of close tolerances and running clearances
- Performance characteristics sensitive to viscosity changes
- High pressure capability may require long pumping elements
- Four sealing points between rotor and casing for timed two-screw design

THEORY

In screw pumps, it is the intermeshing of the threads on the rotors and the close fit of the surrounding housing that creates one or more sets of moving seals in a series between the pump inlet and outlet. These sets of seals or locks, as they are sometimes referred to, act

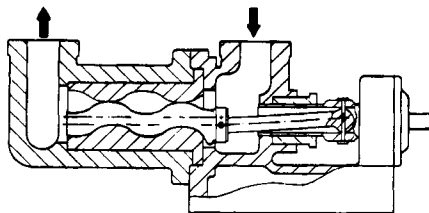


FIGURE 2 The single-screw or progressive cavity pump

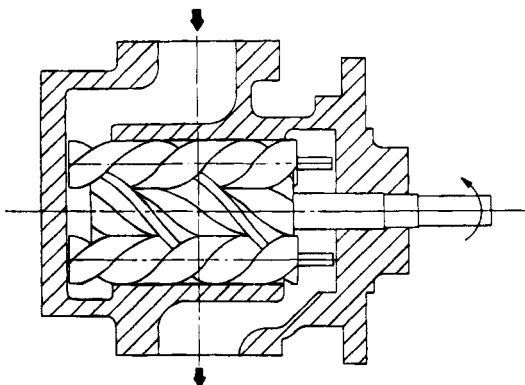


FIGURE 3 Multiple-screw single-end arrangement

as a labyrinth and provide the screw pump with its positive pressure capability. The successive sets of seals form fully enclosed cavities (see Figure 5) that move continuously from inlet to outlet. These cavities trap liquid at the inlet and carry it along to the outlet, providing a smooth flow.

Delivery Because the screw pump is a positive displacement device, it will deliver a definite quantity of liquid with every revolution of the rotors. This delivery can be defined in terms of displacement volume V_D , which is the theoretical volume displaced per revolution of the rotors and is dependent only upon the physical dimensions of the rotors. It is generally measured in cubic inches (cubic millimeters) per revolution. This delivery can also be defined in terms of theoretical capacity or flow rate Q_t , measured in U.S. gallons per minute (cubic meters per hour), which is a function of displacement and speed N in rpm:

In USC units:
$$Q_t = \frac{V_D N}{231}$$

In SI units:
$$Q_t = 6 \times 10^{-8} V_D N$$

If no internal clearances existed, the pump's actual delivered or net flow rate Q would equal the theoretical flow rate. Clearances, however, do exist with the result that whenever a pressure differential occurs, there will always be internal leakage from outlet to inlet. This leakage, commonly called *slip* S , varies depending upon the pump type or model, the amount of clearance, the liquid viscosity at the pumping conditions, and the

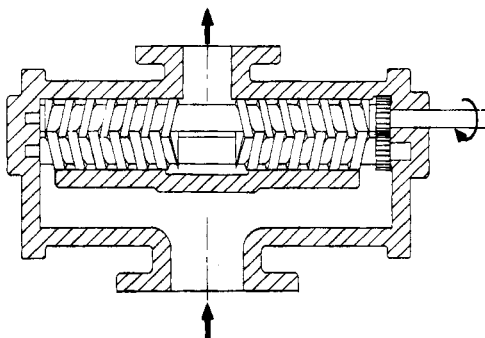


FIGURE 4 Multiple-screw double-end arrangement

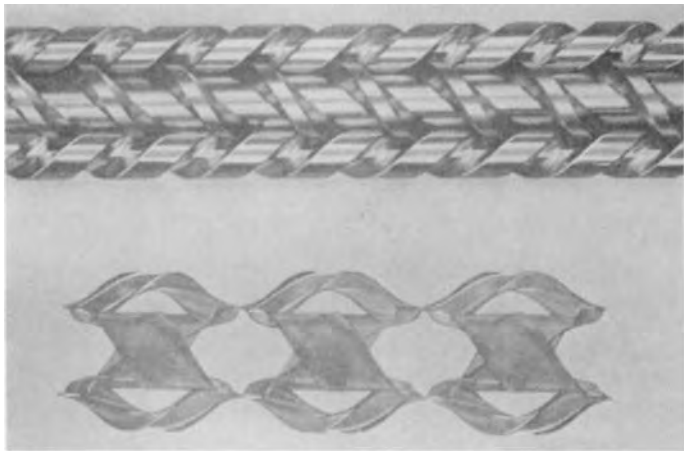


FIGURE 5 Axially moving seals and cavities. Alternate cavities filled with oil shown below (Imo Pump)

differential pressure. For any given set of these conditions, the slip for all practical purposes is unaffected by speed. The delivered flow rate or net flow rate therefore is the theoretical flow rate less the slip: $Q = Q_t - S$. If the differential pressure is almost zero, the slip may be neglected and $Q = Q_t$.

The theoretical flow rate of any pump can readily be calculated if all essential dimensions are known. For any particular thread configuration, assuming geometric similarity, the size of each cavity mentioned earlier is proportional to its length and cross-sectional area. The thread pitch measured in terms of the same nominal diameter, which is used in calculating the cross-sectional area (see Figure 6), defines the length. Therefore, the volume of each cavity is proportional to the cube of this nominal diameter, and the pump's theoretical flow rate is also proportional to the cube of this nominal diameter and the speed of rotation N (rpm):

$$Q_t = kD^3N$$

or, bringing the pitch into evidence,

$$Q_t = k_1 \times \text{pitch} \times D^2N.$$

From Figure 6, where $\text{pitch} = KD$, it follows that $k = k_1 K$.

Thus, for a given geometry, it can be seen that a relatively small increase in pump size can provide a large increase in flow rate.

Slip can also be calculated,^{1,2} but usually it depends upon empirical values developed by extensive testing. These test data are the basis of the design parameters used by every

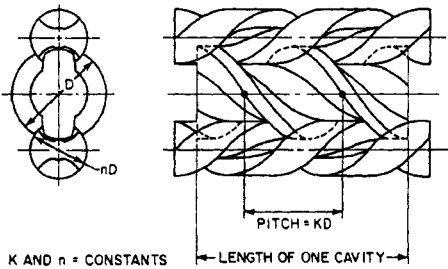


FIGURE 6 Screw-thread proportions, showing lead, diameters, and cavity length

pump manufacturer. Slip generally varies approximately as the square of the nominal diameter. The net flow rate therefore is

$$Q_t = kD^3N - S$$

Pressure Capability As mentioned earlier, screw pumps can be applied over a wide range of pressures, up to 5000 lb/in² (345 bar), provided the proper design is selected. Internal leakage must be restricted for high-pressure applications. Close-running clearances and high accuracy of the conjugate rotor threads are requirements. In addition, an increased number of moving seals between the inlet and outlet are employed, as in classic labyrinth-seal theory. The additional moving seals are obtained by a significant increase in the length of the pumping elements for a given size of rotor and pitch. Here the minimum pump length is sacrificed in order to gain pressure capability.

The internal leakage in the pumping elements resulting from the differential pressure between the outlet and inlet causes a pressure gradient across the moving cavities. The gradient is approximately linear (see Figure 7) when measured at any instant. Actually, the pressure in each moving cavity builds up gradually and uniformly from inlet to outlet pressure as the cavities move toward the outlet. In effect, the pressure capability of a screw pump is limited by the allowable pressure rise across any one set of moving seals. This pressure rise is sometimes referred to as *pressure per closure* or *pressure per lock* and generally is of the order of 125 to 150 lb/in² (8.6 to 10.3 bar) with normal running clearances, but it can go as high as 500 lb/in² (35 bar) when minimum clearances are employed.

The above analysis is used in the design of pumps for handling liquid products where there are sufficient lubricating properties in the liquid to allow the screws to contact the bores of the casing. However, when handling products containing significant amounts of dissolved and/or entrained gas, it is preferable to have no rubbing contact between the rotating elements and the casing bores. For these pumps, the rotors are relatively short and stiff to ensure there is no rotor to bore contact with the lower-viscosity, gas-laden multiphase fluids.

Design Concepts The pressure gradient in the pump elements of all the types of screw pumps produces various hydraulic reaction forces. The mechanical and hydraulic techniques employed for absorbing these reaction forces are among the fundamental differences in the types of screw pumps produced by various manufacturers. Another fundamental difference lies in the method of engaging, or meshing, the rotors and maintaining the running clearances between them. Two basic design approaches are used:

- The *timed rotors* approach relies upon an external means for phasing the mesh of the threads and for supporting the forces acting on the rotors. In this concept, theoretically, the threads do not come into contact with each other or with the housing bores in which they rotate (refer to Figure 4).
- The *untimed rotors* approach relies on the precision and accuracy of the screw forms for the proper mesh and transmission of rotation. They utilize the housing bores as journal bearings supporting the pumping reactions along the entire length of the rotors (refer to Figure 3).

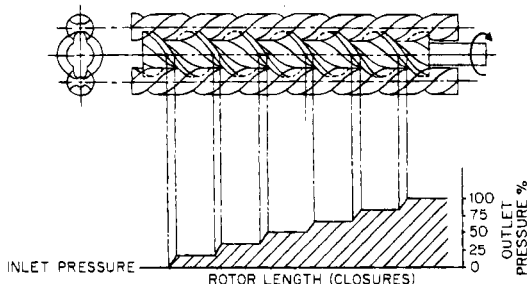


FIGURE 7 Pressure gradient along a screw set

Timed screw pumps require separate timing gears between the rotors and separate support bearings at each end to absorb the reaction forces and maintain the proper clearances. Untimed screw pumps do not require gears or external bearings and thus are considerably simpler in design.

CONSTRUCTION

Basic Types As indicated in the introduction, three major types of screw pumps exist:

- Single-rotor
- Multiple-rotor timed
- Multiple-rotor untimed

The second and third types are available in two basic arrangements, single-end and double-end. The double-end construction (see Figure 8) is probably the best-known version, as it has been by far the most widely used for many years because of its relative simplicity and compactness of design.

Double-End Screw Pumps The double-end arrangement is basically two opposed, single-end pumps or pump elements of the same size with a common driving rotor that has an opposed, double-helix design with one casing. As can be seen from Figure 8, the fluid enters a common inlet with a split flow going to the outboard ends of the two pumping elements and is discharged from the middle or center of the pump elements. The two pump elements are, in effect, pumps connected in parallel. The design can also be provided with a reversed flow for low-pressure applications. In either of these arrangements, all axial loads on the rotors are balanced, as the pressure gradients in each end are equal and opposite.

The double-end screw pump construction is usually limited to low- and medium-pressure applications, with 400 lb/in² (28 bar) being a good practical limit to be used for planning purposes. However, with special design features, applications up to 1400 lb/in² (97 bar) can be handled. Double-end pumps are generally employed where large flows are required or where very viscous liquids are handled.

Single-End Screw Pumps All three types of screw pumps are offered in the single-end construction. As pressure requirements in many applications have been raised, the single-end design has come into much wider use because it provides the only practical means

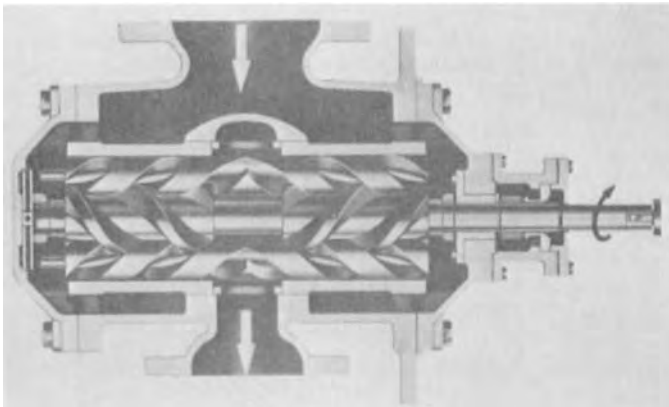


FIGURE 8 Double-end pump. Flow path provides axial balance (Imo Pump)

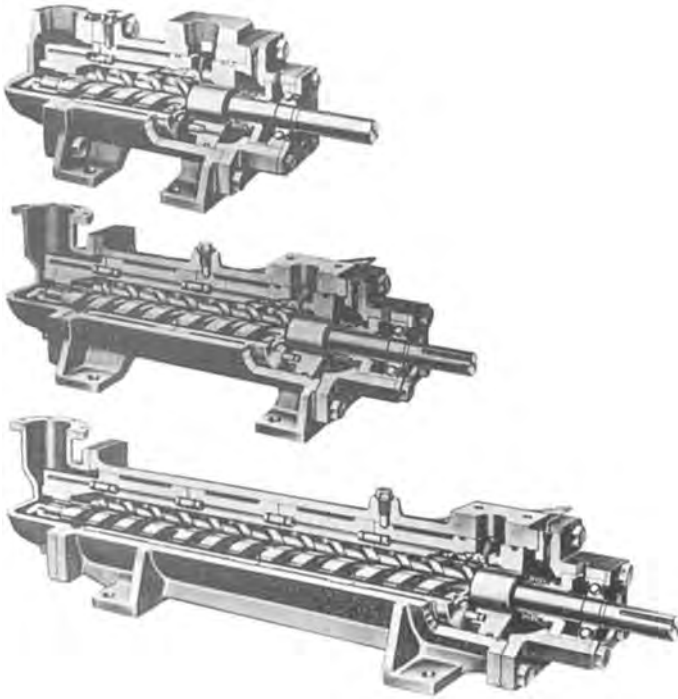


FIGURE 9 Increasing pump pressure capability by modular design (Imo Pump)

for obtaining the greatest number of moving seals necessary for high-pressure capabilities. The main penalties of the single-end pump are the requirement and complexity of balancing the axial loads.

The single-end construction is most often employed for handling low-viscosity fluids at medium-to-high pressures or hydraulic fluids at very high pressures. The single-end design for high pressures is developed by literally stacking a number of medium-pressure, single-end pumping elements in a series within one pump casing (see Figure 9). The single-end construction also offers the best design arrangement for quantity manufacturing.

Special mention must be made of the single-end, single-rotor design (see Figure 10). The pump elements of this design consist of only a stator and one rotor. The stator has a double-helix internal thread and is constructed of hard chrome stainless or tool steel material. One version of this design uses internally developed pressure in the pump to compress the elastomeric stator on the rotor, thus maintaining minimum running clearances.

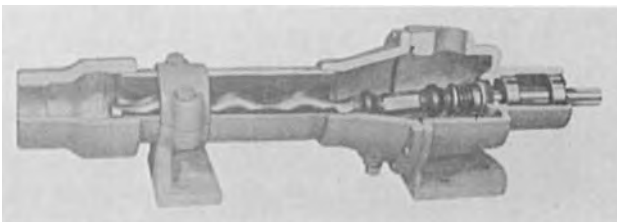


FIGURE 10 Single-rotor pump (Robbins-Meyers)

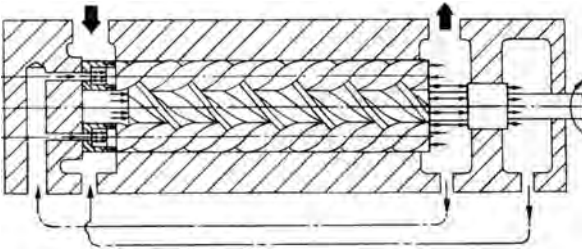


FIGURE 11 Axial balancing of power and idler rotors

In most single-end designs, special axial balancing arrangements must be used for each of the rotors, and, in this respect, the design is more complicated than the double-end construction. For smaller pumps, strictly mechanical thrust bearings can be used for differential pressures up to 150 lb/in² (10.3 bar), while hydraulic balance arrangements are used for higher pressures. For large pumps, hydraulic balance becomes essential at pressures above 50 lb/in² (3.5 bar).

Hydraulic balance is provided through the balance piston (see Figure 11) mounted on the rotors between the outlet and seal or bearing chambers, which are at inlet pressure. This piston is exposed to discharge pressure in the outlet chamber and is equal in area to the exposed area of the driven rotor threads; thus, the hydraulic forces on the rotor are canceled out.

Timed Design Timed screw pumps having timing gears and rotor support bearings are furnished in two general arrangements: internal and external. The internal version has both the gears and the bearings located in the pumping chamber and the design is relatively simple and compact (see Figure 12). This version is generally restricted to the handling of clean lubricating fluids, which serve as the only lubrication for the timing gears and bearings.

The external timing arrangement is the most popular and is extensively used. It has both the timing gears and the rotor support bearings located outside the pumping chamber (see Figure 13). This type can handle a complete range of fluids, both lubricating and non-lubricating, and, with proper materials, has good abrasion resistance. The timing gears and bearings are oil-bath-lubricated from an external source. This arrangement requires the use of four stuffing boxes or mechanical seals, as opposed to the internal type, which employs only one shaft seal.

The main advantage of the timed screw pump is that the timing gears transmit power to the rotors with no contact between the screw threads, thus promoting long pump life. The gears and rotors are timed at the factory to maintain the proper clearance between the screws. With certain designs and loading characteristics, the bearings at each end of

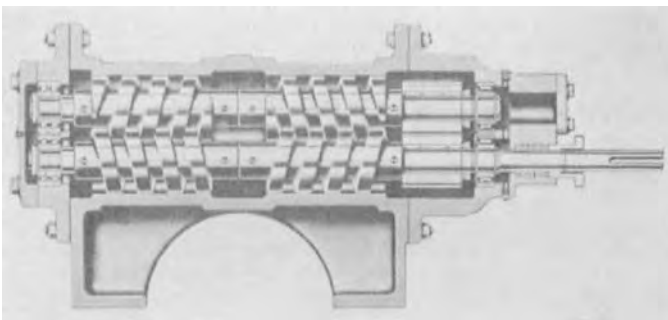


FIGURE 12 Double-end internal gear design (Flowserve Corporation)

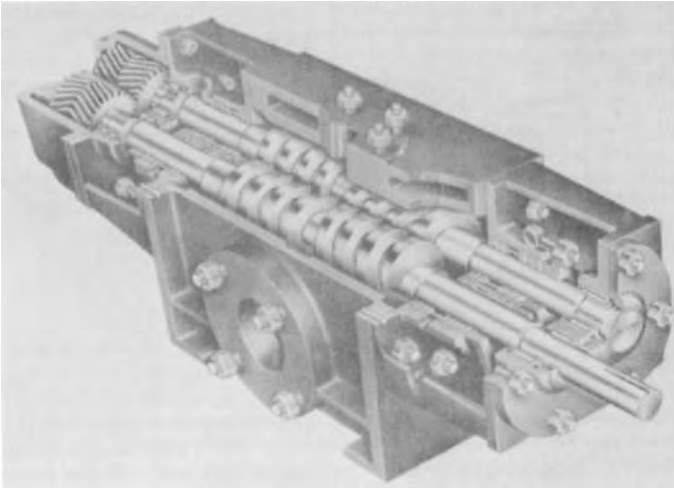


FIGURE 13 Double-end external gear design (Warren)

the rotating elements can support the rotors so that they do not come in contact with the housing; hence, no liner is required. One set of these bearings also positions and supports the timing gears. Under more heavily loaded conditions, the body bores act as sleeve bearings and provide additional support for the rotors.

The timing gears can be either spur or helical, herringbone, hardened-steel gears with tooth profiles designed for efficient, quiet, positive drive of the rotors. Antifriction radial bearings are usually of the heavy-duty roller type, while the thrust bearings, which position the rotors axially, are either double-row, ball-thrust or spherical-roller types.

The housing can be supplied in a variety of materials, including cast iron, ductile iron, cast steel, stainless steel, and bronze. In addition, the rotor bores of the housing can be lined with industrial hard chrome for abrasion resistance.

Since the rotors are not generally in metallic contact with the housing or with one another, they can also be supplied in a variety of materials, including cast iron, heat-treated alloy steel, stainless steel, Monel, and nitralloy. The outside of the rotors can also be furnished with a variety of hard coating materials such as nickel-based alloys, tungsten carbide, chrome oxide, or ceramic.

Untimed Design The untimed type of screw pump has rotors that have generated mating-thread forms that enable any necessary driving force to be transmitted smoothly and continuously between the rotors without the use of timing gears. The rotors can be compared directly with precision-made helical gears with a high helix angle. This design usually employs three rotor screws with a center or driven rotor that meshes with two close-fitting sealing or idler rotors symmetrically positioned about the central axis (see Figure 14). A close-fitting housing provides the only transverse bearing support for both driven and idler rotors.

The use of the rotor housing as the only means for supporting idler rotors is a unique feature of the untimed screw pump. No outboard support bearings are required on these rotors. The idler rotors in their related housing bores are, in effect, partial journal bearings that generate a hydrodynamic film. The key parameters of rotor size, clearance, surface finish, speed, fluid viscosity, and bearing pressure are related as in a journal bearing. Because, in this design, the screws transmit the driving torque to the idlers and because the casing bores support the rotors, the pumped fluid must have some lubricating properties. This prevents this pump design from being used in applications with any solids, abrasives or gas in the pumpage.

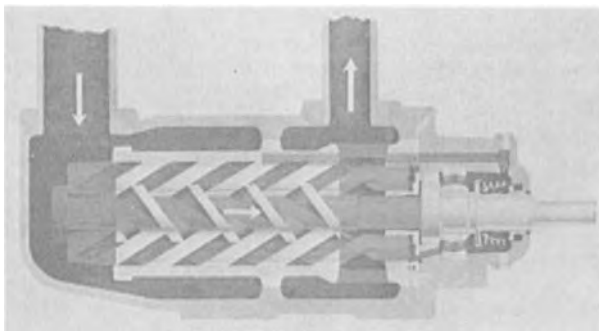


FIGURE 14 Single-end design (Roper)

Since the idler rotors are supported by the bores along their entire length, no significant bending loads are applied to them. The central driven rotor is also not subjected to any significant bending loads because of the symmetrical positioning of the idler rotors and the use of two threads on all the rotors (see Figure 15). This is quite different from the two-rotor design common to the timed type where the hydraulic forces generated in the pump create bending loads on the intermeshing pairs of screws.

In contrast to timed pumps, the untimed design, with its absence of timing gears and bearings, appears very simple, but its success depends entirely upon the accuracy and finish of the rotor threads and rotor housing bores. Special techniques and machine tools have been developed to manufacture these parts. The combination of design simplicity and manufacturing techniques has enabled this design to be used for very long rotor lengths, with a multiplicity of sealing closures, for applications up to 5,000 lb/in² (350 bar).

In special applications handling highly aerated oils, a four-rotor design is sometimes employed in the untimed type. Three idler rotors are equally spaced radially at 120° around the driven rotor. This design is not truly a positive displacement pump, and it falls outside the scope of this section.

The rotors in untimed pumps are generally made of gray or ductile iron or carbon steel. The thread surfaces are often hardened for high pressure and abrasive resistance. Flame hardening, induction hardening, and nitriding are currently used. Through-hardened tool steel or stainless steel can be used in some critical applications.

Rotor housings, or liners, are made of gray pearlitic iron, bronze, or aluminum alloy. In many instances, the bores as well as the rotors can be treated by the application of dry lubricant or toughening coatings. The pump casings are made of gray iron or ductile iron or cast steel where shock or other safety requirements demand it.

In many untimed designs, an antifriction bearing is employed on the shaft end of the driven rotor (refer to Figure 9) to provide precise shaft positioning for mechanical seal and

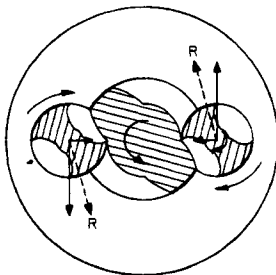


FIGURE 15 Force diagram on rotor set

coupling alignment. This bearing can be either an external grease-sealed bearing or an internal type with the pumped fluid providing the lubrication. The bearing also supports overhung loads with belt or gear drives.

Seals As with any rotary pump, the sealing arrangement for the shafts is important and is often critical. Every type of rotary seal has been used in screw pumps at one time or another. Except for canned or sealless arrangements, all types of pumps require at least one rotary seal on the drive shaft. The timed screw pumps with external timing and bearings require additional seals at each rotor end to separate the pumped fluid from the lubricating oil necessary for the gears and bearings.

For drive shafts, rotary mechanical seals as well as stuffing boxes or packings are used, depending on the manufacturer and/or customer preference. Double back-to-back arrangements or tandem mounted seals with a flushing liquid are sometimes used for very viscous or corrosive substances.

Many different sealing face materials such as carbon, bronze, cast iron, Ni-resist, carbides, or ceramics are used, along with different secondary elastomer seals of various materials chosen to suit the application. Balanced and nonbalanced seals can also be used to handle significant pressures. In some pump designs, the seals are subjected to only the suction pressure, while in other designs the seals must seal against the full discharge pressure. In spite of these advantages, the stuffing box is still preferred by some users. The stuffing box requires regular maintenance and tightening, which many users find objectionable, but a mechanical seal failure usually results in a major shutdown.

PERFORMANCE

Performance considerations of screw pumps are closely related to applications, and so any discussion must cover both. In the application of screw pumps, certain basic factors must be considered to ensure a successful installation. These are fundamentally the same regardless of the liquids to be handled or the pumping conditions.

In most cases, the pump selection for a specific application is not difficult if all the operating parameters are known. It is often quite difficult, however, to obtain this information, particularly inlet conditions and fluid viscosity. It is a common feeling that, inasmuch as the screw pump is a positive displacement device, these items are unimportant.

In any screw pump application, regardless of design, suction lift, viscosity, and speed are inter related. Generally, low operating speeds are required for high-viscosity fluids while higher speeds are possible with lower-viscosity fluids.

Inlet Conditions The key to obtaining good performance from a screw pump, as with all other positive displacement pumps, lies in a complete understanding and control of inlet conditions and the closely related parameters of speed and viscosity. To ensure quiet, efficient operation, it is necessary to completely fill the moving cavities between the rotor threads with liquid as they open to the inlet, and this becomes more difficult as the viscosity, speed, or suction lift increases. It can be said that, if the liquid can be properly introduced into the rotor elements, the pump will perform satisfactorily.

It must be remembered that a pump does not pull or lift liquid into itself. Some external force must be present to push the liquid into the rotor threads initially. Normally, atmospheric pressure is the only force present, but in some applications a positive inlet pressure is available.

As with centrifugal pumps, the concept of NPSH required by the pump is used to ensure that there is sufficient pressure available at the inlet to fill the rotor cavities and suppress cavitation at the inlet. With rotary pumps this is sometimes referred to as *Net Inlet Pressure Required*.

Since the purpose of the NPSH available is to fill the moving inlet cavities with liquid, it is apparent that the faster the pump is operated *the greater is the amount of NPSH* that is required.

Fluids and Vapor Pressure In many cases, screw pumps handle a mixture of liquids and gases, and therefore the general term *fluid* is more descriptive. Most of these fluids, especially petroleum products because of their complex nature, contain certain amounts of entrained and dissolved air or some other gas, which is released as a vapor when the fluid is subject to reduced pressures. If the pressure drop required to overcome entrance losses is sufficient to reduce the static pressure significantly, vapors are released in the rotor cavities and cavitation results.

In all screw pump applications, the absolute static pressure must never be allowed to drop below the vapor pressure of the fluid. This will prevent vaporization or cavitation. Cavitation, as mentioned previously, results when fluid vaporizes in the pump inlet because of incomplete filling of the pump elements and a reduction of pressure. Under these conditions, vapor bubbles, or voids, pass through the pump and collapse as each moving cavity moves into a domain of higher pressure. The result is noisy vibrations, the severity depending on the extent of vaporization or the incomplete filling and the magnitude of the discharge pressure. Since the cavitation bubbles occupy space normally taken by liquid, the output flow rate is reduced by this amount. It is therefore important to be fully aware of the characteristics of entrained and dissolved air as well as of the vapor pressure of the fluid to be handled. This is particularly true when a suction lift exists.

Net Positive Suction Head The suction conditions of screw pumps are normally defined by the *Net Positive Suction Head* (NPSH) available at the pump inlet. In some cases with systems open to the atmosphere, the inlet conditions can also be defined as suction lift. Suction lift occurs when the total available pressure at the pump inlet is below the atmospheric pressure and is normally the result of a change in elevation and pipe friction. Since atmospheric pressure at sea level corresponds to 14.7 lb/in² (1 bar) absolute, or 30 in Hg (762 mm Hg), in a system with the suction open to the atmosphere, this is the maximum amount of pressure available for moving the fluid, and suction lift cannot exceed these figures. In practice, the available pressure is lower because some of it is used up in overcoming friction in the inlet lines, valves, and fittings. It is considered the best practice to keep the suction lift as low as possible to ensure that an adequate NPSH is available to fill the suction cavities of the pumping screw. In all cases, the system NPSHA (available) should be greater than the pump NPSHR (required).

The NPSHA for a screw pump is calculated in the same way as for a centrifugal pump and the same basic application guidelines apply. The pump manufacturers test their pumps and measure the NPSHR (required) under various operating conditions. The end user must understand the suction pressure required and ensure that there is sufficient NPSHA (available) to suppress the cavitation. Generally the NPSHA should be at least 3 ft (1 m) greater than the NPSHR to provide a reliable application.

Many screw pumps operate with suction lifts of approximately 5 to 15 in Hg (127 to 381 mm Hg). Lifts corresponding to 24 to 25 in Hg (610 to 635 mm Hg) are not uncommon, and installations can operate satisfactorily when the absolute suction pressure is much lower. In the latter cases, however, the pumps usually take the fluid from tanks under a vacuum, and no entrained or dissolved air or gases are present. Great care must be taken when selecting pumps for these applications since the inlet losses can easily exceed the net suction head available for moving the fluid into the pumping elements.

The defining of suction requirements by the user and the stating of pump suction capabilities by the manufacturer have always been complex problems. In many cases, the NPSHA is difficult to predict due to changes in the fluid characteristics and the operating conditions. In addition, the NPSH required by the pump is a function of many variables, such as pump design, fluid characteristics, and operating conditions. If the operating conditions can be accurately defined, the pump manufacturers can predict the NPSHR and in many cases can provide pump modifications that can minimize the NPSHR.

To enable the pump manufacturer to offer the most economical selection and also assure a quiet installation, accurate suction conditions should be clearly stated. Specifying a lower NPSHA than actually exists may result in selection of a pump that operates at a lower speed than necessary. This means not only a larger and more expensive pump, but also a costlier driver. If the NPSHA is lower than stated, the outcome could be a noisy pump installation.

Many known instances of successful installations exist where screw pumps were properly selected for low NPSHA conditions. Unfortunately, many other installations with equally low NPSHAs exist, which are not so satisfactory. This is because proper consideration was not given at the time the pump was specified and selected to the actual suction conditions at the pump inlet. Frequently, suction conditions are given as "flooded" simply because the source feeding the pump is above the inlet. In many cases, no consideration is given to outlet losses from the tank or to pipe friction in the inlet lines, and these can be exceptionally high in the case of viscous liquids.

When it is desired to pump extremely viscous products, care should be taken to use the largest feasible size of suction piping to eliminate all unnecessary fittings and valves, and to place the pump as close as possible to the source of the supply. In addition, it may be necessary to supply the liquid to the pump under some pressure, which can be supplied by elevation, air pressure, or mechanical means. These actions will provide the maximum NPSH possible to the pump inlet.

Entrained and Dissolved Air Although screw pumps can handle significant quantities of gas as discussed in the later multiphase section, entrained or dissolved gas in a pump system designed only for liquids can create a problem and must be considered in the application. This is particularly true of installations where recirculation occurs and the fluid is exposed to air through either mechanical agitation, leaks, or improperly located drain lines.

Most liquids will dissolve air or other gases retaining them in the solution, the amount being dependent upon the liquid itself and the pressure to which it is subjected. It is known, for instance, that lubricating oils at atmospheric temperatures and pressures will dissolve up to 10% air by volume and that gasoline will dissolve up to 20%. When pressures below the atmosphere exist at the pump inlet, dissolved air will come out of the solution. Both this and the entrained air will expand in proportion to the existing partial pressure of the air (= absolute pressure minus the vapor pressure of the liquid). This expanded air will accordingly take up a proportionate part of the available volume of the moving cavities, with a consequent reduction in delivered flow rate.

One of the apparent effects of handling liquids containing entrained or dissolved gas is noisy pump operation. When such a condition occurs, it is usually dismissed as cavitation. This should not be the case, particularly with screw pumps. With properly designed systems and pumps, quiet, vibration-free operation can be produced and should be expected. Noisy operation is inefficient; steps should be taken to make corrections until the objectionable conditions are overcome. Correct system inlet designs and optimized pump designs with a proper speed selection can go a long way toward overcoming the problem.

In some applications, the amount of gas can be significant and can make up the majority of the fluid volume. See the later section on handling special multiphase applications.

Viscosity It is not often that a screw pump is called upon to handle liquids at a constant viscosity. Normally, because of temperature variations, a wide range of viscosities will be encountered. For example, a pump may be required to handle a viscosity range from 150 to 20,000 SSU, the higher viscosity usually resulting from cold-starting conditions. This is a perfectly satisfactory range for a screw pump, but a better and a more economical selection may be possible if additional information can be obtained. This information includes such things as the amount of time the pump is required to operate at the higher viscosity, whether the motor can be overloaded temporarily, whether a multi-speed motor can be used, and if the discharge pressure will be reduced during the period of high viscosity.

Quite often, only the type of liquid is specified, not its viscosity, and assumptions must be made for the operating range. For instance, Bunker C or No. 6 fuel oil is known to have a wide range of viscosity values and usually must be handled over a considerable temperature range. The normal procedure in a case of this type is to assume an operating viscosity range of 20 to 700 SSF. The maximum viscosity, however, might easily exceed the higher value if extra-heavy oil is used or if exceptionally low temperatures are encountered. If either should occur, the result may be improper filling of the pumping elements, noisy operation, vibration, and overloading of the motor.

Although it is the maximum viscosity and the expected NPSHA that are used to determine the size of the pump and to set the speed, it is the minimum viscosity that affects the

capacity. Screw pumps must always be selected to give the specified capacity when handling the expected minimum viscosity since this is the point at which the maximum slip, and hence minimum flow rate, occurs (see Figure 16). It should also be noted that the minimum viscosity often determines the selection of the pump model because most manufacturers have special lower-pressure ratings for handling liquids having a viscosity of less than 100 SSU.

Non-Newtonian Liquids The viscosity of most liquids is unaffected by any agitation or shear to which they may be subjected as long as the temperature remains constant. These liquids are accordingly known as *true* or *Newtonian* liquids, for which the viscosity is constant. Another class of liquids, however, such as cellulose compounds, glues, greases, paints, starches, slurries, and candy compounds, displays changes in viscosity as agitation is varied at constant temperature. The viscosity of these substances depends upon the shear rate at which it is measured, and these fluids are termed *non-Newtonian*.

If a substance is known to be non-Newtonian, the expected viscosity under the actual pumping conditions should be determined, because it can vary quite widely from the viscosity under static conditions. Since a non-Newtonian substance can have an unlimited number of viscosity values (as the shear rate is varied), the term *apparent viscosity* is used to describe its viscous properties. The apparent viscosity is expressed in absolute units and is a measure of the resistance to the flow at a given shear rate. It has meaning only if the shear rate used in the measurement is also given.

The grease-manufacturing industry is very familiar with the non-Newtonian properties of its products, as evidenced by the numerous curves that have been published of the apparent viscosity plotted against the rate of shear. The occasion is rare, however, when one can obtain accurate viscosity information when it is necessary to select a pump for handling these products.

It is practically impossible in most instances to give the viscosity of grease in the terms most familiar to the pump manufacturer, such as Saybolt Seconds Universal or Saybolt Seconds Furol, but only a rough approximation would be of great help. For applications of this type, data taken from similar installations are most helpful. Such information should consist of the type, size, flow rate, and speed of the installed pumps; the suction pressure; the temperature at the pump inlet flange; the total working suction head; and, above all, the pressure drop in a specified length of piping. From the latter, a satisfactory approximation of the effective viscosity under the operating conditions can be obtained.

If accurate shear rate-viscosity data are available, they can be used to more accurately predict pump performance. The shear rates in various areas of the pump can be calculated to determine the viscosity changes of the liquid as it passes through the pump. In this way, the effect on suction loss, slip, and friction loss can be analyzed to help predict the NPSH, flow rate, and power.

Speed It was previously stated that viscosity and speed are closely tied together and that it is impossible to consider one without the other. Although rotative speed is the

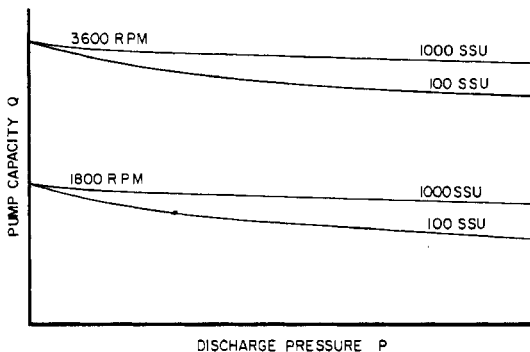


FIGURE 16 Flow rate (capacity)-vs.-pressure rise

ultimate outcome, the basic speed that the manufacturer must consider is the internal axial velocity of the liquid going through the rotors. This is a function of pump type, design, and size.

Rotative speed should be reduced when handling liquids of high viscosity. The reasons for this are not only the difficulty of filling the pumping elements, but also the mechanical losses that result from the shearing action of the rotors on the substance handled. The reduction of these losses is frequently of more importance than relatively high speeds, even though the latter might be possible because of positive inlet conditions.

Capacity The delivered flow rate (capacity) of any screw pump, as stated earlier, is the theoretical capacity less the internal leakage, or the slip, when handling vapor-free liquids. For a particular speed, $Q = Q_t - S$, where the standard unit of Q and S is the U.S. gallon per minute (cubic meter per minute).

The delivered capacity of any specific rotary pump is reduced by

- Decreasing speed
- Decreased viscosity
- Increased differential pressure

Because of the internal clearances between rotors and their housing, lower viscosities and higher pressures increase the slip, which results in a reduced flow rate for a given speed. The impact of these characteristics can vary widely for the various types of pumps. The slip, however, is not measurably affected by changes in speed and thus becomes a smaller percentage of the total flow at higher speeds. This is a significant factor in the handling of low-viscosity fluids at higher pressures, particularly in the case of untimed screw pumps that favor high speeds for the best results and best volumetric efficiency. This will not generally be the case with pumps having support-bearing speed limits.

Pump volumetric efficiency E_v is calculated as

$$E_v = \frac{Q}{Q_t} = \frac{Q_t - S}{Q_t}$$

with Q_t varying directly with speed. As stated previously, the theoretical capacity of a screw pump varies directly as the cube of the nominal diameter. Slip, however, varies approximately with the square of the nominal diameter. Therefore, for a constant speed and geometry, doubling the rotor size will result in an eightfold increase in theoretical flow rate and only a fourfold increase in slip. It follows therefore that the volumetric efficiency improves rapidly with increases in the rotor size, assuming the internal clearances can be maintained.

On the other hand, viscosity changes affect the slip inversely in a complex manner that depends on pump size, speed, and the level of the viscosity itself. Testing or advanced analysis is required to establish this relationship.¹ However, for a given viscosity in the range of 100 to 10,000 SSU an acceptable approximation is that slip varies approximately with the differential pressure. Thus, in this case, a change from 100 to 200 lb/in² (7 to 14 bar) would double the slip.

Figure 17 shows typical results for flow rate and volumetric efficiencies as functions of pump size.

Pressure Screw pumps do not in themselves create pressure; they simply transfer a quantity of fluid from the inlet to the outlet side. The pressure developed on the outlet side is solely the result of resistance to the flow in the discharge line. The slip characteristic of a particular pump type and model is one of the key factors that determine the acceptable operating range, and it is generally well defined by the pump manufacturer.

Power The brake horsepower (bhp), or, in SI units, the brake kilowatts, required to drive a screw pump is the sum of the theoretical liquid horsepower (kilowatts) and the internal power losses. The theoretical liquid power *twhp* (*tkW*) is the actual work done in moving the fluid from its inlet pressure condition to the outlet at the discharge pressure.

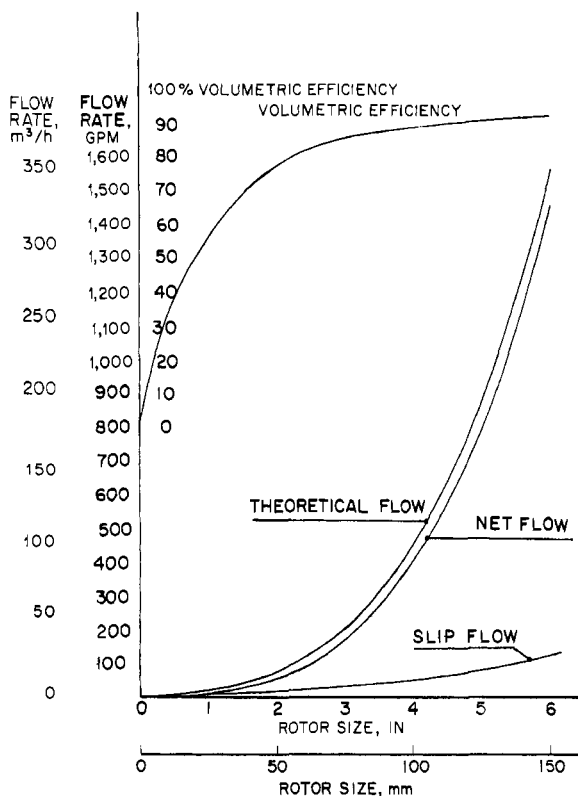


FIGURE 17 Typical variations of flow rate and volumetric efficiency as functions of pump size

Note that this work is done on all the fluid of the theoretical capacity, not just the delivered capacity, as the slip does not exist until a pressure differential DP occurs. Screw pump power ratings are expressed in terms of horsepower (550 ft-lbf/sec) in USCS units and in terms of kilowatts in SI units. The theoretical liquid horsepower (kilowatts) can be calculated as follows:

$$twhp = \frac{Q_t \Delta P}{1714} \left(tkW = \frac{Q_t \Delta P}{36} \right)$$

It should be noted that the theoretical liquid horsepower (kilowatts) is independent of the viscosity and is a function only of the physical dimensions of the pumping elements, the rotative speed, and the differential pressure.

The internal power losses are of two types: mechanical and viscous. The mechanical losses include all the power necessary to overcome the frictional drag of all the moving parts in the pump, including rotors, bearings, gears, and mechanical seals. The viscous losses include all the power lost from the fluid drag effects against all the parts in the pump as well as from the shearing action of the fluid itself. It is probable that the mechanical loss is dominant when operating at low viscosities and high speeds, and the viscous loss is the larger of these two losses at high-viscosity and slow-speed conditions.

In general, the losses for a given type and size of pump vary with the viscosity and the rotative speed and may or may not be affected by pressure, depending upon the type and

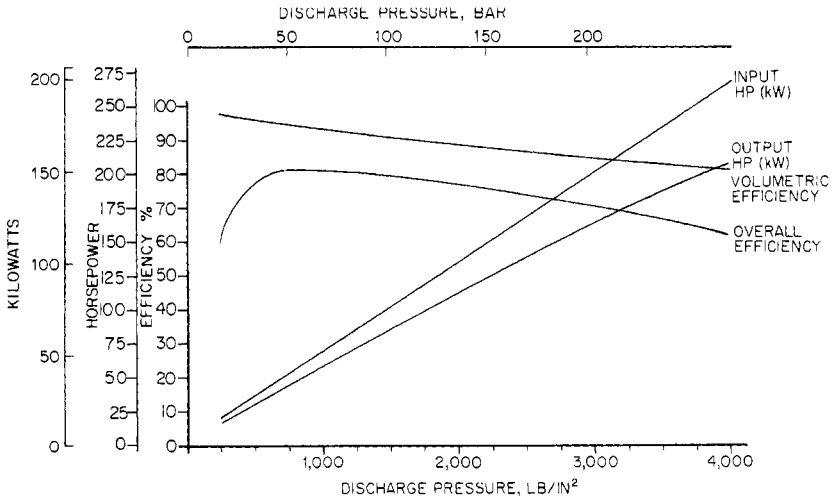


FIGURE 18 Typical overall efficiency curves

model of pump under consideration. These losses, however, must always be based upon the maximum viscosity to be handled since they will be highest at this point.

The actual pump power output, whp (whW), or the delivered liquid horsepower (kilowatts), is the power imparted to the liquid by the pump at the outlet. It is computed similarly to theoretical liquid horsepower (kilowatts) using Q in place of Q_t . Hence, the value will always be less. The pump efficiency E_p is the ratio of the pump power output to the brake horsepower (see Figure 18).

In many cases the efficiency of the pump is not that significant, since there may be only a few pumps capable of handling high viscosity and difficult fluids. The main requirement is to provide a reliable equipment package that is capable of meeting the specified requirements.

SPECIAL MULTIPHASE APPLICATIONS

Screw pumps have been used with gas-entrained applications for many years, but recent process changes in oil field technologies have created requirements for pumping multiphase fluids, containing more than just nominal amounts of gases. In many oil well applications, the liquid oil flow eventually degenerates into all sorts of difficult multiphase mixtures of oil, gas, water, and sand. In the past, it was common for the gas to be separated and flared off at the well head with only the liquid product to be retained for further processing. If the gas is to be processed as well, separators, compressors, and dual pipelines are required to handle the gas phase. Therefore, a pump, which can handle these difficult liquids with high gas content, can save significant equipment costs as well as operating costs. Under various conditions, the well output can vary from 100% liquid to 100% gas and all possible combinations. The applications also require the pumping equipment to be able to switch rapidly between the extremes or to handle slugs of liquid or gas, while maintaining the full discharge pressure. Since these applications involve solid contaminants, internal bearing and non timed pump designs are not suitable. The single rotor design with elastomer stator has proven suitable for lower pressure applications while the timed two-screw type of pump has been proven capable of pumping these multiphase products at higher pressures.²

Oil well applications can include traditional on-shore sites as well as off-shore platform installations. Subsea installations are also being used to reduce the high costs of equipment and operation of traditional oil platforms. In these applications, the pumping equipment is mounted on the sea bed with piping running to floating production, storage and off-loading facilities or to onshore gathering facilities. Although the actual pumping conditions are similar to surface installations, the installation and operating environments are far more challenging.

When pumping multiphase products with high *gas void fractions* (GVF), the pump must be designed with a small pitch to provide a sufficient number of locks. The key to pumping multiphase products is to ensure that some liquid is always available to seal the screw clearances and reduce the slip. Even a small amount of recirculated liquid is sufficient to provide this seal and enable the screw pump to operate with GVF-values approaching 100%. Depending on a number of factors, the volume of liquid required to seal and cool the screws can be two to five percent of the total inlet volume flow rate. In order to ensure that sufficient liquid is available at conditions of high GVF, a separate liquid flush can be provided or a trapping type of pump casing can be used. This type of casing has the suction and discharge openings located on the upper side of the casing to trap the liquid inside the screw chambers to seal the screw clearances. In some cases, some of the liquid portion is separated from the multiphase mixture being pumped. This separated liquid can then be recirculated back to the screws and mechanical seals to provide sealing and cooling liquid at times when the product is almost all gas. Figure 19 shows a special screw pump designed for multiphase applications with a casing designed to trap the liquid inside the casing bores to seal the screw clearances.³

When pumping liquids, the slip through the internal clearances increases with the differential pressure and decreases with increasing viscosity. However, in multiphase applications, as the GVF increases, the slip decreases until the inlet volume flow rate is equal to the pump displacement. This results in an almost constant inlet volume flow rate, regardless of the differential pressure. This performance can be explained by examining the pressure drop of the multiphase product across the finite clearances. This theoretical analysis confirms that a small amount of liquid in the clearances will effectively seal these clearances and reduce the slip to near zero. Figure 20 shows the typical performance characteristics for

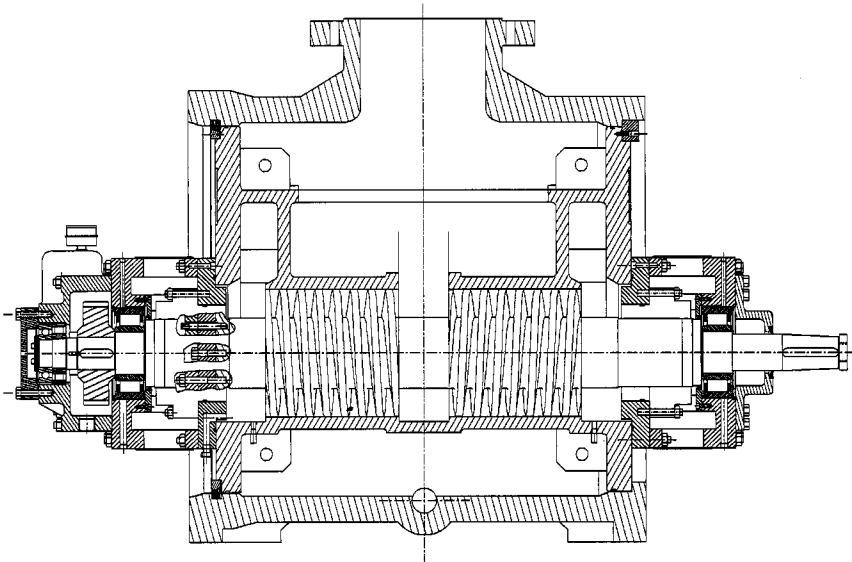


FIGURE 19 Special screw pump for multiphase applications (Flowserve Corporation)

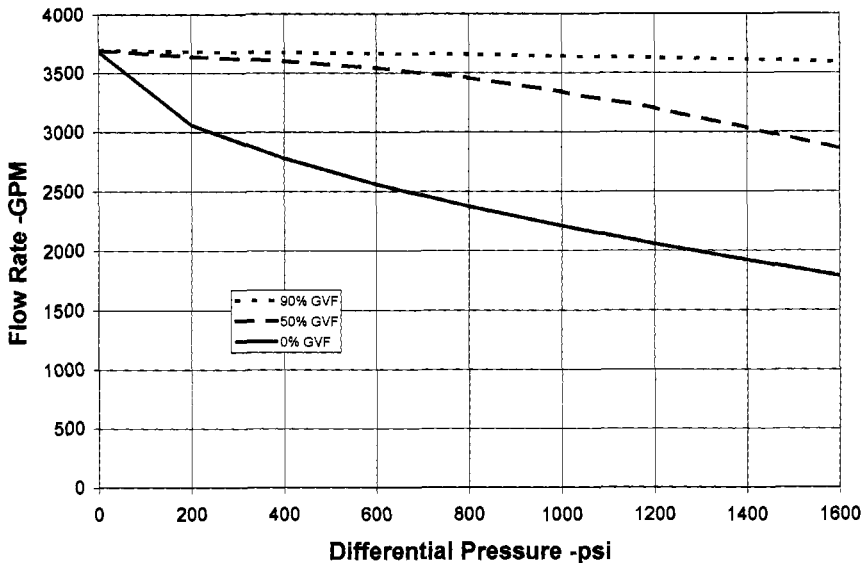


FIGURE 20 Typical performance of a screw pump on multiphase fluid ($\text{m}^3/\text{h} = \text{gpm}/4.403$; $\text{MPa} = \text{psi}/145.04$)

a screw pump in a multiphase application when pumping mixtures of air and water at various GVF values.^{1,4}

It should be emphasized that screw pumps must be sized for the inlet volume conditions. Since the gas portion of the multiphase product is compressible, the inlet pressure and temperature conditions must be known in order to calculate the gas volume. The pump will ingest a fixed volume of product and the amount of liquid being pumped will depend on how much of this volume is being displaced by gas at the inlet to the screws.

When pumping higher GVF multiphase products some significant design changes must be made to the screw pumps to improve reliability and ensure proper pumping action. The normal practice used with viscous liquid pumping of using the casing bores to help support the rotor loads is not used with multiphase liquids due to the lower liquid viscosity, the high percentage of gas and the entrained sand and other contaminants present in oil field applications. As a result, pumps for multiphase applications are generally designed with stiffer rotors so that the rotors do not contact the casing bores. This design change is accomplished by using larger diameter shafts with shorter bearing spans. The integral shaft and rotor design used by some manufacturers significantly increases the rotor stiffness and minimizes the rotor deflection under load.

Another design feature used for multiphase applications involves limiting the pitch of the screws to ensure there are sufficient screw locks available to generate the pressure rise and produce a more gradual pressures rise across the screw length. Since the volume of the gas is compressed as the multiphase fluid moves through the screw passages, any large volume changes will result in pressure pulsations and resulting damaging vibration. A general guideline for higher pressure multiphase applications requires a minimum of three screw locks in order to minimize the pressure pulsations. In multiphase applications the maximum pressure rise occurs in the last lock when it opens to the full discharge pressure of the pump. Special porting of the screw profiles can also be used to help minimize the pressure pulsation and resulting vibration.

Again the mechanical seals are the most critical items in the pump required to ensure pump reliability. The pump should be designed to provide the best environment for the mechanical seals to operate properly. This includes limiting rotor deflection at the seal area and providing suitable flush or barrier fluid. Since these pumps can be designed to operate with the seals in either the suction or the discharge it is normally preferable to have

the seals work against the lower suction pressure as opposed to the higher discharge pressure. When sealing against variable high suction or discharge pressures, the barrier fluid pressure must be controlled to ensure that it is always greater than the product pressure.

In many multiphase applications, the drive motors are equipped with variable speed controls to allow the pump to operate over a wide speed range to match specific start-up and operating well conditions. In these cases the speed is controlled to maintain a constant pressure at the pump suction. If the well flow rate increases, the suction pressure increases, causing the speed of the pump to also increased to maintain a constant suction pressure and thereby match the flow rate from the well. Multiphase pumps are often used as booster pumps to draw down the well head pressure and thereby increase the well flow rate and boost production. These pumps are often sized to handle high flow rates and can then be installed on a cluster of wells to boost the pressure into the pipeline.

Subsea A special area of application for multiphase pumps is subsea. With special modifications, the rotary screw multiphase pumps can be coupled to submersible motors and mounted on the sea bed, instead of on surface platforms. The idea of pumping multiphase products directly from a subsea well head to onshore or floating production and storage facilities by means of submersible multiphase pumps has significant potential savings in separating equipment and platforms.⁵ These special subsea pumps often require much higher pressure capabilities than surface mounted pumps, to handle both the high ambient subsea pressures and also the high discharge pressures required. Again the mechanical seals are a critical design area with special considerations required to ensure a reliable design. The sealing environment can be significantly improved by maintaining a constant differential pressure across the seal faces by means of a pressure compensator. This pressure compensator senses the product pressure on the seals and then provides a slight pressure boost to the barrier fluid to maintain this constant differential pressure. The pump and system must be designed in conjunction with the submersible motor, pressure compensator and lube oil cooler. Obviously, the motor and valve control functions as well as the lube oil top-up capabilities must be integrated into the complete pump module to provide a reliable unit to operate in the subsea environment. Subsea multiphase pumps are being investigated for deeper water applications and higher boost pressures. Initial subsea pumping installations involved water depths up to 1500 m, with boost pressures of 1250 psi (86 bar), applications up to 3000 m water depth with pressure boosts to 3000 psi (207 bar) are now being pursued. Figure 21 shows a special two-screw multiphase

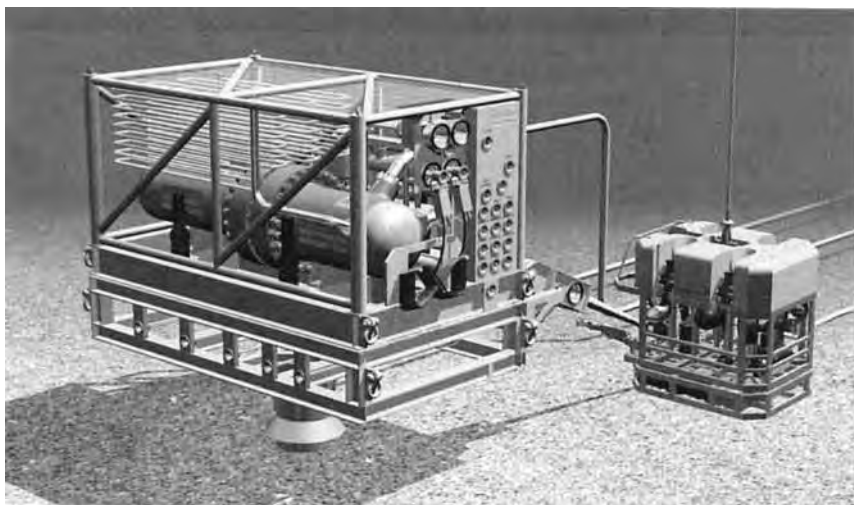


FIGURE 21 Multiphase screw pump module for subsea deployment (Flowserve Corporation)

pump module, configured for subsea applications with high-pressure, shell-type casing design, integrated submersible motor, pressure-compensator and oil cooler.

Wet Gas Compression Another application for multiphase screw pumps is for “wet gas compression.” This application involves pumping multiphase products that are essentially gas but can also have occasional high liquid content. These applications generally involve fluids with gas content in the 98% GVF range. There are two areas of concern with this type of application. As previously indicated, there must always be some liquid available in the pump to seal the clearances between the screws and the casing bores. Also with this high gas content, special considerations must be made for the thermodynamic heating characteristics of the compressed gas. Since the gas provides much lower heat absorption than the normal liquid, the product temperature can increase significantly as the pressure is increased. The pump must also be able to handle the full range of GVF values from slugs of liquid to 100% gas. For these applications special screw designs, liquid injection and recirculating liquid separators can be used to ensure there is sufficient liquid available to seal and cool the rotors.

INSTALLATION AND OPERATION

Rotary pump performance and life can be improved by following the recommendations on installation and operation outlined in this section.

Pipe Size Resistance to the flow usually consists of differences in elevation, fixed resistances or restrictions such as orifices, and pipe friction. Nothing can be done about the first, since this is the basic reason for using a pump. Something can be done, however, about restrictions and pipe friction. Significant amounts of money can be wasted because of piping that is too small for the job. Certainly, not all pipe friction can be eliminated as long as fluids must be handled in this manner, but every effort should be made to use the largest pipe that is economically feasible. Numerous tables are available for calculating frictional losses in any combination of piping. Among the most recent are the tables in the *Hydraulic Institute Engineering Data Book*⁶ along with other similar hydraulic engineering references.

Before any new installation is made, the cost of larger-size piping, which will result in lower pump pressures, should be carefully balanced against the cost of a less expensive pump, a smaller motor, and a savings in power over the expected life of the system. The larger piping may cost a little more in the beginning, but the ultimate savings in power will often substantially offset the original cost. These facts are particularly true for extremely viscous fluids.

Foundation and Alignment The pump should be mounted on a smooth, solid foundation readily accessible for inspection and repair. It is essential that the driver shaft and the pump shaft are in proper alignment. The manufacturer's recommendation of concentricity and parallelism should always be followed and checked occasionally. For high temperature applications the alignments should be checked with the equipment at operating temperature.

The suction pipe should be as short and straight as possible with all joints airtight. It should not contain places where air or other entrapped gases may collect. If it is not possible to have the fluid flow to the pump under gravity, a foot or check valve should be installed at the end of the suction line or as far from the pump as possible. All piping should be independently supported to avoid strains on the pump casing.

Start-Up A priming connection should be provided on the suction side, and a relief valve should be set from five to ten percent above the maximum working pressure on the discharge side. Under normal operating conditions with completely tight inlet lines and wetted pumping elements, a screw pump is self-priming. Starting the unit may involve simply

opening the pump suction and discharge valves and starting the motor. It is always advisable to prime the unit before the initial startup to wet the screws. In new installations, the system may be full of air, which must be removed. If this air is not removed, the performance of the unit will be erratic, and, in certain cases, air in the system can prevent the unit from pumping. Priming the pump should preferably consist of filling not only the pump with fluid but as much of the suction line as possible.

The discharge side of the pump should be vented at startup. Venting is especially essential when the suction line is long or when the pump is initially discharging against the system pressure.

If the pump does not show a discharge of liquid after being started, the unit should be shut down immediately. The pump should then be primed and tried again. If it still does not pick up fluid promptly, there may be a leak in the suction pipe, or the trouble may be traceable to an excessive suction lift from an obstruction, throttled valve, or another cause. Attaching a gage to the suction pipe at the pump will help locate the trouble.

Once the screw pump is in service, it should continue to operate satisfactorily with practically no attention other than an occasional inspection of the mechanical seal or packing for excessive leakage and a periodic check to be certain that the alignment is maintained within reasonable limits.

Care should be taken to ensure that the suction pipe is free of any solid contaminants, left over from the construction phase. Flushing of the suction pipe before connecting to the pump is recommended. In some applications, suction strainers are installed to prevent solid particles from reaching the pump. If required, the strainer can be removed after the system start-up phase is completed.

Noisy Operation Should the pump develop noise after satisfactory operation, this is usually indicative of an excessive suction lift resulting from cold liquid, air in the liquid, misalignment of the coupling, or, in the case of an old pump, excessive wear.

Shutdown Whenever the unit is shut down, if the operation of the system permits, both the suction and discharge valves should be closed. This is particularly important if the shutdown is for an extended period because leakage in the foot valve, if the main supply is below the pump elevation, could drain the oil from the unit and necessitate repriming as in the initial starting of the system.

Abrasives One other point has not yet been discussed, and this is the handling of liquids containing abrasives. Since screw pumps depend upon close clearances for proper pumping action, the handling of abrasive fluids usually causes rapid wear. Much progress has been made in the use of harder and more abrasive-resistant materials for the pumping elements so that a good job can be done in some instances. It cannot be said, however, that performance is always satisfactory when handling liquids laden excessively with abrasive materials. On the whole, screw pumps should not be used for handling fluids of this character unless a shortened pump life and an increased frequency of replacements are acceptable. In these cases, reducing the operating speed can maximize the operating life of the pump.

CONCLUDING REMARKS

As indicated, screw pumps are manufactured in a number of different configurations and designs to suit a variety of different applications. Generally, screw pumps should be considered if there is a requirement for high pressure, high viscosity, or low hydraulic pulsation levels. Screw pumps can be used with lubricating and non-lubricating products and on many special and difficult applications. The keys to the success of screw pump applications include accurate knowledge of the fluid characteristics, proper pump selection and sizing, correct system design, suitable installation conditions, and the proper operating and maintenance procedures.

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SECTION 3.8

VANE, GEAR, AND LOBE PUMPS

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DEFINITIONS AND NOMENCLATURE

Vane, gear, and lobe pumps are positive displacement rotary pumps. The Hydraulic Institute defines them as mechanisms consisting of a casing with closely fitted vanes, gears, cams, or lobes that provide a means for conveying a fluid. Their principle motion is rotating, rather than reciprocating, and they displace a finite volume of fluid with each shaft revolution. When describing them, the general term *fluid* is used, rather than the more restrictive *liquid*. Fluid, in this case, is understood to include not only true liquids, but mixtures of liquids, gases, vapors, slurries, and solids in suspension as well.

How They Work Pumping in a vane, gear, or lobe pump begins with the rotating and stationary parts of the pump defining a given volume or cavity of fluid enclosure. This enclosure is initially open to the pump inlet but sealed from the pump outlet and expands as the pump rotates. As rotation continues, the volume progresses through the pump to a point where it is no longer open to the pump inlet but not yet open to the pump outlet. It is in this intermediate stage where the pumping volume or cavity is completely formed.

Depending on the particular pump, there can be more than one cavity in existence at any one time. As this happens, fluid also fills the clearances between the pumping elements and pump body, forming a seal and lubricating the pumping elements as they in turn pump the fluid. Rotation continues and the cavities progress, moving fluid along the way. Soon a point is reached where the seal between the captured fluid volume and outlet part of the pump is breached. At this point the vanes, gears, or lobes force the volume of captured fluid out of the pump. While this is happening, other cavities are simultaneously opening at the inlet port to receive more fluid in a continual progression from suction to discharge ports.

For optimum pumping action, the *open-to-inlet* (OTI) volume should expand slowly and continuously with pump rotation. The *closed-to-inlet-and-outlet* (CTIO) pumping cavity volume should remain constant once it is formed, and the *open-to-outlet* (OTO) volume

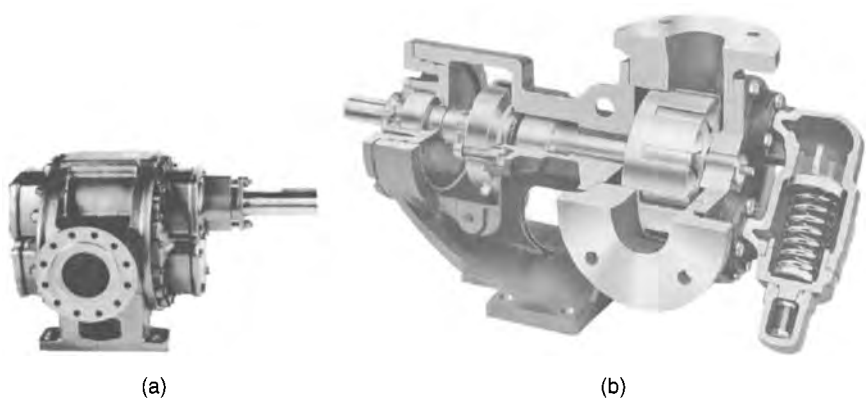


FIGURE 1a, 1b Typical rotary gear pumps (1b: Viking Pump)

should expand slowly and continuously with pump rotation. At no time should any fluid in the pumping chambers be simultaneously open to both the inlet and outlet if the pump is truly a positive displacement pump. When these conditions are met, the result is a very smooth continuum of flow with minimal pulsations or pressure spikes.

With rotary pumps (see Figure 1), a driver turns one shaft and rotor assembly, which in turn physically meshes with another to form the cavities that move the fluid. This is known as an *untimed arrangement*. For some applications, however, there would be problems with the gears meshing this way. For instance, stainless steel gears will gall and seize if rubbed against each other.

To circumvent this, the timed pump was developed. It uses timing gears physically located outside the pumping chamber to transmit torque between the pump shafts and synchronize the pumping elements relative to each other. By preventing them from contacting each other, they eliminate many of the problems of material compatibility, and dry running. Lobe pumps (Figure 2) are built this way.

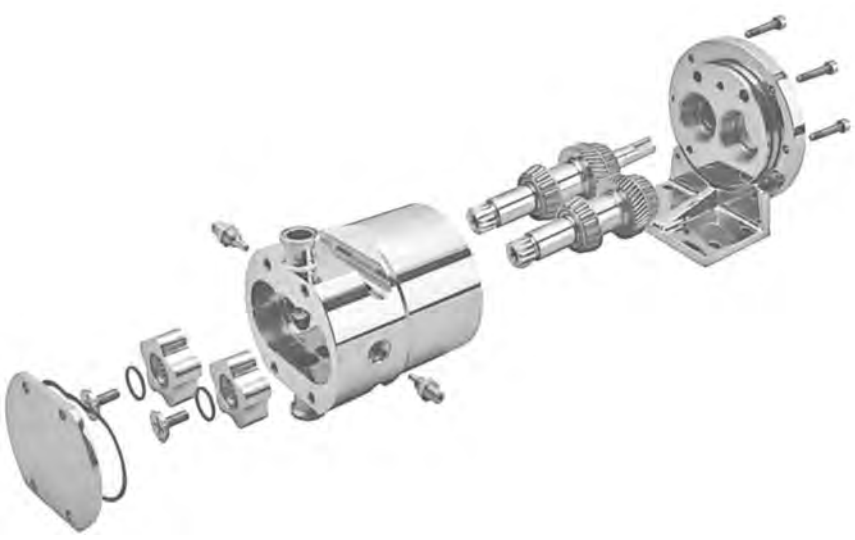


FIGURE 2 Lobe pump (Viking Pump)

In some special cases, pressure-balancing designs, relief valve arrangements, or other such considerations have led to some designs where the pump must only operate in one direction. However, the principle of operation of most rotary pumps permits them to operate equally well in either direction.

Main Components The pumping chamber of a rotary pump is the area containing the pumped fluid while the pump is operating. Fluid enters the pumping chamber through the inlet port and leaves through the outlet port. The body is that part of the pump that surrounds the boundaries of the pumping chamber and is also referred to as a *casing*, *housing*, or *stator*. *End plates* can either be part of the body or separate parts, and they serve to close off the ends of the body to form the pumping chamber. End plates can also be referred to as *pump covers*.

The rotating assembly generally refers to all those parts that rotate when the pump is operating. *Rotors* are usually given more descriptive names, depending on the specific pump type. All rotary pumps employ a drive shaft to accept driving torque from the pump driver. The majority of rotary pumps are mechanically coupled to their driver with various types of couplings, but sealless magnetically driven pumps have become more common in recent years.

The cavity through which the drive shaft protrudes is called the *seal chamber*, and leakage through it is controlled by a *mechanical seal* or *packing*. In a mechanical seal, two faces with opposing axial loads are maintained in close contact with each other. When compressible packing and a stuffing box are used in place of a mechanical seal and seal chamber, the packing is compressed in the stuffing box by a gland that keeps it in intimate contact with the stationary and rotating elements. A *lantern ring* or *seal cage* is often placed between two of the packing rings to enable cooling and lubrication from an external source.

A number of other auxiliary devices and arrangements can be found in vane, gear, and lobe pumps, but two are especially characteristic of these pump types. Given the positive displacing nature of these pumps, and the potentially high pressures that can result at the outlet of the pump if there is an obstruction or blockage, *safety relief valves* must be used with all positive displacement pumps. They limit the pressure by opening an auxiliary passage at a predetermined set pressure and relieving flow back to the inlet side of the pump or, better, to the fluid's original source. They can be installed externally or serve as an integral part of the pump. One exception is flexible member pumps, which, by the nature of their design, usually do not require one, due to the resiliency and expandability of their elastomeric components.

Nonetheless, most rotary pumps are available with integral relief valves. Although these devices are a necessary investment for safety, they are not a substitute for an external system relief valve. They are a secondary safety device at best and are not intended for continuous duty, flow control, or system pressure modulation.

To reduce fluid viscosity in the pump body to facilitate a successful startup and maintain event-free operation, *heating jackets* are used. They can be either integral to the pump (either welded on or part of the body casting) or a separate bolt-on type. They are common with asphalt, gelatins, paraffin, molasses, greases, and similar fluids where, without heating, the power and torque required to drive the pump could easily overload the driver. If viscous enough, a cold startup could actually destroy the mechanical seal, shear a shaft coupling, break a drive shaft, or cause damage to the equipment some other way. Heating jackets are not intended to be the main source of heating in the system. If the system requires the fluid to be heated, some other means, such as heat tracing the system piping, must be used.

Materials of Construction When selecting materials for rotary pumps, consideration must be given to the following material properties:

- The modulus of elasticity (for deformation purposes)
- The coefficient of thermal expansion (for varying temperatures)
- The coefficient of friction (for resistance to galling when in sliding contact)

For rotary pumps with flexible members, further consideration must be given to the materials' bulk modulus for recovery from deformation.

The close running clearances of rotary pumps require that their materials resist deformation and deflection by the various forces present when the pump is operating. If they do not, then such deformation or deflection could open the clearances and lower the operating efficiency dramatically or close the clearances and cause high mechanical loading and seizing between the moving and stationary parts.

The materials must also have compatible coefficients of thermal expansion. With the potential for deflection of the rotating parts always present, the materials selected must also have good bearing characteristics or, in particular, a good so-called "PV-value" (defined below) to resist galling and wear. This is especially important when pumping low-lubricity fluids.

Furthermore, materials used for corrosion resistance in noncontacting surfaces of centrifugal pumps may be unusable in rotary pumps where the continuous sliding contact between parts can wear away their passivating or protective layers. In general, rotary material restrictions become more severe when handling low-viscosity fluids at higher pressures and/or low lubricity fluids with abrasives. Most rotary pumps do not handle solids well, except for small trace amounts. Progressing cavity pumps, however, (see Section 3.7) are an exception and can handle very high solids content. In addition, even where there is no load-bearing contact between the rotating and stationary parts under normal conditions, the high transient forces generated at startup, shutdown, or any other unusual operating conditions (such as cavitation) must be considered when selecting pump materials.

The performance of flexible member pumps heavily depends on the material of the flexible member. Its bulk modulus must be high enough to keep distortion under pressure within functional limits and it must be resilient enough to spring back to its original shape after flexing or compressing. For instance, if, once deflected, the vanes in a flexible vane pump stayed that way, the pump could no longer operate. That is, these materials must be chosen not only to satisfy the desired hydraulic conditions, but also for resistance to deterioration from fatigue, chemicals, and the temperatures to which they may be exposed.

Vane Pumps Two basic types of vane pumps exist. The most common is the rigid sliding metal vane type, and the other is the flexible or elastomeric vane used for low viscosity (e.g., gasoline) or chemically aggressive fluids. Both are based around external sliding vanes rotating about a nonconcentric cam.

All rigid vane pumps have moveable sealing elements in the form of nonflexing blades, rollers, buckets, scoops, and so on. These elements move radially inward and outward by cam surfaces to maintain a fluid seal between the OTI and OTO sectors during pump operation. When the cam surface is internal to the pump body and the vanes are mounted in or on the rotor, the pump is called an *internal vane pump*. The OTI volume is defined by the body walls, the rotor walls, the fluid seal contact between the vanes, and the body. The body wall surface, the rotor surfaces, and the vane-to-rotor and vane-to-body fluid seal points define the CTIO volume. The body surface, the rotor surface, the vane-to-body fluid seal points, and the vane-to-rotor fluid seal points define the OTO volume.

In internal vane pumps, the volume behind the vanes must always be either a composite constant volume or else be vented, because of the piston-like pumping action of the vanes on the fluids trapped there. However, no such venting is required when the vanes are in the form of rocking slippers.

When the cam surface is external to the radial surface of the rotor and the vane, or the vanes are mounted in the body or stator, the pump is called an *external vane pump* and is illustrated in Figure 3. The OTI, CTIO, and OTO volumes are defined the same as for internal vane pumps when multiple external vanes are used. In this case, the rotor surface, the body surface, and the fluid seal points between them define the CTIO volume.

In addition to rigid, sliding metal vane pumps, flexible or elastomeric vane type pumps also exist. This kind of pump, illustrated in Figure 4, has a pumping action similar to that of an internal vane pump with the OTI, CTIO, and OTO volumes defined by the rotor surfaces, the body surfaces, the fluid seal contacts between the rotor flexible vanes, and the body surfaces.

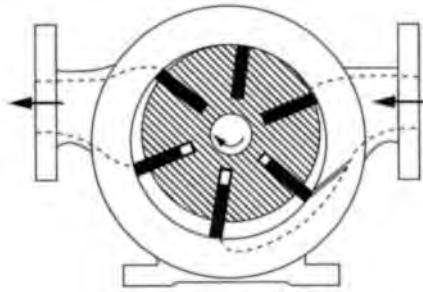


FIGURE 3 A typical external vane pump

The flexible liner pump in Figure 5 is similar in pumping action to the external vane pump, and all three chamber volumes of it are defined by the inner surface of the body, the outer surface of the liner, and the liquid seal contact between the liner and body bore. Most flexible liner pumps, unlike other rotary pumps, have at least one position of the rotor in which no fluid seal exists between the OTI and OTO volumes. The pump depends only on fluid velocity and inertia to limit backflow during this phase of rotation.

Vane pumps offer flows at up to 1,000 gpm (3,785 l/min) and pressures at up to 125 lb/in² (8.6 bar). They are commonly used for low-pressure transfers of gasoline, kerosene, and similar light hydrocarbons.

Gear Pumps The gear pump is the most common of all rotary pumps due to the wide variety of applications it can be used in.

Gear pumps have two gears that mesh to provide its pumping action, with one gear driving the other. The physical contact between the gears forms a part of the moving fluid seal between the inlet and outlet ports. The outer radial tips of the gears and the sides of the gears form part of the moving fluid seal between the inlet and outlet ports. The gear contact locus moves along the tooth surfaces and jumps discontinuously from tooth to tooth as the gears mesh and unmesh during rotation. These two characteristics distinguish gear pumps from lobe pumps where the rotors (lobes) are incapable of driving each other, (timing gears doing that instead,) and the fluid seal contact locus between lobes moves continuously across all the radial surfaces of the lobes.

Gear pumps are classified as *external* or *internal* (see Figure 6). External gear pumps have their gear teeth cut on their external or outside diameter and mesh about their outside diameters. Bearings (which are often product-lubricated bushings) support the shafts at both ends with the gears located between the bearings. This resists shaft deflection and contact between the gears and casing wall.

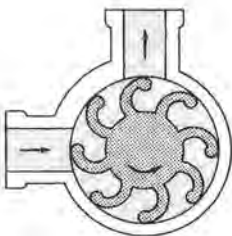


FIGURE 4 Flexible vane pump



FIGURE 5 Flexible liner pump

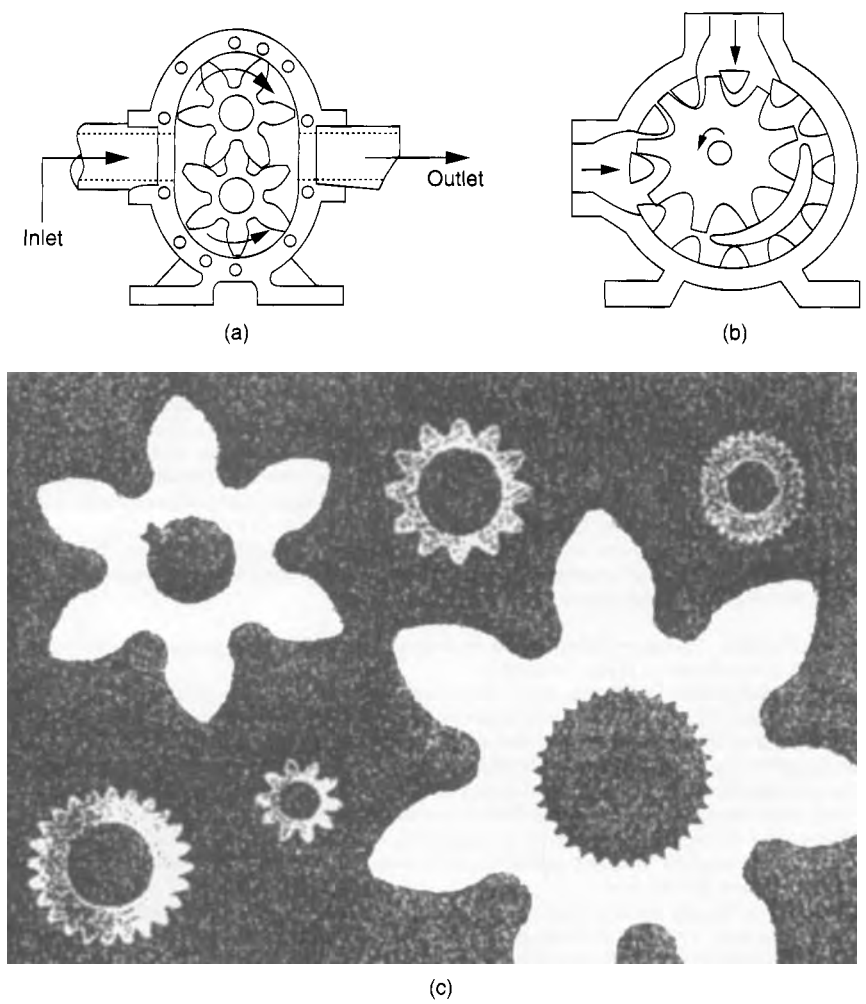


FIGURE 6 (a) external gear pump (Roper), (b) internal gear pump (Viking), and (c) various types of gears used (Roper)

Internal gear pumps, on the other hand, have one larger gear (rotor) with gear teeth cut internally on the major diameter meshing with and driving a smaller externally cut gear (idler). Pumps of this type can be with or without a crescent-shaped partition to define the OIT, CTIO, and OTO zones.

The OTI volume of the pump chamber in gear pumps is defined by the body walls and by where each tooth tip meets and seals with the body walls as it leaves the OTI volume. The fluid trapped between the gear teeth and the body walls is sealed from both inlet and outlet chambers and is the CTIO volume. The OTO volume is defined by the body walls and the gear tooth surfaces between the fluid seal points where each tooth tip leaves the body wall and enters the OTO volume and fluid seal points where the gears mesh.

A part or all of the side (or axial) surfaces of the gears run in small-clearance contact with the axial end faces of the pumping chamber. The gear teeth run in small-clearance

contact with each other where they mesh. The tips run in small-clearance contact with the radial surfaces of the pumping chambers in their travel from the OTI to OTO volume. Load-bearing contact between the rotors or between the rotors and the stator may exist in all three of these zones, and the apertures defined by the running clearances in these zones determine the amount of slip between the OTO and OTI volumes for any given pressure difference and viscosity between them.

Both gears share pumping torque, and the proportional amount of the total torque experienced by each gear at any instant is determined by the locus of the fluid seal point between the gear teeth. As this fluid seal point moves toward the center of gear rotation, the pumping torque on that gear increases, and as the seal point moves away from the center, the torque decreases. When external timing gears are used, they transfer torque from one rotating assembly to another to safeguard against accelerated wear when dry running or handling low-lubricity or somewhat more abrasive fluids (e.g., foodstuffs). Here, bearing (bushing) life depends on projected loading and journal velocity. A product of projected loading (P , psi) times journal velocity (V , ft/min) is used to determine the ability of internally (product) lubricated bearings to operate without significant wear. In the example given with Figure 7, the idler shaft of the gear pump is shown supporting its gear and is subject to the differential pressure, resulting in the projected bearing loading as illustrated in the figure. Data exist for various material pairs running against each other. Bronze, iron, carbon, and silicon carbide are some examples of popular materials used for the internally lubricated bushings of rotary pumps. For bronze, the PV-value

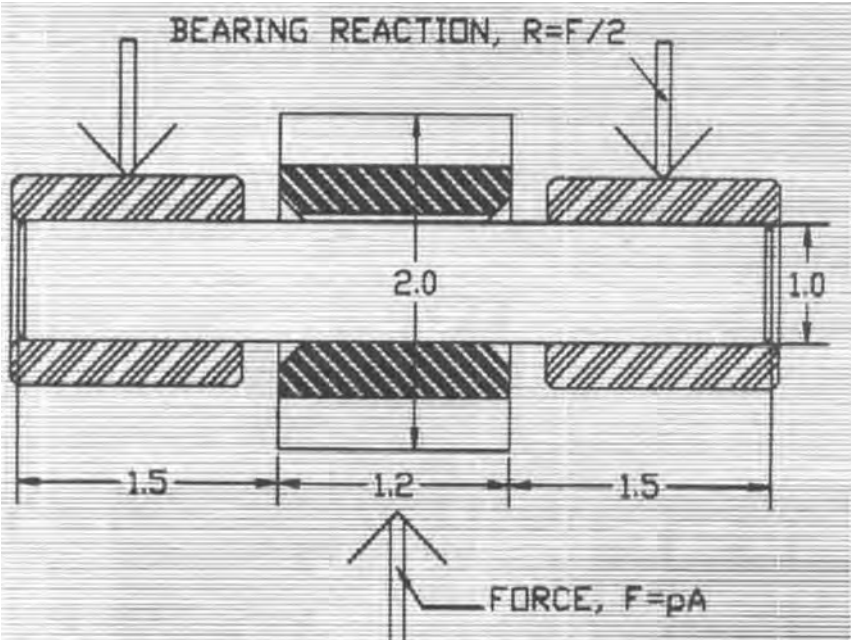


FIGURE 7 Example of dry-running PV-evaluation for a gear pump bushing. Given: rpm = 1800; suction pressure = 6 psia (0.414 bara); discharge pressure = 128 psig (8.825 barg); and dimensions are shown in inches (mm = in $\times$ 25.4). Thus, $\Delta p = (128 + 14.7) - 6 = 136.7$ psi (9.425 bar); and force on gear = $136.7 \times (2 \text{ in} \times 1.2 \text{ in}) = 328$ lbf (1459 N). Half of this force is shared by each bearing, namely 164 lbf (730 N). The unit load on each bearing $P = 164 / (1.5 \text{ in} \times 1 \text{ in}) = 109$ psi (7.54 bar). Since the journal velocity $V \text{ (ft/min)} = 2\pi \times \text{rpm} \times d \text{ (ft)} / 2 = 2\pi \times 1800 \times [(1 \text{ in} / 12) / 2] = 471 \text{ ft/min}$ (2.39 m/s), $PV = 109 \times 471 = 51,400$ psi-ft/min (18 bar-m/s). (Although the result is given here also in SI units, the PV-loading of a bearing is generally correlated in terms of psi-ft/min.)

is 60,000; for iron it is under 30,000; and carbon is rated at approximately 120,000. Some grades of silicon carbide reach PV-values exceeding 200,000. On the other hand, plastics have very low values of PV, typically under 3000.

External gear pumps are capable of flows up to 1,500 gpm (5,680 l/min), pressures up to 500 lb/in² (34.5 bar), and viscosities up to 1,000,000 SSU (216,000 centistokes). They are found in both clean and dirty services serving the *original equipment manufacturer* (OEM), refinery, tank farm, marine, and API-related industries. Internal gear pumps are capable of flows up to 1,100 gpm (4,165 l/min), pressures up to 225 lb/in² (15.5 bar), and viscosities up to 1,000,000 SSU (216,000 centistokes). They are typically used for lower pressure transfers of fuel oils, paints, and various chemicals in the chemical processing and OEM industries.

Lobe Pumps The lobe pump (Figure 2) receives its name from the rounded shape of the rotor radial surfaces that permits the rotors to continuously overlap each other as they rotate. Lobe pumps can be either single- or multiple-lobe pumps and carry fluid between their rotor lobes much in the same way a gear pump does.

Unlike gear pumps, however, neither the number of lobes nor their shape permits one rotor to drive the other, and so all true lobe pumps require timing gears. The body surfaces, rotor surfaces, the contact between rotors, and the contact between rotor lobe ends and the pump body define the OTI volume of a pump. The contact between the lobe ends and the body wall and the adjoining body wall and lobe surfaces define the CTIO volume. The body walls, rotor surfaces, lobe-to-body wall contacts, and the lobe-to-lobe contacts define the OTO volume.

In the two-rotor lobe pumps shown in Figure 8, the torque is shared by both rotors with the proportional amount of torque dependent on the position of the rotor-to-rotor contact point on the rotor contact locus. When the contact point is at the major locus radius (maximum lobe radius of one rotor in contact with the minimum lobe radius of an adjoining rotor), one rotor sees the full pumping torque, while the other rotor feels a balanced torque. The transfer of a full pumping torque from one rotor to the other takes place as many times in each complete revolution of a rotor as there are lobes on the rotor.

Lobe pumps are capable of flows up to about 1,000 gpm (3,785 l/min) and pressures up to 225 lb/in² (15.5 bar). They are commonly used to pump sludge in wastewater treatment plants and in stainless steel systems for handling foodstuffs in the food, beverage, dairy, and pharmaceutical industries.

A summary of the main application parameters of vane, gear, and lobe pumps is shown in Table 1 together with similar data for other rotary pumps; e.g., screw pumps (Section 3.7).

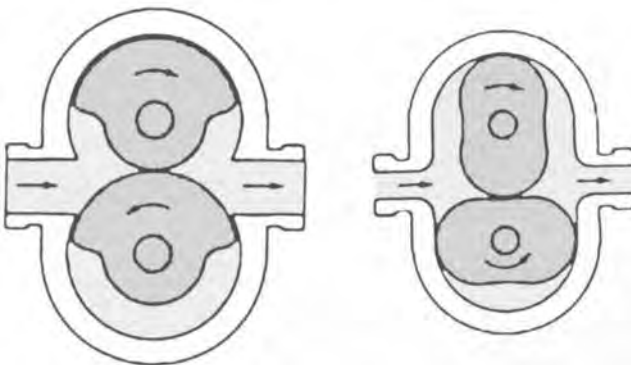


FIGURE 8 Typical single-lobe (left) and multiple-lobe (right) pumps

TABLE 1 Application Matrix for Rotary Pumps

Pump Type	Flow, gpm	Pressure, psi	Temperature, °F	Viscosity, cP	Solids, %	Dry-run	Corrosion resistance	Pulsations	Lift ft	Price index
External gear	1 to 1000	3000	400	20 to 10,000	<1%	N	G	G	15'	110%
Internal gear	1 to 1000	3000	400	10 to 10,000	<1%	N	G	G	15'	110%
Lobe	50 to 1000	400	400	100 to 500,000	<20%	Y	G	G	10'	150%
Vane	50 to 1000	200	250	0.3 to 1,000	<1%	N	M	M	10'	100%
Diaphragm	50 to 1000	100	200	1,000 to 100,000	<20%	Y	E	B	20'	130%
Progressing cavity	100 to 1000	500	200	1,000 to 100,000	<80%	N	B	E	15'	200
Two-screw	100 to 8000	1500	650	1,000,000	<1%	N	M	G	25'	400%
Three-screw	10 to 3500	4500	550	1,000,000	<1%	N	M	E	25'	300%
Metering (diaphragm)	<1	200	400	1 to 1,000	<1%	N	E	B	5'	100%

Note: m³/h = gpm/4.40; bar = psi/14.5; deg C = (deg F - 32)/1.8; Pa-s = cP/1000; m = 0.3048 × ft.) Values are approximate, for rough guidance only.
Code Letters: *Excellent*; *Good*; *Marginal*; *Bad*

COMPARISONS TO CENTRIFUGAL PUMPS

A rotary pump uses mechanical and hydraulic forces to create a flow against a system backpressure. Centrifugal pumps, on the other hand, create pressure by imparting a velocity to the fluid and converting the velocity energy to a pressure energy as the fluid flows around the casing and out the discharge nozzle. A comparison of the resulting performance of these two different pump types is shown in Figure 9.

The conditions of service will usually determine the best pump for an application. For instance, for varying flow rates, a centrifugal pump would be a good choice. An example of this is a municipal water system where consistent head is maintained. By contrast, for a constant flow in the presence of varying back pressures, a rotary pump would be better. An example of this is an oil pipeline, where system economics dictate constant flow rates, regardless of any system pressure variations from changes in viscosity or the pipe diameter or condition.

Other differences exist between centrifugal and rotary pumps as well. The performance curves, affinity laws, and terminology used to describe rotary pumps are all different. And since rotary pumps are primarily for viscous fluids, the applications and markets served by these two pumps are also different. Other factors, such as vertical versus horizontal mounting, metal versus nonmetallic materials, sealless and magnetically driven versus dual-containment mechanical seals and conventional drivers, are similar whether considering a centrifugal or rotary pump.

Rotary Pump Curves Centrifugal pump curves plot the flow on the X-axis with the discharge head on the Y-axis. However, rotary pumps develop the flow against a system backpressure, rather than developing head with a corresponding flow rate. Their performance curves therefore show the flow on the Y-axis with differential pressure along the X-axis, as shown in Figure 10.

The influence of differential pressure on the flow is greatest with lower viscosity. Conversely, the maximum viscosity represents the worst (most) case from a power standpoint and is therefore the point around which the driver is sized.

Instead, a rotary pump is selected for a given differential pressure and viscosity at the nearest commercially acceptable speed (with a tendency toward the lower speeds). For viscosities under 3,000 SSU (650 centistokes), this means at synchronous motor speeds with the flow rate falling at the design point or (ideally) slightly above it. For higher viscosities,

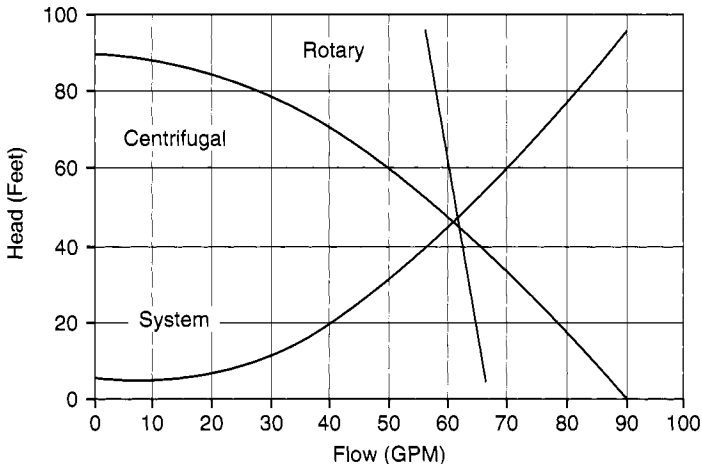


FIGURE 9 Typical performance curves ($m = ft \times 0.3048$; $m^3/h = gpm/4.403$)

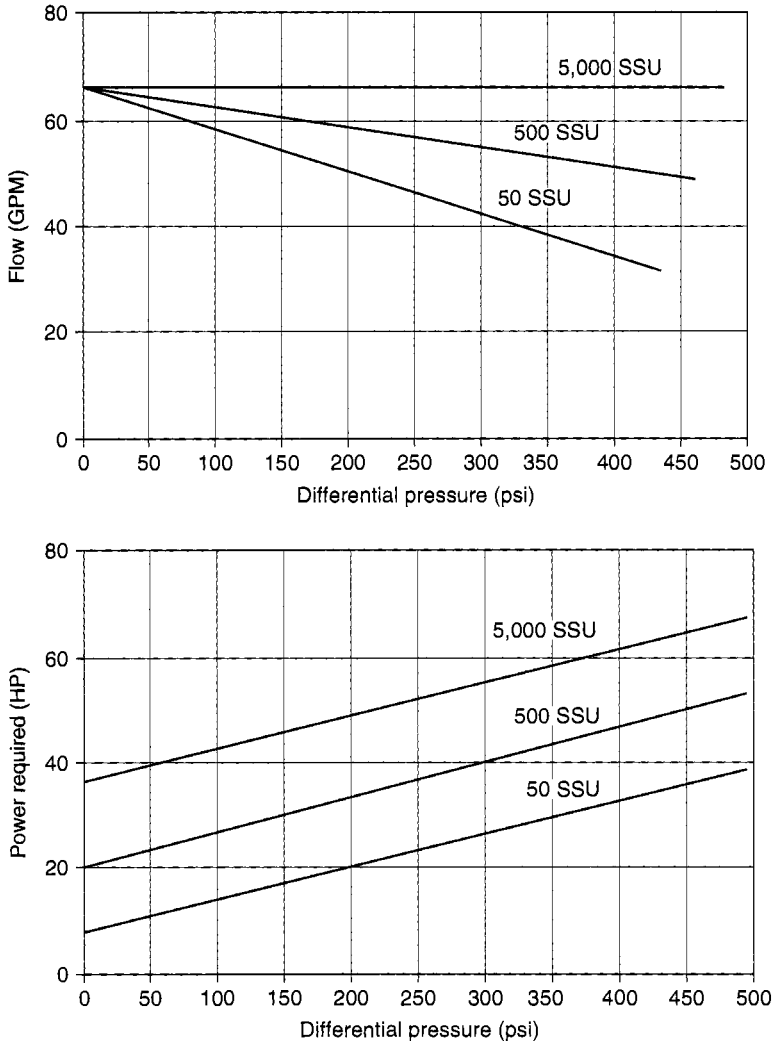


FIGURE 10 Typical rotary pump flow and power curves (bar = psi/14.5; kW = hp × 0.746; 50, 500, 5000 SSU = 7, 108, and 1080 centistokes respectively.)

rotary pumps will be run at reduced speeds, which in some cases can go well below 100 revolutions per minute (rpm).

Performance with Viscous Fluids Viscous drag can be considerable with higher viscosities. With an increasing viscosity, an increasing amount of energy must be expended to overcome these forces and produce the same amount of hydraulic work. At some point, viscosity will simply overtake the centrifugal pump, and it can no longer overcome the inertia and viscous drag losses of the fluid. The effect of this can be seen in Figure 11.

Since rotary pumps do not work this way, they are better suited for high-viscosity fluids. Certain rotary pumps, such as multiple screw pumps, can even go as high as 1,000,000 SSU

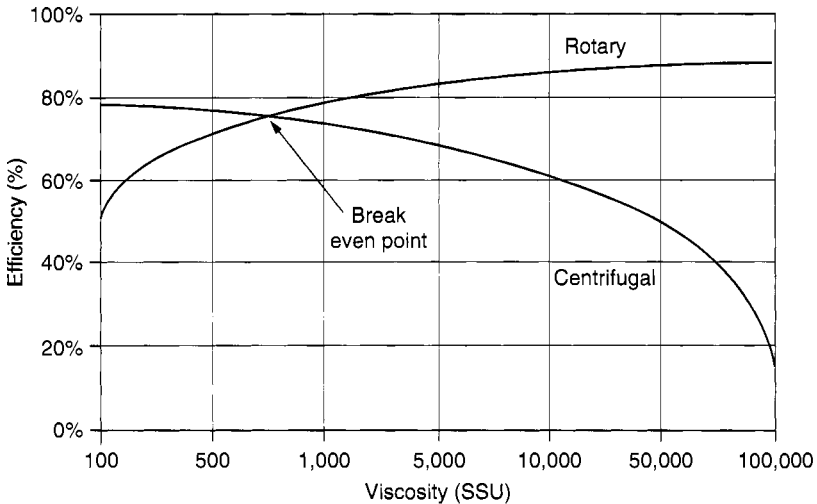


FIGURE 11 Typical efficiencies as a function of viscosity. (In this range, centistokes $\approx$ SSU/5.)

(216,000 centistokes) without a significant deterioration of performance or efficiency. Compared to centrifugal pumps, rotary pumps are less efficient at lower viscosities but more efficient at higher viscosities. From an applications standpoint, this crossover point will vary from pump to pump and depends on many factors, but as a rule of thumb, it usually falls between 500 and 1,000 SSU (108 and 216 centistokes).

Mechanical Installation All rotary pumps, particularly rigid rotor pumps, must be installed so no mechanical forces other than those imposed by pump-generated pressures can act to warp or distort the pump chamber or rotating assembly. This is important because relatively small distortions of a few thousandths of an inch can cause interference between rotating and stationary parts and generate high wear rates or pump damage.

To avoid such distortions, the pump must not be installed with overly long, rigid fittings or in such a way that the pump body supports the weight of the piping system. Large mechanical forces that distort the pump body may also cause distortions of the mechanical seals and accelerate wear of the bearings. A cardinal rule, then, is that a rotary pump must not be rigidly coupled to the piping system in a way that causes it to support the weight of the piping or otherwise exposes it to any forces from thermal expansion. Poor mechanical installation is the cause of many field problems.

Dry Running Most rotary pumps will be damaged if allowed to run dry. As long as the system provides a positive static pressure on fluid present at the inlet, the pump should prime and not run dry. If a negative inlet gage pressure, or suction lift, is present at the pump inlet, the installation should be checked to ensure airtight seals throughout the entire upstream piping.

The sealing arrangement on the pump must also be checked for any leakage of air or gas into the pump through the seal. If these conditions are satisfied, fluid will enter the pump soon after pump operation starts because of the vacuum generated by pump operation. If these conditions are not met, air or other gases will flow through leaks in the inlet system or seal to satisfy the pump flow rate requirements and the pump will run dry or partially dry.

Suction Strainers Like most other pumps, rotary pumps last longer when handling clean fluids. Nonetheless, this is an ideal scenario, and the pump will more realistically encounter dirty or abrasive-laden fluids of varying degrees. Fine particles and abrasives will cause wear in the close clearances of the pump, which eventually reduces pump flow rate by increasing the slip through the increased clearances.

As such, all rotary pumps should have a suction strainer to exclude larger materials such as welding slag, scale, rust, chips, rags, bolts, nuts, and so on. Since a suction strainer contributes to suction line losses, this reduces the net inlet pressure available. The finer the filtration, the greater the restriction and the more frequently it must be maintained. This leads to a trade-off between the cost of the added maintenance versus the cost of replacing the worn pump parts earlier than they would be otherwise.

When pumping fluids over 5,000 SSU (1080 centistokes), the finest strainer screen practical is a $\frac{1}{16}$ -inch (1.5 mm) perforation. Strainers and filters not only require periodic maintenance, but should also be instrumented accordingly. It is important for the user to provide some means of monitoring, such as a differential pressure gauge or switch, since a clogged strainer will cause the pump to cavitate or even run dry.

Entrained Air and Dissolved Gases An important consideration with rotary pumps is the amount of entrained air or gas in the fluid. It is generally neglected, since a rotary pump cannot become vapor-bound the way a centrifugal pump can. Nonetheless, with entrained air present, there can be a loss in the outlet flow. If the entrained air is a large enough percentage, there may be unacceptable noise and vibration levels as well.

For example, if a fluid contains 5% entrained gas by volume and the suction pressure is atmospheric, the mixture is 95% liquid and 5% gas. This mixture fills up the moving voids on the inlet side, with 5% of the space filled with gas and the remainder with liquid. Therefore, in terms of the amount of liquid handled, the output is reduced directly by the amount of gas present, or 5%. Unless this is understood up front, it could lead to a less than satisfactory output flow rate through no direct fault of the pump itself.

Entrained air is common in systems where the liquid is cycled frequently. In many cases, the foaming or air entrainment cannot be avoided, such as with the lubrication system on a large reduction gearbox. Instead, the condition must be known and well understood before selecting a pump for the application.

If dissolved gases (gases different than the fluid's own vapor) are present in the fluid, the effect on the output flow is the same as with entrained gases. This is because the dissolved gases will come out of solution when the pressure is lowered, just as the fluid's own vapor will. This will have the same net effect as the entrained gas and will occupy the available displacement capacity. Although the fluid mass transfer rate will not be affected, this is likely to be small comfort since the measured liquid displacement will be reduced.

Noise Pumps are often the most offensive noise sources in hydraulic machinery. High-pressure pulsations and heavily loaded sliding elements within the pump produce broadband, high-energy airborne noise. Vane, gear, and lobe pumps (as well as the screw pumps of Section 3.7), however, are among the lowest noise producers of any fixed displacement-type pump. Flow is delivered continuously without the variations that produce noise in conventional hydraulic pumps. Pumping elements utilize a fluid film, reducing the sliding contact, and the visco-elastic properties of the fluids they pump help dampen whatever fluid-borne pressure pulsations are present. These design features are responsible for the wide use of these pumps wherever noise is critical. For instance, they are widely installed on die-casting machines, plastics equipment, presses, and an enormous variety of machine drives and machine tools.

Inlet Pressure The absolute pressure above the vapor pressure *available* at the pump inlet must always exceed the absolute pressure above the vapor pressure *required* by the pump.

Rotary pumps are not effective at stopping the flow through them. In applications where the flow cannot be permitted through an idle pump, or where inlet or outlet static

pressure heads exist, check valves in the system must be used to stop the flow. For example, in intermittent deliveries where the pump is “lifting” liquid from a source below its inlet without any valving in place to address this when the pump is stopped, the fluid will gradually drain backward through the pump and back to its source.

Cavitation The creation of vapors and the subsequent collapse of the vapor bubbles upon reaching the higher-pressure discharge side of the pump is known as *cavitation*. Cavitation forces the liquid into the vapor voids at high velocities and produces local pressure surges of high intensities impinging on the pump surfaces. These forces can exceed the tensile strength of the metal, eroding it in the process. Left uncorrected, cavitation can cause pitting of the vanes, gears, or lobes and interior casing walls; bearing failures; and even shaft breakage. In addition, cavitation causes noise, vibration, and a loss of output flow. The bigger the pump, the greater the noise and vibration can be.

Lowering the static suction lift of a system, increasing the suction pipe diameter, and simplifying the suction piping layout can reduce cavitation by raising the net inlet pressure or NPSH available. In reality though, the net inlet pressure available is not something that can easily be altered. Instead, the pump vendor will usually be asked to select a pump with a lower net inlet pressure required. The best way to do this is to select a larger pump running slower. This will give the fluid more residence time to fill the void on the suction side, and with larger internal passages and ports, pump entry losses will be reduced as well.

FLUID CONSIDERATIONS

Very few commercially handled fluids are homogenous liquids. In actual systems, most fluids have air or other gases dissolved or entrained in them. In other cases, solids (abrasive or nonabrasive) may be in the fluid. The variety of fluids handled by rotary pumps requires that each pump application be uniquely considered in terms of the effects of the pumped fluid on pump performance. General summaries of the effects of various fluid characteristics on pump performance are covered in the following paragraphs.

Temperature The temperature of the pumped fluid affects pump performance in three main ways. If the pump is to handle fluid at temperatures considerably different from ambient ones, the materials of construction, both in the pump and in the seals, and the operating clearances of the pump must be selected to provide the desired operating characteristics at the temperature of operation. The selection process becomes even more stringent when the pump is operated over a wide range of temperatures. Such a use may preclude the use of pump construction materials having high thermal coefficients of expansion. Furthermore, in the actual application, it may be necessary to preheat or pre-cool the pump to the operating temperature before the pump is started in order to avoid thermal shocking the internal components when the fluid enters the pump. If this is not done, the resulting rapid heating or cooling of the pump members from the inside out can damage the pump. Preheating or pre-cooling can usually be accomplished with a secondary medium such as a heating jacket.

Viscosity Another effect of fluid temperature is on the viscosity of the fluid. For the majority of the commercial liquids, the viscosity increases with decreasing temperature and decreases with increasing temperature. If the pump application requires usage over a wide temperature range, the highest viscosity at this temperature range must be known to determine whether the pump is operating below the upper speed limit imposed by this fluid viscosity. The effects of viscosity on speed and the net inlet pressure (above the vapor pressure of the liquid) required is shown in Figure 12.

Lubricity Still another important fluid characteristic is lubricity. Many rotary pumps are not designed to operate on liquids with no lubricity. Either the wear rate or the pump mechanical friction may be inefficiently high if these pumps are used on nonlubricating

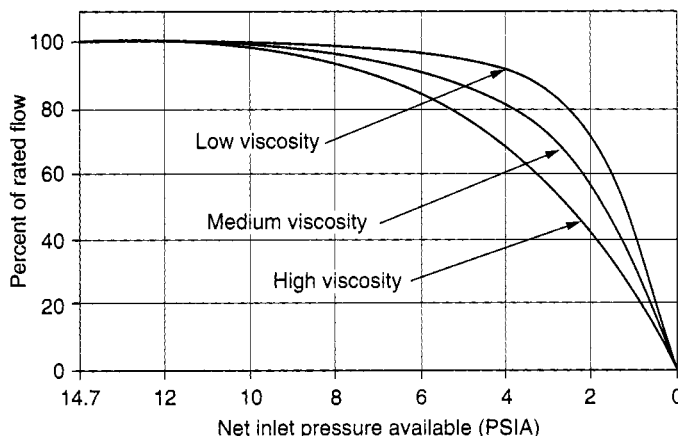


FIGURE 12 How viscosity and net inlet pressure available influence flow (bar = psi/14.5. Expressed as head of pumped liquid, the net inlet pressure is called NPSH.)

fluids. Vane pumps are the exception here; for they can handle even liquid CO_2 , a very poor lubricating substance.

Non-Newtonian Fluids The viscosity of most fluids is unaffected by agitation or shearing as long as the temperature remains constant. They are known as *true* or *Newtonian fluids*. Certain fluids, however, change viscosity as a shear is applied at a constant temperature. The viscosity of these fluids will depend upon the shear rate at which it is measured, and these fluids are termed *non-Newtonian*. Some examples are cellulose compounds, glues, greases, paints, starches, slurries, and candy compounds.

If a fluid is non-Newtonian, the viscosity under actual pumping conditions must be determined and can vary quite a bit from the viscosity under static conditions. An example is grease, where the static viscosity is around 20,000 SSU (4300 centistokes). Under actual pumping conditions, however, the viscosity is closer to 500 SSU (108 centistokes). Without realizing this, a larger, slower pump may have been offered when a smaller, less expensive pump running at a higher speed would have been acceptable.

Tests or computations should determine the effective fluid viscosity under actual operating conditions. If the pump is to operate over a range of speed and pressures, the maximum and minimum effective viscosities over this range should be known to allow for the additional slip or horsepower that may be required and to ensure the correct operating speed and pump driver power selections. If these data are unavailable, as is often the case, then data from a similar installation can be helpful. Ideally, the information will include the pump size, flow rate, and speed as well as the NPSH available and the pressure drop over a specific length of piping. With this information, assumptions can be made regarding the actual viscosity.

Corrosiveness Knowledge of the corrosiveness of the pumped fluid on the pump materials in contact with the fluid is important to satisfactory pump application. The clearances in rotary pumps are small, and corrosion rates of only a few thousandths of an inch per year may seriously affect the efficiency of the pump, particularly when it is handling low-viscosity fluids. In addition, general compatibility of the fluid with the materials of pump construction must be considered. For example, a certain solvent to be pumped may soften, dissolve, or destroy the elasticity of flexible members in the pump, including those flexible members used in the sealing arrangements. Hence, the effect of the pumped fluid on the physical and chemical state of materials used in pump construction should be considered when choosing a particular application.

Abrasiveness Abrasives present a problem for every pump type, and rotary pumps are particularly vulnerable. With small operating clearances, the tendency of the abrasive is to wear down and open the tight clearances of the pump. Most rotary pumps' internal clearances are long, narrow, rectangular cross sections that can be modeled as two parallel flat plates, with one plate stationary and the other moving. These clearances range from essentially zero to a few thousandths of an inch. Thus, even minor variations in manufacturing tolerances or the effects of wear over time can cause considerable variations in the percentage change of the aperture volume.

Because rotary pumps depend upon close clearances for their pumping action, whenever abrasive or dirty fluids are encountered, accelerated wear can be expected. Wear is difficult to predict because it depends on many variables that are difficult to measure (such as particle size, size distribution, or whether smooth or jagged shaped).

Classic theory tells us that laminar slip flow Q_s between two plates follows the general equation:

$$Q_s = k \left\{ \frac{P_d \times w \times d^3}{v \times l} \right\} \quad (1)$$

where Q_s = slip through the clearance, gpm (m^3/hr)

k = a constant

P_d = differential pressure, lb/in² (bar)

w = width of clearance, in (mm)

d = clearance gap or depth, in (mm)

v = absolute viscosity, lb-sec/ft² (centipoises) [Note: (absolute viscosity in centipoises) = (kinematic viscosity in centistokes) $\times$ sp. gr.]

l = length of clearance, in (mm)

Rotary pumps rely on close running clearances to seal between the suction and discharge pressures. This presents certain manufacturing trade-offs. For instance, the clearances must be tight enough to yield an acceptable volumetric efficiency with lower viscosities, yet large enough to facilitate internal lubrication with higher viscosities. For a given pump with a fixed viscosity and differential pressure, the slip flow is a function of the cube of the clearances only.

Unlike a centrifugal pump impeller, which can be trimmed to about 85% of its maximum diameter, rotary pump gears, screws, or lobes cannot be trimmed. As Figure 13 shows, up to a certain critical point, wear has little effect on the flow. Beyond this point, which varies from case to case, performance starts to deteriorate rapidly.

Much progress has been made in the use of harder and more abrasive-resistant materials for the pumping elements. However, rotary pumps should not be used for abrasive fluids, unless a decreased pump life and/or increased maintenance intervals are acceptable.

Other fluid characteristics could be important in the selection and application of vane, gear, and lobe pumps. Some liquids are sensitive to shear and must be handled in pumps with low shear rates. Others cannot tolerate exposure to the atmosphere, either because they can explode, crystallize, and damage the seal arrangement or otherwise interfere with the pump operation. Rotary pumps for these applications should have only static seals or multiple-section rotary seals with a protective liquid or gas in the seal zones between the fluid and outside atmosphere. In aseptic pumps, for example, stationary and moving seals can be designed with multiple seals to enable the use of steam under pressure as a sterile barrier between the pumping chamber and the atmosphere. Pumps in industrial applications where it could be dangerous if the fluid were exposed to the atmosphere may use multiple sealing arrangements with an inert liquid under pressure acting as a barrier between the pumping chamber and the outside atmosphere.

For the effective application of rotary pumps, the temperature, viscosity, lubricity, corrosiveness, and nonliquid content of the pumped fluid must be known, along with any special characteristics of the liquid, such as shear or atmospheric sensitivity. With these known, the proper pump type, seal arrangements, horsepower requirements, and speed can be determined.

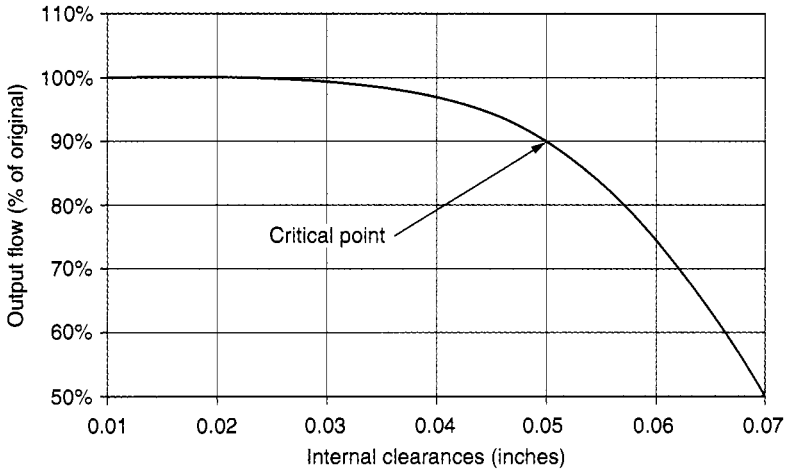


FIGURE 13 Output flow rate as a function of clearance size and wear

Vapor Pressure A second effect of fluid temperature is on the vapor pressure of the fluid. Vapor pressure is the pressure that must be maintained at a given temperature to prevent the liquid from partially vaporizing into a gaseous state. Vapor pressure increases with temperature, and if the local pressure falls below the vapor pressure, vaporization and cavitation will occur.

Vapor pressure is particularly important when handling hydrocarbons and petrochemicals, which can have very high vapor pressures. For instance, low-sulfur crude oil vapor pressures can be as high as 100 lb/in² absolute (6.89 absolute) under summer ambient temperatures. Unless this is known when selecting a pump, the results could be a pump that works in cold weather but cavitates in warm weather.

OPERATING CHARACTERISTICS

The operating characteristics covered in this section assume the fluid is a true incompressible liquid with a viscosity independent of the rate of shear (shear strain). Common fluids used in testing are either a light lubricating oil or cool water with small amounts of soluble oil added for lubricity.

Displacement The *theoretical* or *geometric displacement* q_0 of a rotary pump is the fluid volume transferred during each revolution. For any given pump, the displacement depends only upon the physical dimensions of the pump elements and the pump geometry and is independent of other operating conditions. In those pumps designed for variable displacement, the pump usually is rated at its maximum displacement. Thus, neglecting slip, the *theoretical displacement volume flow rate* of the pump Q_d is the *theoretical displacement per revolution* q_0 multiplied by the rotative speed or rpm; namely, $Q_d = q_0 \times rpm$.

Slip Flow slip, Q_s , is an important aspect of rotary pump performance. It is defined as that quantity of fluid that slips from the OTO volume to the OTI volume per the unit of time. Slip is a function of the clearances between the rotating and stationary members, the differential pressure between the OTO volume, the OTI volume, and the fluid viscosity. Hydraulically, it is equivalent to a bypass line from the pump outlet back to the inlet. The most common unit of measuring slip is U.S. gallons per minute (cubic meters per hour). Slip is essentially independent of *RPM*.

The major slip paths through the pump in the presence of a positive differential pressure are the clearances between the end faces of the rotors and the endplates of the pump chamber, and those between the outer radial surfaces of the rotors and the inner radial surfaces of the chamber. If the differential pressure across the pump remains constant during a revolution, then the instantaneous slip rate usually varies throughout the revolution. This variation in the slip is caused by the same effect that would be produced if the physical dimensions of the equivalent bypass around the pump were varied as a function of the angular rotation of the drive shaft. This is also one of the common causes of flow pulsation in rotary pumps. It is particularly dominant when pumping low-viscosity fluids at high pressures.

The average slip for any set of operating conditions can be found by measuring the flow rate from the outlet port (assuming an incompressible liquid) and subtracting that flow rate from the theoretical displacement flow rate Q_d that would otherwise be expected at those operating conditions. The paths can also vary considerably in length with the changing positions of the rotary and body-sealing surfaces during rotation.

The effect of pressure on the slip is complex. The primary effect is direct in that slip increases in direct proportion to pressure. However, several secondary effects should be considered as well. The first is the effect of pressure differences across the pump on the dimensions of the slip path. This occurs because of the deflection of pump elements as a function of pressure. This is relatively small in rigid element pumps but can be significant in flexible vane pumps where the pressure can cause the vanes to flatten out and move away from the body walls. In addition, while the slip may increase in flexible member pumps at high pressures, in rigid rotor pumps it can actually decrease as clearances close, due to the high-pressure deflection of the rotors.

Another secondary consideration is the indirect effect of pressure on the fluid velocity through the slip paths. At any given viscosity, the flow through these paths can have the characteristics of turbulent, laminar, or slug flows. The majority of practical applications would require that the slip be a minor percentage of the pump displacement. To remain so, the velocity of fluid flow through slip paths would normally be in the laminar flow region, and the slip would then be directly proportional to the pressure difference. A pressure increase could cause a change to turbulent flow and a corresponding change in the slip as a function of pressure.

Also, an indirect effect of pressure exists on the effective compression ratio of compressible fluids. The compression ratio reduces the amount of net volume flow through the outlet port relative to the displacement of the pump. Although not a true slip in the sense discussed up to this point, the type of slip caused by this effect reduces the net volume delivered through the outlet port and consequently affects the volumetric efficiency. This effect is a secondary effect in most liquids but can become a large component of the slip in aerated or compressible liquids. An increase in the compression ratio caused by an increase in the pressure difference causes an increase in slip from this effect.

Flow Rate and Displacement The flow rate or *capacity* Q_c of a rotary pump is the net quantity of fluid delivered by the pump per the unit of time through its outlet port or ports under any given operating condition. When the fluid is incompressible, the flow rate is numerically equal to the total volume of liquid displaced by the pump per the unit of time minus the slip, all expressed in the same units. When a rotary pump is operating with zero slip, the theoretical or geometrical displacement Q_d of the pump becomes the flow rate Q_c . A common unit of flow rate is U.S. gallons per minute (cubic meters per hour):

$$Q_c = Q_d - Q_s \quad (2)$$

The theoretical displacement q_0 per revolution represents the fluid confined within the gear spaces and can be found by integrating the differential rate of a net volume transfer over one shaft revolution with respect to the angular displacement of the drive shaft through any complete planar segment taken through the pump chamber between the inlet and outlet ports. This integration depends on the gear shape, which is defined by the involute; and so the exact calculation of this volume is mathematically involved. Therefore, in most practical applications, an approximation can be made with reasonable accuracy, as the following example illustrates.

Gear pump displacement calculation (approximate method). Assume an external gear pump with two gears measuring 3 in (7.62 cm) OD, and 2 in (5.08 cm) ID (root), and 4 in (10.16 cm) gear width W . Assuming that the liquid occupies half of the volume between the OD and ID cylinders (the other half being the solid material of the teeth), the volume of fluid between OD and ID for the two gears is

$$2 \times \frac{1}{2} \times \frac{\pi}{4} \times (\text{OD}^2 - \text{ID}^2) \times W = 15.71 \text{ in}^3 (257.4 \text{ cm}^3) \text{ per revolution.}$$

Therefore,

$$q_0 = \frac{15.71}{231 \text{ in}^3/\text{gal}} = 0.068 \text{ gal/rev.} = \frac{257.4}{1,000,000 \text{ cm}^3/\text{m}^3} = 0.0002574 \text{ m}^3/\text{rev.}$$

For a rotative speed of 1000 rpm, the *theoretical displacement flow rate* Q_d (which does not account for slip) thus becomes

$$Q_d = q_0 \times \text{rpm} = 0.068 \times 1000 = 68 \text{ gpm} = 0.0002574 \times 1000 \times 60 \text{ min/h} = 15.44 \text{ m}^3/\text{h}$$

From a practical standpoint, pump displacement often cannot be computed precisely from the geometry of the pump and is instead established by testing. In either case, the manufacturer will usually state this because it is needed to determine the efficiency of the pump under various operating conditions.

Speed Considerations Centrifugal pumps typically run at synchronous motor speeds. The flow and head are fine-tuned by such things as trimming the impeller or underfiling the impeller vanes. Rotary pumps, on the other hand, operate with their internal dimensions fixed. Performance is usually fine-tuned externally by speed adjustments where the operating speed N is the number of revolutions of the driving or main rotor per unit time. When no gear reduction or increase exists between the drive shaft and the main rotor, the speed is measured or set at the drive shaft.

The most common unit of speed is rpm, and its direction is usually described as either clockwise or counterclockwise when viewed from the front or drive end of the pump looking from the driver.

Usually, no minimum speed exists for a rotary pump, but certain vane pumps that depend on a centrifugal force to draw the vanes out of their slots do have minimum speed requirements. Another exception is with low-viscosity fluids at high pressures where the slow speed enables slip to equal the theoretical displacement, resulting in no net output flow. To avoid this situation and assure the pump is running at a suitable volumetric efficiency, the pump should always run within its permitted speed range. Figure 14 shows a typical speed versus viscosity curve for a rotary pump.

Since flow is directly proportional to speed, it can be tempting to increase the pump speed above its maximum speed to obtain more flow. However, besides all the aforementioned concerns, one other item must be considered. With any rotary pump, as the cavities rotate, they present a void for the incoming fluid to fill. This void is available for a fixed amount of time, and the fluid in the suction chamber must accelerate to fill this void in the available time. The higher the fluid viscosity, the more energy is required to accelerate the fluid to fill the void. If the fluid cannot fill the void in the time available, a partial void and cavitation will occur. Recognizing a need to limit pump speed, most manufacturers provide charts showing the maximum speed, which decreases with increasing viscosity of the liquid. This speed is related to the required excess of the inlet total pressure P_1 above the vapor pressure p_v of the liquid (called NPSH when expressed in terms of the equivalent head of the liquid) increases with both the pitch line velocity V_p of the gears and the viscosity. Thus, if the available level of P_1 is limited, the pump speed is also limited accordingly. For external gear pumps, this relationship can be approximated by the formula:

$$P_1 - p_v = V_p^{0.8} \times \nu^{0.1}/61 \text{ psi (Multiply by 6.895 to obtain the answer in kPa)}$$

where ν is the kinematic viscosity in SSU, and

$$V_p = D_p \times \text{rpm} \times 3.14/n,$$

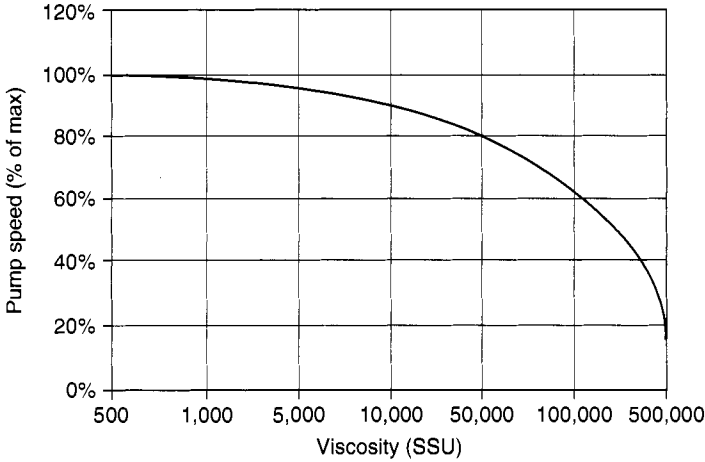


FIGURE 14 Typical speed reduction as a function of viscosity (In this range, centistokes = SSU/4.6)

and D_p is the pitch diameter of the gears in inches (= mm/25.4) and n_t is the number of teeth on a gear.¹

Pressure The absolute pressure of the fluid at any location in the pump, expressed in lb/in² (bar), is the total pressure there and the basis for all other pressure definitions associated with pump operation. Helping to simplify matters, the velocity pressure component of the fluid is usually small enough relative to the total pressure to be neglected.

The *inlet or suction pressure* p_{in} is the total pressure at the inlet to the pump. In common practice, the inlet pressure can be variously expressed as absolute pressure (lb/in² absolute or bar), as positive or negative gage pressure (lb/in² gage or bar), or as vacuum (inches or millimeters of mercury).

The total *differential pressure* Δp (or P_d) is the algebraic difference between the outlet pressure and the inlet pressure, with both expressed in the same units. This differential pressure is used in the determination of power input and in evaluating the slip characteristics of the pump:

$$\Delta p = p_{out} - p_{in} \quad (3)$$

The *net inlet pressure* $p_{net\ in}$ of a rotary pump is the difference between the inlet pressure expressed in absolute units and the vapor pressure of the fluid expressed in absolute units:

$$p_{net\ in} = p_{in} - p_{vapor} \quad (4)$$

Expressed as head of the pumped liquid, this is the net positive suction head, NPSH. The required NPSH or *net inlet pressure required* p_{nir} is the minimum net inlet pressure that can exist without the creation of enough vapor in the inlet to interfere with proper operation of the pump. The inlet pressure usually is not the true minimum pressure in the OTI volume. The flow of fluid in its passage through the OTI volume causes a fluid friction pressure loss, which causes the pressure to drop below the inlet pressure at some point in the OTI volume or inlet chamber. This loss increases with fluid velocity and hence with pump speed and fluid viscosity. It is a function of the pump geometry, which determines the fluid path lengths and local velocities where the fluid changes direction in flowing around curves or corners in the pump OTI volume boundary.

The required net inlet pressure is established by the manufacturer for particular speeds, pressures, and fluid characteristics. In practice, the pump user is warned that the

net inlet pressure is near or at the required net inlet pressure by noisy and rough operation of the pump caused by the incomplete filling of the CTIO volume with an accompanying reduction in pump flow rate.

The two other pressure ratings used are the maximum allowable working pressure, which is the maximum gage pressure at the outlet port permitted for safe operation. It is usually determined by the stiffness and strength of the pump body and the type of seals used. The maximum differential pressure is the maximum allowable difference between the outlet pressure and the inlet pressure, measured in the same units, and is determined by the capability of the rotating assembly and its fluid seal contact zones to withstand the pressure difference between the OTO and OTI volumes.

Efficiency and Power Overall efficiency (E_{tot}) for any type of pump is the ratio of the output fluid power P_{out} to the input shaft power P_s . In a development similar to that for centrifugal pumps in Section 2.1.1, E_{tot} can then be broken into three main component efficiencies; namely, volumetric, hydraulic, and mechanical. Volumetric efficiency (E_{vol}) is the ratio of ingested flow rate Q_{net} to the theoretical or displacement flow rate Q_d , where $Q_{net} = Q_d - S$, where S is the internal leakage flow rate (slip) through clearances back to the inlet. Thus the volumetric efficiency is defined in the same way as it is for screw pumps (Section 3.7). Hydraulic efficiency (E_{hyd}) is the ratio of the total fluid power P_{tf} of both the net flow and the slip flow to the actual power P_ℓ delivered to the fluid, where $P_{tf} = P_{out} / E_{vol}$. P_ℓ in turn is the sum of P_{tf} and the additional power required to overcome internal hydrodynamic friction P_{frict} . Mechanical efficiency (E_{mech}) is the ratio of the P_ℓ to P_s , where the shaft power $P_s = P_\ell$ plus the mechanical losses P_{mech} , namely the power due to bearing friction, seal drag, internal rubbing, and so on. For incompressible flow, $P_{out} = [\Delta p \text{ (psi)} \times Q_{net} \text{ (gpm)} / 1714] \text{ hp}$ or $[\Delta p \text{ (bar)} \times Q_{net} \text{ (m}^3/\text{h)} / 36] \text{ kW}$, where Δp is the pressure rise of the pump, and Q_{net} is also the flow rate leaving the pump.

As in the case of centrifugal pumps, the above definitions yield the following relationship of the component efficiencies with the overall efficiency:

$$E_{tot} = E_{vol} \times E_{hyd} \times E_{mech} \quad (5)$$

For rotary pumps, efficiency evaluation requires a significantly different perspective in comparison to centrifugal pumps. There are several reasons for that. Centrifugal pumps handle primarily low-viscosity liquids, typically under 100 cSt, while rotary pumps operate at much wider range of viscosities, often above 10,000 cSt and at times reaching even to 1,000,000 cSt. The Hydraulic Institute developed a chart (Figure 11 in Section 2.1.1) to correct centrifugal pump flow, head and power, when viscosities exceed normally very low, water-like values, but such cases are very rare, and thus a need to apply those corrections is rare. Also, centrifugal pumps can reach much higher power levels, as compared to rotary pumps, and thus overall efficiency (E_{tot}) evaluation, as it impacts energy consumption, is more important for centrifugal pumps. In contrast, rotary pumps are typically smaller (lower power), and their ability to deliver a constant (or nearly constant) flow rate, with little dependence on system restriction (relative insensitivity of flow to system pressure), is a consideration in the selection and application of rotary pumps. For these reasons, volumetric efficiency (E_{vol}) has historically played an important role in the world of rotary pumps, while overall (E_{tot}) efficiency is of primary importance in the world of centrifugal pumps.

The above efficiency definitions for rotary pumps are stated as follows in equation form. Volumetric efficiency is given by

$$E_{vol} = Q_{net} / Q_d = (Q_d - S) / Q_d \quad (6)$$

where

$$Q_d = \text{theoretical (ideal) flow, in the absence of slip,} = q_0 \times \text{rpm} \quad (7)$$

and where

Q_{net} = net flow rate leaving a rotary pump

S = slip (internal recirculation back to inlet)

q_0 = unit flow rate (constant for a given pump) per revolution

Overall efficiency is formally stated as

$$E_{tot} = P_{out}/P_s \quad (8)$$

where, assuming incompressible flow, useful fluid power is given by

$$P_{out} = [Q_{net} \text{ (gpm)} \times \Delta p \text{ (psi)/1714}] \text{ hp} = Q_{net} \text{ (m}^3\text{/h)} \times \Delta p \text{ (bar)/36 kW} \quad (9)$$

and total fluid power (useful plus that which drives the slip) is

$$P_{tf} = [Q_d \times \Delta p/1714] \text{ hp} = [Q_d \times \Delta p/36] \text{ kW} \quad (10)$$

The actual power P_{ft} delivered to the fluid is P_{tf} augmented by the internal hydrodynamic friction P_{frict} :

$$P_{ft} = P_{tf} + P_{frict} \quad (11)$$

and the input shaft power delivered to the pump is the total fluid power plus the mechanical losses:

$$P_s = P_{ft} + P_{mech} \quad (12)$$

Slip S and the viscous drag, P_{visc} , where,

$$P_{visc} = P_{frict} + P_{mech} \quad (13)$$

are determined (usually by testing) by pump manufacturers. These are specific to each design type and size and are published in pump catalogues as functions of pump speed, differential pressure, and viscosity.

Volumetric efficiency has numerical significance only when comparing similar designs (their comparative/competitive ability to maintain a constant flow rate with minimum slip), operating at the same speed. The reason for this is that, for rotary pumps, the ideal, theoretical, or displacement flow rate Q_d is directly proportional to pump speed, but slip does not depend on pump speed, and only depends on differential pressure, viscosity, and internal clearances. Consider, for example, a rotary pump delivering a required flow rate of 90 gpm (20.4 m³/h) at 1000 rpm, with 10 gpm (2.3 m³/h) internal slip (the ideal flow rate thus being 100 gpm or 22.7 m³/h). The volumetric efficiency of this pump is 90/100 = 90%. If this pump speeds up to 2000 rpm, its ideal flow will double (200 gpm or 45.4 m³/h), but the slip remains the same 10 gpm (2.3 m³/h), with a net flow rate of 200 - 10 = 190 gpm (43.2 m³/h), thereby yielding higher volumetric efficiency 190/200 = 95%, but too great a flow rate as compared to the required amount.

If a competitive pump were presented for evaluation, operating at the same speed (1000 rpm), same unit flow (q_o) but with smaller clearances and a slip of only 5 gpm (1.1 m³/h), then its volumetric efficiency would be (100 - 5)/100 = 95%, which makes it a more efficient design with regard to volumetric capability.

However, if the width of the gears of the 2000-rpm pump were reduced by half, its ideal flow would be reduced to 100 gpm (22.7 m³/h) with no change in slip, because end and radial clearances remain the same. Volumetric efficiency would then become (100 - 10)/100 = 90%. This altered 2000-rpm pump now has lower volumetric efficiency (a greater proportion of slip), but it is also less expensive due to the halved gear length. These facts would need to be evaluated together by the end user.

NUMERICAL EXAMPLE

The trends of the above example and the distinctions discussed between rotary pumps (gear, vane, or lobe) and centrifugal pumps become clearer in the following series of performance computations pertaining to a single rotary pump operating at two greatly differing viscosities (Cases 1 and 2) and at two greatly differing speeds at each of these two viscosities (subcases of Cases 1 and 2). Most rotary pump manufacturers publish pump performance curves showing four main relationships that enable these computations to proceed. Specifically, Figures 15 through 18 illustrate these relationships for the pump of this example.

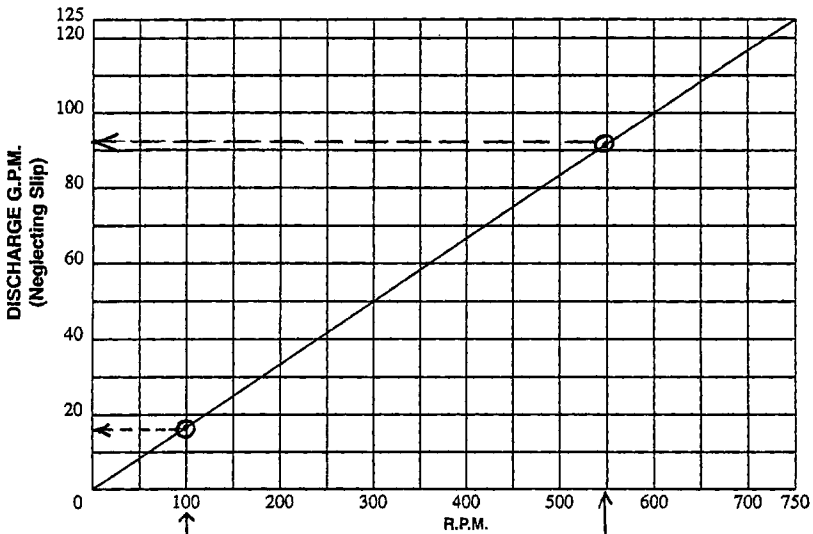


FIGURE 15 Ideal (theoretical) flow rate versus speed. This is typically a straight line, with a slope equal to pump unit flow q_0 . ($\text{m}^3/\text{h} = 0.227 \times \text{gpm}$) (Ranger, Inc.)

Figures 15 and 17 are plotted for convenience, because they can be alternatively calculated from Eqs. 7 and 10; but Figures 16 and 18 can be obtained only from the manufacturer's curves (internally tested values).

Case 1: Viscosity = 30 SSU (~ 1 cSt)

Subcase 1.1: Speed = 100 rpm. Displacement flow rate: $Q_d = q_0 \times 100 = 0.17 \times 100 = 17.0$ gpm ($3.86 \text{ m}^3/\text{h}$) (or read from Figure 15).

Results for $\Delta p = 25$ psi (1.72 bar)

Slip flow rate, $S = 4.2$ gpm ($0.95 \text{ m}^3/\text{h}$) from Figure 16

Net flow rate, $Q_{net} = 17.0 - 4.2 = 12.8$ gpm ($2.91 \text{ m}^3/\text{h}$)

Total fluid power $P_{tf} = \Delta p \times Q_d / 1714 = 25 \times 17.0 / 1714 = 0.25$ hp ($= 0.185$ kW)
(or read from Figure 17)

Useful output power, $P_{out} = \Delta p \times Q_{net} / 1714 = 25 \times 12.8 / 1714 = 0.187$ hp (0.139 kW)

Viscous power, $P_{visc} \approx 0.1$ hp (0.075 kW) from Figure 18

Shaft power, $P_s = P_{tf} + P_{visc} = 0.25 + 0.1 = 0.35$ hp (0.26 kW)

Volumetric efficiency, $E_{vol} = Q_{net} / Q_d = 12.8 / 17.0 = 75.2\%$

Overall efficiency $E_{tot} = P_{out} / P_s = 0.187 / 0.35 = 53.4\%$

Results for $\Delta p = 50$ psi (3.45 bar)

$S = 8.4$ gpm ($1.91 \text{ m}^3/\text{h}$)

$Q_{net} = 17.0 - 8.4 = 8.6$ gpm ($1.95 \text{ m}^3/\text{h}$)

$P_{tf} = \Delta p \times Q_d / 1714 = 50 \times 17.0 / 1714 = 0.496$ hp (0.370 kW)

$P_{out} = \Delta p \times Q_{net} / 1714 = 50 \times 8.6 / 1714 = 0.251$ hp (0.187 kW)

$P_{visc} \approx 0.1$ hp (0.075 kW)

$P_s = P_{tf} + P_{visc} = 0.496 + 0.1 = 0.596$ hp (0.445 kW)

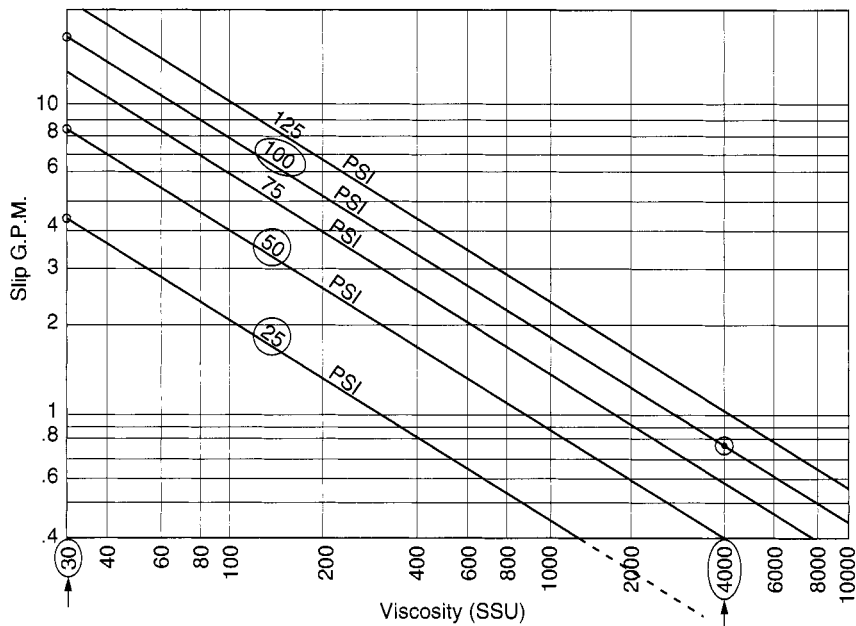


FIGURE 16 Relationship between internal slip and viscosity at several values of differential pressure. ($\text{m}^3/\text{h} = 0.227 \times \text{gpm}$ and $\text{bar} = \text{psi}/14.5$) (Ranger, Inc.)

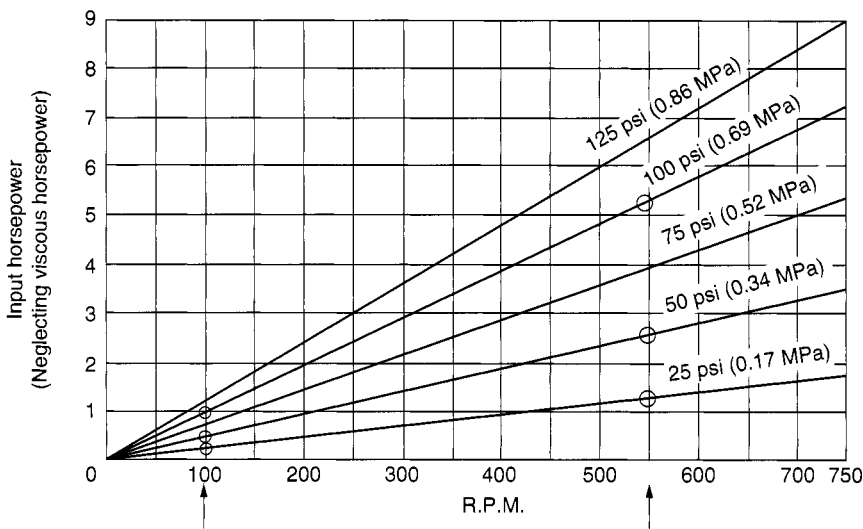


FIGURE 17 Total fluid power (minus power due to internal hydrodynamic friction or "viscous power") versus speed at several values of differential pressure ($\text{kW} = \text{hp}/1.34$, and $\text{bar} = \text{psi}/14.5$) (Ranger, Inc.)

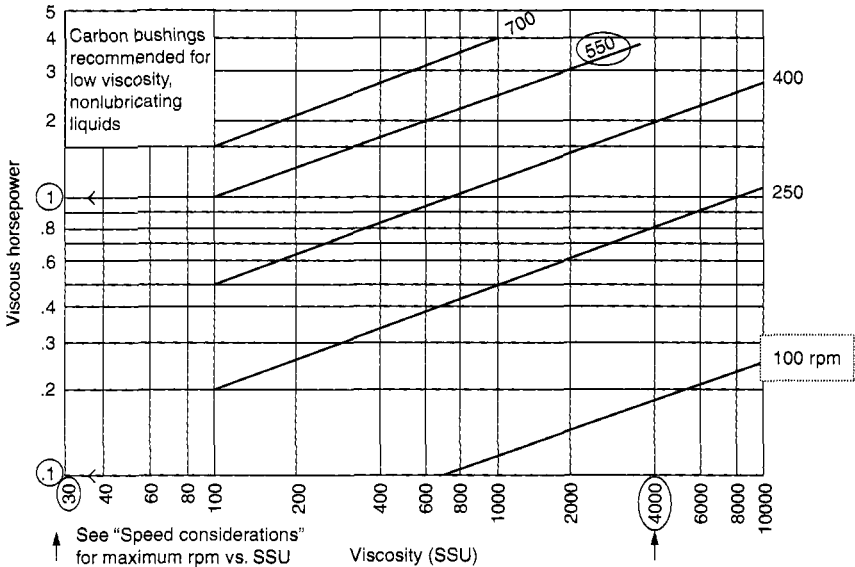


FIGURE 18 Viscous power loss versus viscosity, at several values of speed (kW = hp/1.34) (Ranger, Inc.)

$$E_{vol} = Q_{net}/Q_d = 8.6/17.0 = 50.6\%$$

$$E_{tot} = P_{out}/P_s = 0.251/0.6 = 41.8\%$$

Results for $\Delta p = 100$ psi (6.89 bar)
$$S = 16.8 \text{ gpm (3.82 m}^3\text{/h)}$$
$$Q_{net} = 17.0 - 16.8 = 0.2 \text{ gpm (0.045 m}^3\text{/h)}$$
$$P_{tf} = \Delta p \times Q_d / 1714 = 100 \times 17.0 / 1714 = 0.992 \text{ hp (0.740 kW)}$$
$$P_{out} = \Delta p \times Q_{net}/1714 = 100 \times 0.2/1714 = 0.012 \text{ hp (0.009 kW)}$$
$$P_{\text{visc}} \approx 0.1 \text{ hp (0.075 kW)}$$
$$P_s = P_{tf} + P_{frict} = 0.992 + 0.1 \approx 1.092 \text{ hp (0.815 kW)}$$
$$E_{vol} = Q_{net}/Q_d = 0.2/17.0 = 1.2\%$$
$$E_{tot} = P_{out}/P_s = 0.012/1.092 = 1.1\%$$

Observation: The slip S appears to vary linearly with differential pressure. This is not always the case for all gear pumps and is strictly coincidental in this particular example.

Subcase 1.2: Speed = 550 rpm. Displacement flow rate: $Q_d = q_0 \times 550 = 0.17 \times 550 = 93.5 \text{ gpm}$ (21.24 m³/h) (or read from Figure 15).

Results for $\Delta p = 25$ psi (1.72 bar)

$$S = 4.2 \text{ gpm (0.95 m}^3\text{/h)}$$
$$Q_{net} = 93.5 - 4.2 = 89.3 \text{ gpm (20.28 m}^3\text{/h)}$$
$$P_{tf} = \Delta p \times Q_d / 1714 = 25 \times 93.5 / 1714 = 1.364 \text{ hp (1.017 kW)}$$
$$P_{out} = \Delta p \times Q_{net}/1714 = 25 \times 89.3/1714 = 1.303 \text{ hp (0.972 kW)}$$

$$P_{visc} = 1.0 \text{ hp (0.746 kW)}$$

$$P_s = P_{tf} + P_{frict} = 1.364 + 1.0 = 2.364 \text{ hp (1.764 kW)}$$

$$E_{vol} = Q_{net}/Q_d = 89.3/93.5 = 95.5\%$$

$$E_{tot} = P_{out}/P_s = 1.303/2.364 = 55.1\%$$

Results for $\Delta p = 50 \text{ psi (3.45 bar)}$

$$S = 8.4 \text{ gpm (1.91 m}^3\text{/h)}$$

$$Q_{net} = 93.5 - 8.4 = 85.1 \text{ gpm (19.33 m}^3\text{/h)}$$

$$P_{tf} = \Delta p \times Q_d/1714 = 50 \times 93.5/1714 = 2.728 \text{ hp (2.035 kW)}$$

$$P_{out} = \Delta p \times Q_{net}/1714 = 50 \times 85.1/1714 = 2.482 \text{ hp (1.852 kW)}$$

$$P_{visc} = 1.0 \text{ hp (0.746 kW)}$$

$$P_s = P_{tf} + P_{frict} = 2.728 + 1.0 = 3.728 \text{ hp (2.781 kW)}$$

$$E_{vol} = Q_{net}/Q_d = 85.1/93.5 = 91.0\%$$

$$E_{tot} = P_{out}/P_s = 2.482/3.728 = 66.6\%$$

Results for $\Delta p = 100 \text{ psi (6.89 bar)}$

$$S = 16.8 \text{ gpm (3.82 m}^3\text{/h)}$$

$$Q_{net} = 93.5 - 16.8 = 76.7 \text{ gpm (17.42 m}^3\text{/h)}$$

$$P_{tf} = \Delta p \times Q_d/1714 = 100 \times 93.5/1714 = 5.455 \text{ hp (4.069 kW)}$$

$$P_{out} = \Delta p \times Q_{net}/1714 = 100 \times 76.7/1714 = 4.475 \text{ hp (3.338 kW)}$$

$$P_{visc} = 1.0 \text{ hp (0.746 kW)}$$

$$P_s = P_{tf} + P_{frict} = 5.455 + 1.0 = 6.455 \text{ hp (4.815 kW)}$$

$$E_{vol} = Q_{net}/Q_d = 76.7/93.5 = 82.0\%$$

$$E_{tot} = P_{out}/P_s = 4.475/6.455 = 69.3\%$$

Case 2: Viscosity = 4000 SSU (~ 900 cSt)

Subcase 2.1: Speed 100 rpm. Displacement flow rate: $Q_d = q_0 \times 100 = 0.17 \times 100 = 17.0 \text{ gpm (3.86 m}^3\text{/h)}$ (or read from Figure 15).

Results for $\Delta p = 25 \text{ psi (1.72 bar)}$

Slip flow rate, $S \approx 0.1 \text{ gpm (0.023 m}^3\text{/h)}$ from Figure 16

Net flow rate, $Q_{net} = 17.0 - 0.1 = 16.9 \text{ gpm (3.84 m}^3\text{/h)}$

Total fluid power $P_{tf} = \Delta p \times Q_d/1714 = 25 \times 17.0/1714 = 0.25 \text{ hp (= 0.185 kW)}$ (or read from Figure 17)

Useful output power, $P_{out} = \Delta p \times Q_{net}/1714 = 25 \times 16.9/1714 = 0.247 \text{ hp (0.184 kW)}$

Viscous power, $P_{visc} = 0.18 \text{ hp (0.134 kW)}$ from Figure 18

Shaft power, $P_s = P_{tf} + P_{frict} = 0.25 + 0.18 = 0.43 \text{ hp (0.32 kW)}$

Volumetric efficiency, $E_{vol} = Q_{net}/Q_d = 16.9/17.0 = 99.4\%$

Overall efficiency $E_{tot} = P_{out}/P_s = 0.247/0.43 = 57.4\%$

Results for $\Delta p = 50 \text{ psi (3.45 bar)}$

$$S = 0.4 \text{ gpm (0.091 m}^3\text{/h)}$$

$$Q_{net} = 17.0 - 0.4 = 16.6 \text{ gpm (3.77 m}^3\text{/h)}$$

$$P_{tf} = \Delta p \times Q_d/1714 = 50 \times 17.0/1714 = 0.496 \text{ hp (0.370 kW)}$$

$$P_{out} = \Delta p \times Q_{net}/1714 = 50 \times 16.6/1714 = 0.484 \text{ hp (0.361 kW)}$$

$$P_{visc} = 0.18 \text{ hp (0.134 kW)}$$

$$P_s = P_{tf} + P_{frict} = 0.496 + 0.18 = 0.676 \text{ hp (0.504 kW)}$$

$$E_{vol} = Q_{net}/Q_d = 16.6/17.0 = 97.6\%$$

$$E_{tot} = P_{out}/P_s = 0.484/0.676 = 71.6\%$$

Results for $\Delta p = 100$ psi (6.89 bar)

$$S = 0.75 \text{ gpm (0.170 m}^3\text{/h)}$$

$$Q_{net} = 17.0 - 0.75 = 16.25 \text{ gpm (3.69 m}^3\text{/h)}$$

$$P_{tf} = \Delta p \times Q_d/1714 = 100 \times 17.0/1714 = 0.992 \text{ hp (0.740 kW)}$$

$$P_{out} = \Delta p \times Q_{net}/1714 = 100 \times 16.2/1714 = 0.945 \text{ hp (0.705 kW)}$$

$$P_{visc} = 0.18 \text{ hp (0.134 kW)}$$

$$P_s = P_{tf} + P_{frict} = 0.992 + 0.18 = 1.172 \text{ hp (0.874 kW)}$$

$$E_{vol} = Q_{net}/Q_d = 16.25/17.0 = 95.6\%$$

$$E_{tot} = P_{out}/P_s = 0.945/1.172 = 80.6\%$$

Subcase 2.2: Speed = 550 rpm. Displacement flow rate: $Q_d = q_o \times 550 = 0.17 \times 550 = 93.5$ gpm (21.24 m³/h) (or read from Figure 15).

Results for $\Delta p = 25$ psi (1.72 bar)

$$S \approx 0.1 \text{ gpm (0.023 m}^3\text{/h)}$$

$$Q_{net} = 93.5 - 0.1 = 93.4 \text{ gpm (21.21 m}^3\text{/h)}$$

$$P_{tf} = \Delta p \times Q_d/1714 = 25 \times 93.5/1714 = 1.364 \text{ hp (1.017 kW)}$$

$$P_{out} = \Delta p \times Q_{net}/1714 = 25 \times 93.4/1714 \approx 1.362 \text{ hp (1.016 kW)}$$

$$P_{visc} = 3.9 \text{ hp (2.909 kW)}$$

$$P_s = P_{tf} + P_{frict} = 1.364 + 3.9 = 5.264 \text{ hp (3.927 kW)}$$

$$E_{vol} = Q_{net}/Q_d = 93.4/93.5 = 99.9\%$$

$$E_{tot} = P_{out}/P_s = 1.362/5.264 = 25.9\%$$

Results for $\Delta p = 50$ psi (3.45 bar)

$$S = 0.4 \text{ gpm (0.091 m}^3\text{/h)}$$

$$Q_{net} = 93.5 - 0.4 = 93.1 \text{ gpm (21.14 m}^3\text{/h)}$$

$$P_{tf} = \Delta p \times Q_d/1714 = 50 \times 93.5/1714 = 2.728 \text{ hp (2.035 kW)}$$

$$P_{out} = \Delta p \times Q_{net}/1714 = 50 \times 93.1/1714 = 2.716 \text{ hp (2.026 kW)}$$

$$P_{visc} = 3.9 \text{ hp (2.909 kW)}$$

$$P_s = P_{tf} + P_{frict} = 2.728 + 3.9 = 6.628 \text{ hp (4.944 kW)}$$

$$E_{vol} = Q_{net}/Q_d = 93.1/93.5 = 99.6\%$$

$$E_{tot} = P_{out}/P_s = 2.716/6.628 = 41.0\%$$

Results for $\Delta p = 100$ psi (6.89 bar)

$$S = 0.75 \text{ gpm (0.170 m}^3\text{/h)}$$

$$Q_{net} = 93.5 - 0.75 = 92.8 \text{ gpm (21.08 m}^3\text{/h)}$$

$$P_{tf} = \Delta p \times Q_d/1714 = 100 \times 93.5/1714 = 5.455 \text{ hp (4.069 kW)}$$

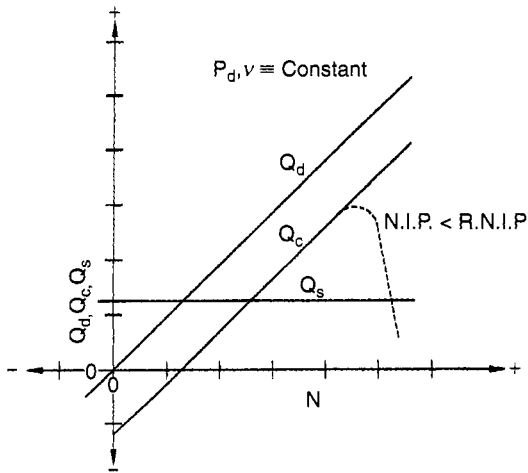
$$P_{out} = \Delta p \times Q_{net}/1714 = 100 \times 92.8/1714 = 5.414 \text{ hp (4.039 kW)}$$

$$P_{visc} = 3.9 \text{ hp (2.909 kW)}$$

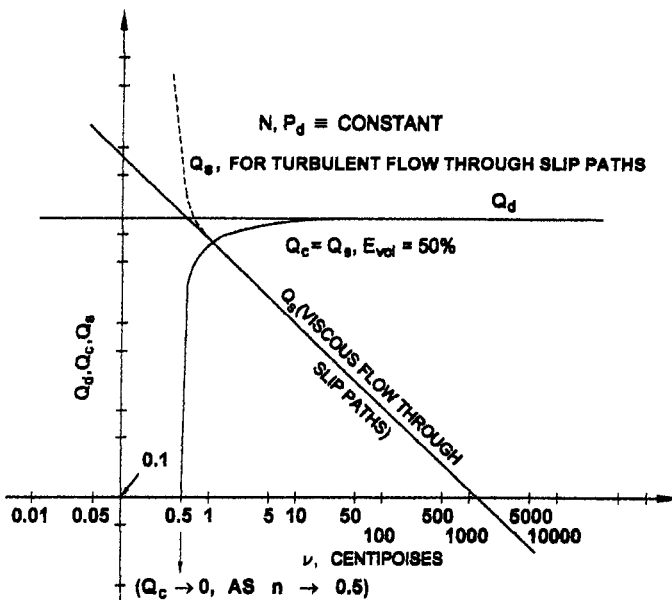
$$P_s = P_{tf} + P_{frict} = 5.455 + 3.9 = 9.355 \text{ hp (6.979 kW)}$$

$$E_{vol} = Q_{net}/Q_d = 92.8/93.5 = 99.3\%$$

$$E_{tot} = P_{out}/P_s = 5.414/9.355 = 57.9\%$$

FIGURE 20 Variations of Q_d , Q_c , and Q_s with N , P_d , and v constant

The graph in Figure 19 is plotted with the flow rate, slip, and displacement flow rate on a linear scale as the ordinate (Y-axis) and pressure on a linear scale as the abscissa (X-axis). A further assumption is that the clearances and viscosity are such that the slip increases proportionately with pressure. The solid lines are the ideal characteristics when secondary effects are neglected. It may be noted that at zero pressure, slip Q_s is zero and Q_c equals Q_d . As the pressure increases, Q_s increases until it equals Q_d at pressure B. If the

FIGURE 21 Variations of Q_d , Q_c , and Q_s with ν , N and P_d constant

pressure imposed on the pump were to increase past this point, Q_s would exceed Q_d and the flow through the pump would be from outlet to inlet, causing a negative Q_c .

Although rotary pumps do not operate in this end of the pressure range, the condition of pressure B can be reached when a valve is closed, blocking the outlet of the pump. Pressure B , then, represents dead-ended pressure developed by a rotary pump when its outlet line is blocked. Should Q_d be numerically equal to 100 in whichever units are used, then the plot of Q_c is numerically equal at each point to the volumetric efficiency E_{vol} of the pump and Q_c becomes zero at pressure B .

Pressure B data is usually not available from pump manufacturers since these values are usually far beyond the normal safe pressure rating of the pump. However, they can be estimated by extrapolating the data given. Cautions should be observed, however. A flexible member can quickly reach a pressure at which flexible member deflection becomes excessive, and the pressure at which Q_c equals zero is very quickly reached. This is illustrated in Figure 19 by the dashed line breaking away from the solid Q line and intersecting the abscissa at pressure point A . This characteristic of flexible member pumps provides a self-limited maximum pressure should the pump be dead-ended. In rigid rotor pumps, as the pressure increases beyond the normal operating range, deflections of the shafts and rotors bring the rotors into heavy bearing contact with the body chamber walls, reducing the dimensions of the slip clearance path. The dashed line leaving the Q_c and Q_s curves and intersecting the abscissa and Q_d line at pressure point C illustrates this. Consequently, the zero flow pressure, which is ideally at point B , may be considerably different from this extrapolated value, and it should be measured if it is important to the application.

Under the same assumptions given for Figure 19, Figure 20 shows the relative independence of the slip with speed when differential pressure is constant. Here the chart of Q intersects the abscissa at the point where the speed is low enough for the displacement flow rate to be equal to the slip at the pressure of operation. The speed at which Q_d equals $2 \times Q_s$ is the speed at which Q_s equals Q_c and the volumetric efficiency E_{vol} becomes 50%.

As the speed increases, the pumping action of the shear stress in the clearances tends to reduce the slip below the ideal line. However, for most rotary pumps, the detrimental effects of increasing the speed above the normal operating range is usually caused by inlet losses in the pump. This effect is cavitation and is illustrated by the dashed line, for which the net inlet pressure (N. I. P.) or NPSH is less than that required.

Figure 21 shows the effect of viscosity on the slip and flow rate in a rotary pump. The graphs are on a log-log scale. In this chart, it is assumed that the pressure, speed, and viscosity combine to keep the flow through clearances of the pump in the laminar or "viscous" flow region.

As viscosity increases, slip Q_s becomes very small and the flow rate of the pump Q_c approaches the displacement flow rate Q_d . As the viscosity decreases, the slip very rapidly approaches the displacement flow rate and the flow rate of the pump drops rapidly to zero or to a negative quantity. For any given pump with speed N and differential pressure P_d , there is a viscosity below which the slip flow, through clearances of the pump, will change to a turbulent flow. It is unlikely that this change will occur simultaneously through all slip paths. However, once it begins to occur, the slip will increase much more rapidly with further reductions in viscosity because of the turbulent flow relationship of the slip, pressure, and viscosity expressed in Eq. 14:

$$Q_s = \frac{KP_d^{1/2}}{v^{1/x}} \quad (14)$$

where x usually is in the range of 4 to 10, and K is a constant.

Volumetric efficiency E_{vol} drops very rapidly with the viscosity if the viscosity is lower than that required for 50% volumetric efficiency (represented by the crossover point of the slip and capacity curves). This crossover point occurs for most rotary pumps operating at rated differential pressure P_d for viscosities between 0.1 and 10 centipoise. For the majority of commercially available models, this point usually falls in the viscosity range of 0.3 to 3.0 centipoise at the maximum rated differential pressure.

The effect of inlet pressure on the capacity can be seen clearly if all secondary effects are eliminated. This assumes the pump is operating within its normal pressure limits and speed

range and that viscosity is high enough to reduce the slip to a negligible value. A graph of the capacity as a function of inlet pressure is shown in Figure 22 with these assumptions.

There is no change in capacity or flow rate as inlet pressure is lowered until the pressure reaches pressure A on the graph. If the inlet pressure were lowered further, the flow rate would drop as shown. The cause of this drop is complex in detail but simple in concept. The liquid flow from the inlet port through the inlet chamber of the pump causes a pressure drop, which causes a minimum pressure point somewhere in the inlet chamber. When the pressure in the liquid at this minimum pressure point approaches the vapor pressure of the liquid, vapor begins to form there. When the amount and time persistence of this vapor cause the vapor to be swept into the CTIO volume of the pump, the amount of liquid in this volume is reduced, leaving a deficiency of liquid volume. For example, if half the fluid volume swept from the inlet chamber is vapor, then only half the normal-capacity liquid volume is available at the outlet chamber, and the flow rate is reduced accordingly.

An increase in speed would mean an increase in flow rate, all other things being equal. This would increase the pressure drop between the inlet port and the inlet chamber and correspondingly increase the absolute inlet pressure (which is measured at the inlet port) at which the flow rate would begin to drop (pressure A). Correspondingly, if the speed and flow rate were constant and the viscosity increased, the pressure drop between the inlet port and the minimum pressure point in the inlet chamber would increase with viscosity. This also would cause pressure A to move to higher inlet absolute pressures. Operation with absolute inlet pressures below pressure A for any given speed and viscosity is usually unsatisfactory both because of the drop in capacity (and hence in volumetric efficiency) and because of the noisy and rough operation caused by the formation and collapse of the vapor.

For lower viscosity liquids where the collapse of the vapor bubbles may be quite rapid, cavitation may cause a significant amount of damage to the body or rotor surfaces. It is important to understand that cavitation damage may occur even though the absolute inlet pressure is above pressure A. Locations may be found in the inlet chamber, particularly where the flow direction changes rapidly, as around sharp corners, where vapor formation in the form of very fine bubbles, and hence cavitation, may occur. However, these vapor cavities may be swept

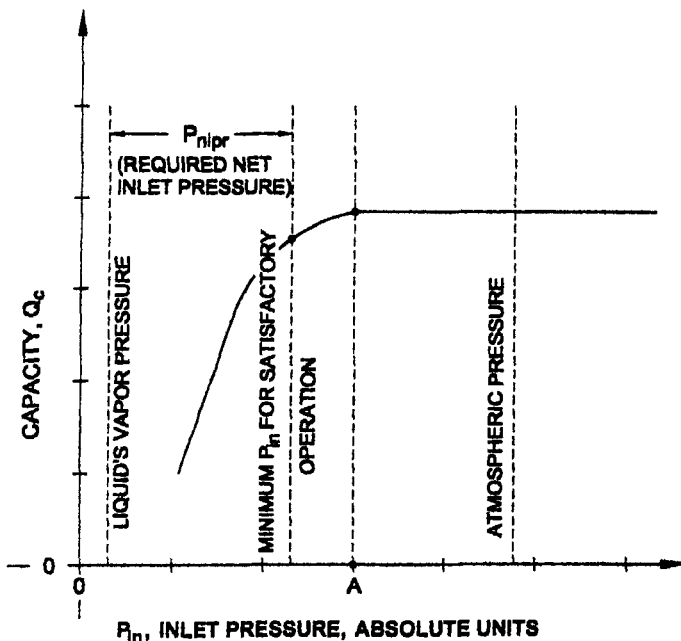


FIGURE 22 Variations of Q_c and P_n with other operating conditions constant

into higher pressure regions of the inlet chamber and collapse near body or rotor surfaces to cause cavitation damage, although the capacity of the pump is not affected. For any given viscosity then, there is an upper limit to the speed at which the pump can be operated.

The inlet pressure may be allowed to drop slightly below pressure A without a significant deterioration in pump performance. However, there is a point at which the pressure becomes too low for satisfactory pump operation. This pressure determines the required net inlet pressure P_{nir} or NPSHR for the particular pump and the particular set of operation conditions. For any given set of operating conditions, satisfying the required net inlet pressure or NPSHR is a main limitation on the pump operating speed.

The other main limit on the pump operating speed is the pump outlet pressure. In every application, some fluid frictional losses occur in the outlet system of the pump. Even if the pump outlet is opened to the atmosphere, a pressure drop exists between some maximum pressure point in the pump outlet chamber and the pump outlet itself. However, by far, the most common situation is one in which a significant fluid friction pressure is developed in the system external to the outlet of the pump. This fluid pressure usually is a function of pump capacity and therefore of pump speed. If the inlet conditions are maintained to keep the inlet pressure above pressure A as the pump speed increases, a speed will be reached at which the pump outlet pressure equals the outlet pressure rating of the pump. Operating at speeds higher than this will cause the pump outlet pressure to exceed the rated pressure and this could result in damage to the pump.

In most applications, one of the two limits described determines the maximum permitted speed of pump operation. However, if neither of these conditions limits the speed and the speed is continuously increased, a speed will be reached at which the peripheral velocity of the rotors will exceed the cavitation velocity of the liquid. A further increase in speed beyond this point would be limited by the cavitation occurring at the rotor outer radial surfaces.

SPECIFICATIONS, INDUSTRIES, AND APPLICATIONS

In addition to the many application-oriented classifications for vane, gear, or lobe pumps, many technical classifications are also issued by various industrial, governmental, or even military organizations. It is also important for both the manufacturer and the user to know of any voluntary or regulatory specifications governing the construction and materials of the pumps destined for use in a specific industry and application.

Many industry-based specifications, such as the International Association of Milk, Food, and Environmental Sanitarians, the United States Public Health Service, and the Dairy Industry Committee, have jointly issued specifications called "The 3-A Sanitary Standards for Pumps for Milk and Milk Products." These standards govern the materials of construction of the pump, the surface finish and shape details of the contact surfaces, the finish and shape of external surfaces, the method of mounting, and restrictions on gaskets, seals, and other pump auxiliary features. Pumps constructed to meet 3-A Sanitary Standards are called *sanitary* pumps and carry a 3-A seal of approval mounted on the pump body. Similar specifications have been developed by the International Association of Milk, Food, and Environmental Sanitarians, the United States Public Health Service, the United States Department of Agriculture, the American Poultry Industries, and the Dairy and Food Industries Supply Association to cover design features used in the handling of cracked eggs.

Standards for pumps used in various processes in the petroleum industry are concerned with the materials and design features that are intended to prevent catastrophic failures when explosive, flammable, or toxic fluids are being handled. Others generated by governmental agencies and professional societies, such as the American Society of Mechanical Engineers and the American Petroleum Institute Standard 676, establish manufacturing procedures and design limitations on pumps for refineries, nuclear power stations, and so on. In these cases, the terms *sanitary*, *aseptic*, *explosion proof*, *API pump* or *N stamp* are not merely generic terms and cannot be casually applied. They must instead be carefully applied and used only when the manufacturer warrants that the pumps actually meet these specifications. In addition, as an unavoidable consequence, the cost of manufacturing pumps to meet any of these standards makes them more expensive than pumps built for general service applications.

TABLE 2 Markets and applications

<u>Animal Renderings Industry</u> General transferring to burners of: <ul style="list-style-type: none"> • animal feed • beef by-products • bone meal • molten fats • poultry by-products 	<u>General Industry and OEMs</u> Various fluid handling requirements for: <ul style="list-style-type: none"> • burners and heater sets • centrifuge separator systems • cooking fat filtration units • filtering and metering units • fluid reclamation systems • hydraulic auto lifts and elevators • lube oil supply units for rotating equipment • machinery coolant and lubricant circulation • paint spray booths • roofing tar systems • textile dyeing and printing machinery • wood preservation machinery 	<u>Pulp and Paper Industry</u> Various general mill and coating applications for: <ul style="list-style-type: none"> • adhesives • black liquor • de-inking solvents • fuel oil • printing ink • sulfate soap • tall oil • titanium dioxide slurry • turpentine • viscose • waste pond recycling • white liquor
<u>Automobile Industry</u> Distribution and assembly line supply of: <ul style="list-style-type: none"> • grease • lubricating oil • motor fuel • paint, lacquers and thinners • test stand systems 	<u>Marine Industry</u> Unloading, tank stripping and engine room supply of: <ul style="list-style-type: none"> • asphalt • heating oil • heavy fuel oil • marine diesel oil • lubricating oil • molasses 	<u>Plastics and Petrochemical Industry</u> Process transfer of: <ul style="list-style-type: none"> • cellophane • epoxy hardener • film dope • insulation coatings • isocyanate • polyester • rayon
<u>Chemical Industry</u> Any process or distribution system handling of: <ul style="list-style-type: none"> • acids • adhesives • bleaches • caustics • cellulose acetate • glycerin • nitrates 	<u>Paint and Lacquer Industry</u> Various process handling applications for: <ul style="list-style-type: none"> • dyes and pigments • lacquer • latex • paint • titanium dioxide slurry • thinners and solvents 	<u>Public Utilities and Power Stations</u> General transfer and supply of: <ul style="list-style-type: none"> • central lubrication systems • fuel oil supply to burners • oil pressure for damper controls • transfer of gashouse tars, creosote, etc.
<u>Cosmetics and Soap Industry</u> Nonemulsifying supply to packaging machinery of: <ul style="list-style-type: none"> • detergent • lanolin • liquefied soap • shampoo 	<u>Petroleum Industry</u> Blending, packaging, and unloading of: <ul style="list-style-type: none"> • asphalt • bitumen • creosote • grease • fuel oil • heavy fuel oil • lubricating oils 	<u>Steel Industry</u> Circulation and supply of: <ul style="list-style-type: none"> • coolants • quench oils • lube oil to rolling mills • hydraulic oil to hydraulic lifters
<u>Food Products Industry</u> General process transferring of: <ul style="list-style-type: none"> • coconut oil • corn oil • fish oils • glucose • glycerin • lard • milk and milk by products • molasses • palm oil • sucrose solutions • vegetable shortening 		

Rotary pumps are used for both metering and transfer applications; although, small diaphragm metering pumps are now more common. Metering pumps tend to be smaller and operate with varying flow rates. Transfer pumps typically have a fixed speed, although variable speed units are becoming more common as the cost for this equipment falls.

In general, the three most basic applications for these pumps are as follows:

1. **Liquid handling** In this class, well over 2,000 fluids are pumped and performance is judged on the pump's capability to handle the specific liquids, and the hydraulic power generated by the pump is a secondary consideration.
2. **Hydraulic fluid power** In this class, the hydraulic power generated by the pump is used to actuate valves, pistons, cylinders, rotary actuators, and similar devices. Pumps in this class are usually special vane, gear, piston, or screw types and are designed for pressures up to 9,000 lb/in² (620 bar) on selected hydraulic power fluids.

- 3. Commercial and retail** In this class are the pumps designed for a single specific use, application, or even fluid. Pumps in this miscellaneous class include everything from miniature pumps for home aquariums to well water pumps for agricultural use to fuel injection pumps for automobiles.

Among these three application areas, the most common is liquid handling tasks, and they make up the largest percentage of uses for rotary vane, gear, and lobe pumps.

Markets and Applications The markets and applications served by vane, gear, and lobe pumps are among the broadest of all rotary pumps. Although it would not be possible to list every application served by these pumps, a broad sampling of the more common ones is shown in Table 2.

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C • H • A • P • T • E • R • 4

SOLIDS PUMPING

SECTION 4.1

HYDRAULIC TRANSPORT OF SOLIDS

KENNETH C. WILSON
ANDERS SELLGREN

INTRODUCTION

Applications of Slurry Transport Vast tonnages are pumped every year in the form of solid-liquid mixtures, known as slurries. The majority of applications are short-distance in-plant installations, mainly in the minerals industry. A large plant may contain several hundred centrifugal pumps handling various types of slurries.

The application that involves the largest quantities is the dredging industry, continually maintaining navigation in harbors and rivers, altering coastlines and winning material for landfill and construction purposes. As a single dredge may be required to maintain a throughput of 7000 tons of slurry per hour or more, very large centrifugal pumps are used with impeller diameters of over 100 in. (2.5 m). (See Wilson et al., 1997, and Section 4.2.)

The manufacture of fertilizer is another process involving massive slurry-transport operations. In Florida phosphate matrix is recovered by huge draglines in open-pit mining operations. It is then slurried, and pumped to the wash plants through pipelines with a typical length of about 6 miles (10 km). The total drive capacity can be in excess of 13,400 hp (10 MW).

Recent decades have seen a great increase in the transport of waste materials, in slurry form, to suitable deposit sites. Waste-disposal environmental problems require that wastes be conveyed to dedicated and monitored disposal sites, either underground or on the surface. This requirement can often be satisfied by backfilling mines (either deep or open-pit), and slurry transport is the favored placing method.

Large slurry pumping operations are found in Alberta, Canada, where oil sands obtained by open-pit mining operations are directly pumped for processing. The mixing and exposure time during the slurry transportation has eliminated some of the processing carried out at the processing plant, resulting in large savings. After the extraction operations are completed, the sand is used as backfill in areas previously mined.

Coarse-particle slurries with maximum particle sizes of 4 in (100 mm) or more can often be pumped cost-effectively by combining them with smaller particles, to produce a broad and even particle size distribution (Sellgren & Addie, 1998).

Slurry transportation may play an important role in the development of integrated mining systems of tomorrow. In mines there is sometimes a considerable inflow of groundwater that has to be removed. When the mine dewatering installations are integrated with a hydraulic hoisting system, the cost of power needed to pump out the groundwater can be excluded from the cost of hoisting in making cost comparisons with other modes of transporting the solids to the surface. The economic effectiveness of hydraulic hoisting, together with hydraulic design considerations, have been discussed by Kostuik (1965) and Sellgren et al. (1989) for both small shallow mines and large deep underground mines.

Long-distance slurry pipeline transport is a thoroughly tested and cost-effective mode of transportation of ores, coal, and industrial minerals. For example, in Brazil up to 13 million tons (12 million tonnes) of iron ore concentrate (0.15 mm) have been pumped per year since 1977 from an inland mine to a pellet plant at the coast 250 miles (400 km) away. The well-known Black Mesa pipeline in southwestern United States transports a partially processed slurry coal (1.5 mm) about the same distance to an electrical generating station.

Long distance slurry pipelines involve heads of 75 atmospheres or more at each pumping section, the pumps employed are therefore usually of the positive-displacement type (see Section 4.3). Descriptions of long-distance pipelines and associated slurry pumps can be found, for example, in Brown & Heywood (1991).

Principles of Slurry Flow Slurries are mixtures of solid particles and a liquid (typically water). The interaction of the solids and the liquid can produce a large variety of flow behaviors. The major types will be outlined here and described in more detail in subsequent sections. The simplest type of slurry behavior occurs, for example, when silt or fine sand is dredged. It is represented by the “equivalent fluid” model. This evaluates the friction loss (pressure gradient) in the pipeline on the basis of an equivalent fluid with the density of the mixture and a friction factor near that of water at the same flow rate. This simple case is not particularly common in practice. On one hand, there are slurries of finer particles, for example natural clays or industrial materials like red mud. Typically, these are non-Newtonian in nature, and require rheological techniques to evaluate the pressure gradient. These will be outlined in the following section on homogeneous slurries.

On the other hand, as particle size or density increases, then settling of particles becomes significant. In turbulent flow the fluctuating velocity of the turbulent eddies (v') acts against the particle settling velocity (terminal fall velocity v_t). The importance of particle settling increases as the ratio v_t/v' becomes larger. When settling is significant the slurry is no longer homogeneous. It is particle-lean near the top of the pipe and particle-rich near the bottom. In the lower area, contact between particles and the bottom of the pipe add a granular (Coulombic) shear stress to that produced by the fluid, thus increasing the pressure gradient (for a given flow rate). The solids contributing to this Coulombic stress are known as “contact load” solids. The stratification ratio, representing the fraction of total solids that travels as contact load, tends to increase with v_t/v' . The turbulent fluctuating velocity v' varies with the mean flow velocity V_m (flow rate divided by $\pi D^2/4$ where D is the internal pipe diameter). Thus, decreasing V_m from an initially large value increases the stratification ratio, and hence the solids effect on the pressure gradient. With further reductions in V_m the liquid effect on pressure gradient decreases, combining with the increasing solids effect to produce a minimum in the pressure gradient (for constant delivered solids concentration). This minimum, shown schematically on Figure 1, was observed long ago by Blatch (1906).

Calculations for this type of pressure-gradient behavior are outlined in the next section on settling slurries, as are other features of this type of flow. It is worth noting here that, as V_m is decreased below the minimum-gradient value, there comes a lesser velocity at which particles at the bottom of the pipe cease moving and form a stationary deposit, also shown on Figure 1. This deposit leads to a further increase in pressure gradient (and hence in the power required) and may cause system instability. Beginning with Durand (1951a, 1951b), deposition has been studied extensively (see the settling slurry section), primarily so that it can be predicted and hence avoided in engineering design.

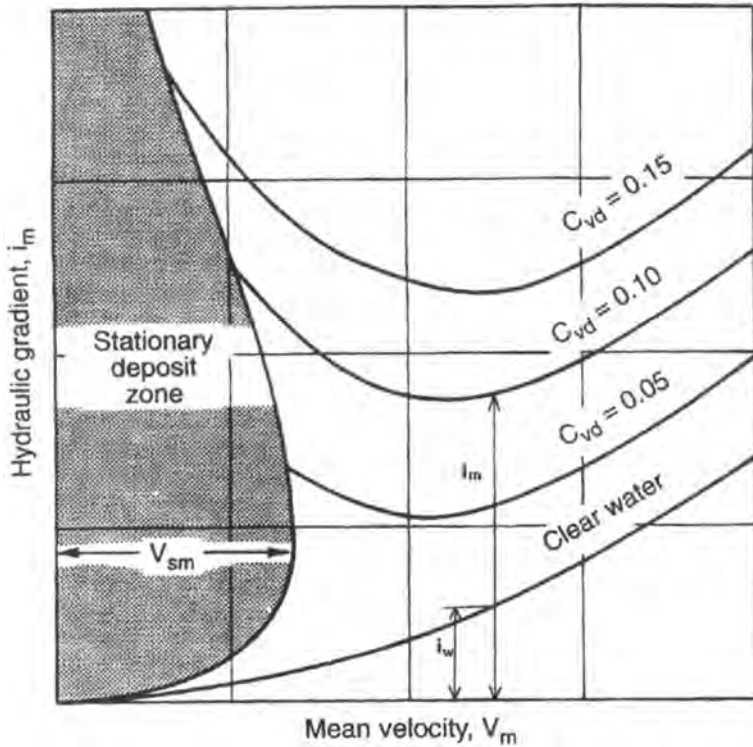


FIGURE 1 Definition sketch

Newitt et al. (1955) divided slurry flows into three types. These include the equivalent-fluid type and the partially stratified type, which have been previously described. The remaining type is known as fully stratified. In this case, either the particle fall velocity is large or the particle size is a significant fraction of the pipe diameter. These factors preclude any support of the solids by the fluid turbulence, so that the submerged weight of all the particles must be carried to the pipe by granular contacts. Calculations (outlined on the section on settling slurries) and experiments show that pressure gradients can be very high for this type of flow. As a result, it is not suitable for long-distance or medium-distance transport; but it has found some applications for short hauls.

HOMOGENEOUS SLURRIES

Equivalent-Fluid Calculations As mentioned in the previous section, pseudo-homogeneous flow (e.g., aqueous slurries of silt or fine sand) shares with truly homogeneous flow the property that the pressure gradient increases with throughput velocity in a fluid-like fashion. An increase of this type can be expressed, at least to a reasonable approximation, by the statement that the pressure drop for turbulent flow of a homogeneous or pseudo-homogeneous mixture is proportional to that obtained for an equal discharge of carrier fluid alone.

Before this statement can be put into mathematical form, it is necessary to consider various ways of expressing the pressure drop due to friction. The change of pressure per

unit length of pipe, $\Delta p/\Delta x$, is not commonly employed in practice, in part because it includes the effect of differences in elevation as well as friction losses. For liquids, especially water, the hydraulic gradient, i , is normally used. This quantity is the slope of the hydraulic grade line, which is based on the levels to which the fluid would rise in a series of imaginary tubes tapped into the pipeline. As the slope of the hydraulic grade line, the hydraulic gradient represents the drop in level per unit length of pipe. For water (density ρ_w) flowing in a horizontal pipe, i is related to the pressure gradient as follows

$$i = \left(- \frac{\Delta p}{\Delta x} \right) / \rho_w g \quad (1)$$

where g is gravitational acceleration.

As water forms the carrier liquid in the majority of slurry flows, it is convenient to use it as a standard for comparison of frictional pressure losses. Thus, throughout this chapter the usual expression for frictional losses is in terms of i (height of water per unit length of pipe). The value for i for clear-water flow is written i_w , and that for a mixture (i.e., frictional loss in height of water per unit length) is written i_m . The statement made previously for homogeneous and pseudo-homogeneous flows amounts to a direct proportionality between i_m and i_w .

Calculation of i_w uses the Stanton-Moody friction factor, f , in the equation

$$i_w = f \frac{V^2}{2gD} \quad (2)$$

For turbulent flow, the friction factor depends on both the Reynolds number ($Re = \rho V_m D / \mu$) and the relative roughness ϵ/D . These two quantities can be entered on Figure 2, which is

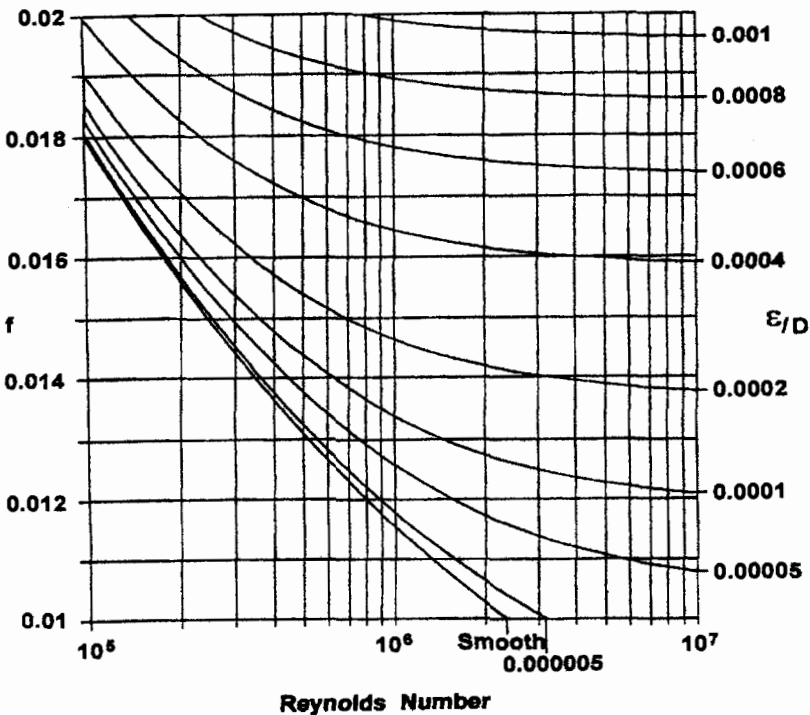


FIGURE 2 Friction factor in normal operating range

used to obtain f . For example, with $Re = 1.5 \times 10^6$ and $\epsilon/D = 0.0001$, $f = 0.013$ (a typical value). As f is approximately constant for turbulent flow in a given pipe, Eq. 2 shows that the hydraulic gradient varies roughly as V^2 (or as Q^2).

In nonslurry applications, if a liquid other than water is conveyed, the hydraulic gradient concept is applied by substituting the density of this liquid for that of water in Eq. 1, so that the hydraulic gradient is expressed in height of flowing liquid per unit length. In considering, say, a pseudo-homogeneous slurry of fine sand, it may be of interest to compare its behavior to that of a liquid with density equal to that of the flowing mixture ρ_m , which is given by

$$\rho_m = \rho_w(1 + (S_s - 1)C_v) \quad (3)$$

Here S_s is the relative density of the solids (compared to water) and C_v is the volumetric concentration of solids.

Substituting ρ_m for ρ_w in Eq. 1 gives the mixture-height gradient (i.e., measured in height of mixture rather than of water). For clarity, a different symbol must be used, and j is employed for this purpose. For a mixture flowing in a horizontal pipe:

$$j = \left(-\frac{\Delta p}{\Delta x} \right) / \rho_m g \quad (4)$$

Although j is expressed in height of mixture per unit length of pipe, actual measurements based on columns of mixture in vertical tubes would not be feasible, and j would have to be obtained indirectly from pressure measurements. It can be seen from Eqs. 1 and 4 that the ratio of i to j equals the relative density of the mixture S_m (i.e., ρ_m/ρ_w).

The equivalent-fluid model of slurry flow assumes that the solids have little effect on friction factor, and that the mixture acts as a liquid as far as the relative-density effect is concerned. The resulting hydraulic gradient for homogeneous mixture flow, i_{mh} , is equivalent to the product of S_m and i_w . Although the equivalent-fluid model has been widely employed in the past, it is not generally supported by the experimental evidence. For example, sand-water experiments by Carstens & Addie (1981) show that for some pseudo-homogeneous flows i_{mh} does not exceed i_w at all. An appropriate equation for the hydraulic gradient is:

$$i_{mh} = [1 + A'(S_m - 1)]i_w \quad (5)$$

Setting the coefficient A' equal to unity gives the relative-density effect of an equivalent-fluid model, whereas $A' = 0$ gives the behavior observed by Carstens & Addie (1981). Intermediate types can be represented by values of A' between zero and unity. Equation 5 will be referred to as the "homogeneous flow" equation, with the specific case of $A' = 1.0$ called the "equivalent fluid" model.

Modelling Non-Newtonian Flows In pipeline transport of non-Newtonian materials, the variation of pressure drop with velocity is typically rather flat for laminar flow, while it is much steeper for turbulent flow. Figure 3 shows these features. This figure is a plot, on logarithmic axes, of hydraulic gradient j_m (i.e., frictional losses in height of equivalent fluid per unit length of pipe) versus mean velocity V_m (volumetric discharge/cross-sectional area). The data shown on this plot refer to various concentrations of a red mud tested at the GIW Hydraulic Laboratory in a pipeline with internal diameter of 3.19 in. (81 mm). The plotted data are for mixtures with relative densities from 1.14 to 1.28. Note that where laminar flow occurs the slope of the lines (referred to the logarithmic coordinates) is rather small and does not vary significantly with particle concentration. However, the vertical position of the laminar lines increases strongly with increasing concentration. Conversely, for turbulent flow, concentration has very little effect; all the points fall close to a single line with slope (for logarithmic coordinates) close to 2.0. This behavior implies that the friction factor f has a near-constant value for the observed turbulent flows. Note that the transitions between laminar and turbulent flow are rather abrupt.

Rheograms, that is, curves of shear stress versus strain rate, are obtained from tests under laminar conditions, using either rotary viscometers or tube viscometers. The latter

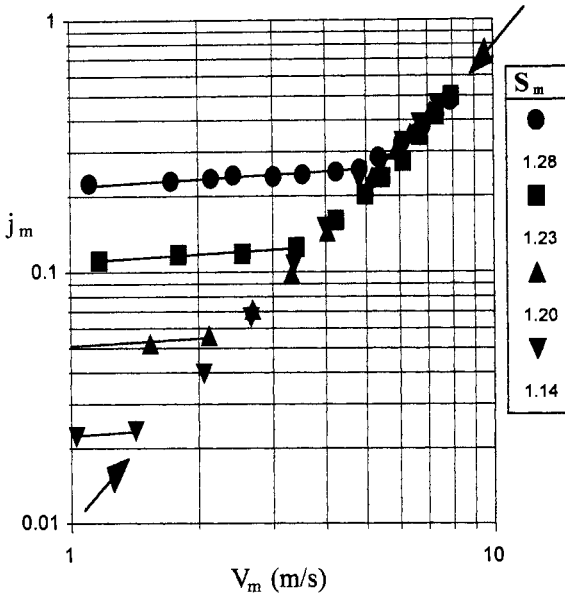


FIGURE 3 Friction gradients of a red-mud slurry

are preferable, as they are geometrically similar to the pipeline configuration. For a Newtonian fluid of viscosity μ , shear stress τ is related to strain rate du/dy by the classic expression

$$\tau = \mu(du/dy) \quad (6)$$

Rheograms for non-Newtonian materials are not so simple. For example, Figure 4 shows a rheogram for a fine-particle slurry. This rheogram does not pass through the origin (the strain rate remains zero until a certain yield stress τ_y is exceeded) and is not straight (although in this case the behavior is approximately linear at large values of strain rate). The definition of viscosity given by Eq. 6 can be retained, although this viscosity no longer represents the slope of the rheogram itself, but rather that of the secant line shown on the figure. Hence μ as defined by Eq. 6 can also be called the “secant” viscosity. The tangent to the rheogram, though less meaningful physically, is often referred to. This “tangent” viscosity will be denoted η_t . Both μ and η_t vary with position, and hence they both depend on du/dy (or, alternatively, on τ). The stippling of Figure 4 indicates the area beneath the leftward portion of the rheogram. The area ratio α (which will be used below) is the ratio between the stippled area and the triangular area below the secant line. As with μ and η_t , both the area beneath the rheogram and the area ratio α depend on du/dy (or on τ).

It is common practice to represent rheograms by simple mathematical functions (i.e., functions having only two or three parameters). For materials exhibiting a yield stress, the most common engineering choice is the Bingham model, a straight line given by

$$\tau = \tau_B + \eta_B(du/dy) \quad (7)$$

Here the Bingham viscosity η_B is the tangent viscosity given by the slope of the fit line, and the Bingham yield stress τ_B is the intercept of the fit line with the shear-stress axis. For the data of Figure 4, τ_B is larger than the observed stress at zero strain rate, a situation that is commonly encountered.

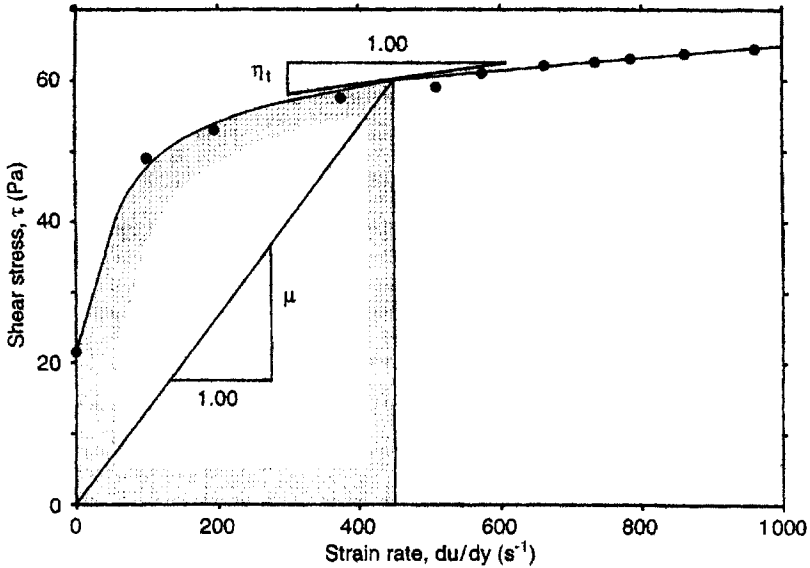


FIGURE 4 Rheogram for phosphate slimes tested at GIW Hydraulic Laboratory (after Wilson, 1986).
[Pa $\times$ 0.000145 = lb/in²]

The dimensionless ratio τ/τ_B (denoted θ and called the stress ratio) is a useful parameter when dealing with Bingham physics. For example, in laminar flow, the secant viscosity ("apparent viscosity") μ is related to the stress ratio by the equation

$$\mu/\eta_B = \theta/(\theta - 1) \quad (8)$$

This equation indicates a gradual decrease of μ with increasing stress ratio. Another parameter appropriate for Bingham plastics is the Hedström number, He

$$He = D^2 \rho \tau_B / \eta_B^2 \quad (9)$$

where ρ is the mixture density.

The friction factor for laminar flow of a Bingham plastic can be expressed in terms of these parameters as

$$\sqrt{\frac{8}{f}} = \sqrt{He} \cdot \left(1 - \frac{4\theta^{-1}}{3} + \frac{\theta^{-4}}{3}\right) \cdot \frac{\sqrt{\theta}}{8} \quad (10)$$

As noted previously, turbulent flows of non-Newtonians are often characterized by a roughly constant value of friction factor, in contrast with the large variations typical of laminar flow. An old way of estimating f for turbulent flow, proposed by Hedström (1952), is to use the Bingham viscosity η_B in place of μ in the relationship for Newtonian turbulent flow. This approach should be considered as only a rough approximation. A better method is to begin with, say, the smooth-wall Newtonian equation

$$V_m = 2.5 U_* \ell n(\rho D U_* / \mu) \quad (11)$$

where ℓn denotes the natural logarithm and the shear velocity U_* , that is, $\sqrt{\tau/\rho}$, is evaluated at the pipe wall. For turbulent pipe-flow tests of a non-Newtonian material, all quantities except μ will be available for each test run. Therefore, in each case Eq. 11 can be

solved for the viscosity, which is now denoted μ_{eq} , the equivalent turbulent-flow viscosity. This viscosity is not constant (as would be the case for a Newtonian fluid, or for Hedström's postulate, i.e., $\mu_{eq} = \eta_B$) but varies with wall shear stress, and hence with shear velocity, as described by Wilson et al. (1997).

If turbulent-flow tests are available, scale-up to prototype size can be accomplished directly (as shown in the following section). Otherwise, μ_{eq} can be predicted on the basis of the laminar-flow rheogram. The prediction method follows the model of Wilson & Thomas (1985), which in turn is based on a proposal of Lumley (1973, 1978) regarding the viscous sublayer. Increasing the size of the dissipative micro-eddies leads to an increase in the thickness of this sublayer. The thickened sublayer, in turn, produces a higher mean velocity for the same wall shear stress. The model uses the ratio of the integrals under the non-Newtonian and Newtonian rheograms (denoted by α , and defined earlier in connection with Figure 4) to estimate the size increase of the turbulent micro-eddies. It is found that both the micro-eddy size and the thickness of the viscous sublayer should be multiplied by a factor equal to α . This results in a simple expression for the equivalent turbulent-flow viscosity. For a Bingham plastic α can be expressed in terms of the stress ratio as $(\theta + 1)/\theta$, and the expression for μ_{eq} becomes

$$\mu_{eq}/\eta_B = [(\theta + 1)/(\theta - 1)]\exp(-4.64/\theta) \quad (12)$$

A plot of this function (Wilson, 1997) shows that, as θ increases above 1.0, μ_{eq}/η_B begins by dropping below unity, descends to a minimum (0.216 at $\theta = 1.32$) and then rises gradually toward unity. The resulting expression for turbulent-flow friction factor depends only on the parameters He and θ . The equivalent expression for laminar flow (Eq. 10) involves the same two parameters. The intercept of turbulent and laminar lines occurs where these two equations give the same value of f . At this intercept it is possible to eliminate θ using an iterative technique, thus producing a plot of f (at intercept) versus Hedström number. On logarithmic coordinates the plot is not far from a straight line in the region $200 \leq \sqrt{He} \leq 10000$, and a power-law approximation can be used in this range, as follows:

$$f \approx 0.0877(He)^{-0.113} \quad (13)$$

Experimental evidence from pipe-flow tests of fine-particle slurries typically shows abrupt laminar-turbulent transitions, followed in the turbulent range by a plateau of virtually constant friction factor. Equation 13 gives a valuable tool for predicting f for this plateau. This friction factor falls below the traditional curve of Hedström, by which the equivalent turbulent-flow viscosity is equated to the Bingham tangent viscosity, η_B .

This technique can be extended to predict the transition between laminar and turbulent flow. For a Newtonian fluid, with a single rheological parameter (μ), the transition is defined by a specific value of a single dimensionless ratio (the Reynolds number $\rho V_m D/\mu$). A second rheological parameter, required for a non-Newtonian material such as a Bingham plastic, introduces the need for a second dimensionless ratio. In this case the two ratios are the "Bingham" Reynolds number, that is, $Re_B = \rho V_m D/\eta_B$ and Hedström number which has been previously defined. As pointed out by D. G. Thomas (1963) the laminar-turbulent transition must now be defined by a functional relationship linking Re_B at transition with He .

Following the method used in obtaining Eq. 13, the laminar and turbulent relations can be equated at the intercept and f (as well as θ) can be eliminated to obtain a plot of Re_B at intercept versus He . It was found that the result could be closely approximated by the simple relationship that Re_B equals $25\sqrt{He}$. On cancelling common quantities, it is seen that the flow velocity at transition, say V_T , is given by:

$$V_T \cong 25\sqrt{\tau_B/\rho} \quad (14)$$

The approach of D. G. Thomas (1963) produced a similar form in the limit, but with a coefficient of 19 rather than 25.

Scale-up of Laminar and Turbulent Flows The laminar flow of any given fluid can be scaled from one pipe size to another by use of the Rabinowitsch-Mooney technique provided that the flow remains laminar in both pipes (turbulent flow will be dealt with separately below). The basis of the Rabinowitsch-Mooney technique is the plot of wall shear stress τ_0 versus $8V_m/D$. Rabinowitsch (1929) and Mooney (1931) proved that for steady uniform laminar flows in a pipe τ_0 and $8V_m/D$ are linked in a functional relationship. In other words, for a given material the values of both τ_0 and $8V_m/D$ can be determined from experiments in a single pipe. The experimentally determined plot or function linking these two variables can then be applied to all laminar flows of the material in question, whatever the pipe diameter.

The application of the method is based on a logarithmic plot of τ_0 versus $8V_m/D$ obtained from a pipe of internal diameter D_1 . For some other pipe, of diameter D_2 , the pressure gradient dp/dx can be obtained from the plotted values of τ_0 by multiplying them by $(4/D_2)$. Since the factors by which each of the coordinates are multiplied depend only on D_2 , a simple rescaling of the axes of the figure based on D_1 gives a plot of dp/dx versus V_m for a pipe of diameter D_2 . For plots on logarithmic coordinates, this type of scaling amounts to a pair of linear translations, which do not affect the shape of the plot. In this regard it should be noted that a Newtonian fluid shows a slope of unity on a logarithmic plot of pressure gradient versus mean velocity, whereas the slopes of logarithmic plots for non-Newtonian laminar flow (such as those on Figure 3) are much flatter.

An alternative way of dealing with scale-up to a larger pipe is to transform the data from the experimental pipe (diameter D_1) to the prototype (diameter D_2) on a point-by-point basis. This process is similar to using the affinity laws for scaling head and discharge data for a centrifugal pump. For the pipe, the analogous quantities are the frictional gradient i_m (ft of water/ft or m of water/m) and the mean velocity V_m . With the subscripts 1 and 2 denoting experimental and prototype conditions, respectively, the scaling relations (affinity laws) for laminar flows in pipes are:

$$(i_m)_2 = (i_m)_1 \left(\frac{D_1}{D_2} \right) \quad (15)$$

and

$$(V_m)_2 = (V_m)_1 \left(\frac{D_2}{D_1} \right) \quad (16)$$

If it is foreseen that flow in the prototype pipeline may be turbulent, it is important to extend the small-scale experimental tests into the turbulent-flow region, and then to scale these test results up to prototype size. For turbulent scaling, as for the laminar case, the quantity that remains unchanged is the wall shear stress τ_0 . As in laminar flow, the wall shear stress fully determines the stress distribution within the pipe. For example, with a material having a yield stress τ_y , a specified value of τ_0 is sufficient to give the fraction of the pipe area where $\tau < \tau_y$, a factor which can affect the mean velocity (Wilson & Thomas, 1985; Thomas & Wilson, 1987). Likewise, a given value of τ_0 , and hence of U_* , determines conditions within the viscous sublayer and hence establishes both μ and the area ratio α . The result is that τ_0 (or U_*) is sufficient to determine the equivalent turbulent-flow viscosity. For each value of τ_0 two scaling laws apply to turbulent pipe flow. The first, for pressure gradient, is the same as that established for laminar flow, that is, Eq. 15. The second scaling law, for mean velocity, is given by

$$(V_m)_2 = (V_m)_1 + 2.5U_*\ell n(D_2/D_1) \quad (17)$$

As with Eqs. 15 and 16 for laminar flow, the combined use of Eqs. 15 and 17 allow scale-up of data for turbulent non-Newtonian flow from one pipe size to another. Wilson (1986) gives an example where scaled-up values from a small pipe are compared to measured data in a larger pipe, with very satisfactory agreement. Equation 17 is also valid for cases where μ is constant, that is, for Newtonian fluids, and for turbulent flow of dilute polymer solutions exhibiting drag reduction (Wilson, 1989).

TABLE 1 Data for phosphate slimes slurry in 8 in. (203 mm) pipe

Run	V_m		$8V_m/D(s^{-1})$	i_m	τ_0		U_*	
	ft/s	m/s			lb/ft ²	Pa	ft/s	m/s
1	1.7	0.53	21.0	0.1004	1.04	49.9	0.69	0.210
2	5.0	1.52	60.1	0.1130	1.17	56.2	0.73	0.223
3	6.6	2.00	78.8	0.1150	1.19	57.2	0.74	0.225
4	8.5	2.59	102.1	0.1189	1.24	59.1	0.75	0.229
5	10.6	3.24	127.9	0.1218	1.27	60.1	0.76	0.231
6	12.5	3.81	150.3	0.1237	1.29	61.5	0.77	0.233
7	14.5	4.43	174.7	0.1273	1.32	63.4	0.78	0.237
8	16.8	5.12	202.0	0.1348	1.40	67.0	0.80	0.243
9	18.5	5.64	222.6	0.1472	1.53	73.2	0.83	0.254

EXAMPLE 1 A clay-water slurry of phosphate slimes is to be pumped over a horizontal distance of 2300 ft. (700 m), using a pipe of internal diameter 12 in. (305 mm). The slurry will be taken from a pond in which the relative density of the mixture is 1.13. Tests have been carried out with this material using a pipe of internal diameter 8 in. (203 mm). Test data for i_m and V_m appear in Table 1 together with values of $8V_m/D$ and τ_0 (the wall shear stress equals $\rho_m g i_m D/4$). Figure 5 shows the values of i_m plotted versus V_m for the test pipe.

The preliminary design of the pumping system calls for a discharge Q_m of 4770 gpm (0.30 m³/s) in the prototype pipe. However, this value is not yet definite, and Q_m values of 3180 gpm (0.20 m³/s) and 6360 gpm (0.40 m³/s) are also to be considered.

- a. Find the values of i_m for the three values of Q_m just noted. First, the pipe area (0.785 ft² or 0.0731 m²) is used to obtain the required velocities, that is, 9.0, 13.5 and 18.0 ft/s (2.74, 4.11 and 5.48 m/s).

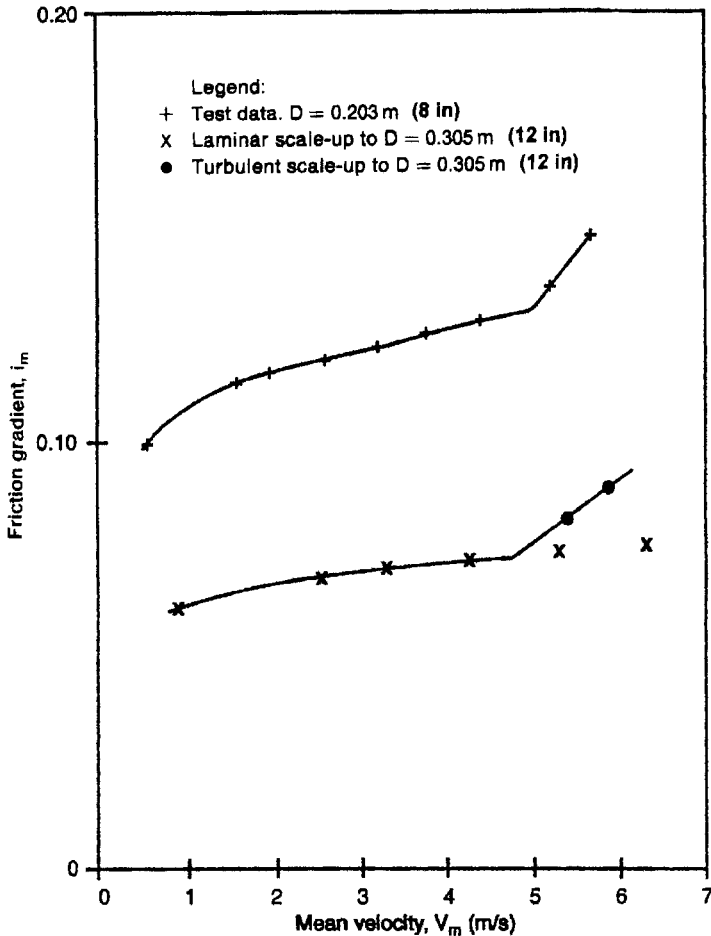
The next step is to scale the data points from the test pipe to the prototype pipe. For the laminar-flow points the appropriate scaling laws are given by Eqs. 15 and 16, which show that i_m scales inversely with the diameter ratio while V_m scales directly with this ratio. Points from test runs 1 to 7 have been scaled on this basis and the first six are shown on Figure 5. For test runs 8 and 9 the flow is turbulent. The hydraulic gradient can still be scaled by Eq. 15, but V_m must now be scaled by Eq. 17. This requires evaluation of U_* , that is, $\sqrt{\tau_0/\rho}$. For example, run 8 with $\tau_0 = 0.01$ lb/ft² (67.0 Pa) has $U_* = 0.8$ ft/s (0.243 m/s), for which Eq. 17 gives a scaled-up value of V_m equal to 17.6 ft/s (5.37 m/s).

The laminar-turbulent transition point is obtained by projecting the turbulent line back to intercept the laminar line. For the data in the test pipe this intercept occurred to the right of point 7, at a velocity of about 16 ft/s (say 5 m/s). For the prototype pipe the intercept lies between points 4 and 5, with a slightly lower velocity. Note that the scaled values for points 5 to 7 are not physically meaningful, because the equivalent flows in the prototype pipe are turbulent, not laminar.

Figure 5 shows that conditions will be laminar for the two lower flows to be investigated, and turbulent for the highest flow. The values of i_m can be taken directly from the figure, and are listed in Table 2.

The flat curve for laminar flow, plotted on Figure 5, shows a very small rise in i_m as Q_m goes from 7.1 to 10.6 ft³/s (0.20 to 0.30 m³/s). On the other hand, the equal increase of Q_m from 10.6 to 14.1 ft³/s (0.30 to 0.40 m³/s) requires a much larger increment in i_m , as a result of the shift from laminar to turbulent flow.

- b. Suppose now that testing had stopped after run 6, so that no turbulent-data flow points were available.

FIGURE 5 Test data and scale-up for Example 1 [$\text{m/s} \times 3.28 = \text{ft/s}$]

In this case the laminar line would be scaled as before, but the transition to turbulent flow must be predicted from the parameters of the laminar-flow rheogram. It was found that all laminar points except the lowest one can be considered as obeying a Bingham formulation of the rheogram, with $\tau_y = 0.0076 \text{ lb/in}^2$ (52.7 Pa) and $\eta_B = 0.020 \text{ Pa} \cdot \text{s}$. Substitution of these quantities into Eq. 14 yields $V_T = 17.7 \text{ ft/s}$

TABLE 2 Hydraulic gradients and pump heads from scaled-up values

	0.20	0.30	0.40
Q_m (m^3/s)			
Q_m (US gpm)	3180	4760	6360
V_m (m/s)	2.74	4.11	5.48
V_m (ft/s)	9.0	13.5	18.0
i_m	0.0760	0.0795	0.0915
Pump Head (m water)	53.2	55.6	64.0
Pump Head (ft water)	174	183	210

(5.4 m/s) which is somewhat above the observed transition. The similar expression of D. G. Thomas (1963) gives V_T of 13.4 ft/s (4.1 m/s), significantly below the observed transition point. The use of Eq. 13 to evaluate turbulent friction produces a turbulent line that passes through the transition point predicted by Eq. 14.

SETTLING SLURRIES

Velocity at Limit of Stationary Deposition The deposition limit will be discussed in this section, and the modelling of fully stratified coarse-particle flow at velocities above the deposition limit in the following section.

The results of a detailed force-balance computer model for the limit of stationary deposition showed that the throughput velocity V_m at this limit is concentration-dependent, having small values at low concentration, rising to a maximum (denoted V_{sm}) at some intermediate concentration (which depends on pipe size, particle size, and density) and then dropping off again as the delivered solids concentration approaches the loose-poured value, C_{lv} . This behavior was shown on Figure 1. It should be noted that the velocities used here are obtained simply by dividing the mixture flow rate (Q_m) by the pipe area ($\pi D^2/4$).

The computer output is unwieldy for a designer concerned with many alternative proposals. Moreover, the conservative designer may be content to know only the maximum velocity at the limit of deposition, V_{sm} , since maintaining the operating velocity above this value ensures that deposition will not occur. The value of V_{sm} depends on internal pipe diameter, particle diameter and relative density, and the effect of these variables is expressed concisely by a nomographic chart which was developed at Queen's University (Wilson & Judge, 1978; Wilson, 1979) with the help of the late Professor F. M. Wood's expertise in nomography (Wood, 1935). This chart, reproduced here as Figure 6, is recommended as a practical design aid.

It should be noted, by way of explanation of the chart, that the left-hand panel deals with sand-weight materials ($S_r = 2.65$). The internal pipe diameter appears on the left vertical axis, with V_{sm} on the central vertical axis. The particle diameter is plotted on a curve known, on the basis of its shape, as the "demi McDonald." This shape illustrates that for large particles, but not for small ones, the shear stress at the interface between the upper and lower layers increases with increasing particle size. Thus, for coarse-particle transport the velocity at the limit of deposition (i.e., the velocity beyond which no bed can remain stationary) will decrease with increasing particle size. This finding, which is amply supported by experimental evidence, shows how the fully stratified mode of transport can display behavior quite different from that of the heterogeneous mode.

To demonstrate the particle size effect consider particles of $S_r = 2.65$ in a pipe 12 inches (0.30 m) in diameter. This diameter is located on the vertical scale on the left-hand side of the chart, and connected by straight-edge to any desired particle size on the curved scale. V_{sm} is then obtained by projected to the central vertical scale. For instance, a particle size of 0.025 in. (0.6 mm) gives V_{sm} of almost 13 ft/s (4 m/s), which is the largest value found for this pipe diameter and solids density. For a larger particle of, say 0.2 in. (5 mm), the deposition-limit velocity is diminished to about 9 ft/s (2.7 m/s).

When operating with centrifugal pumps it may be difficult to take advantage of the decrease of V_{sm} with increasing particle size. The question is one of obtaining a stable intercept of pump and pipeline characteristics, and will be discussed later in this section. For applications where control of particle size is limited, the conservative designer may wish simply to assume particles of "Murphian" size, i.e. those which give the largest value of V_{sm} for the pipe under consideration. In this case, it should be noted that the values of V_{sm} obtained from Figure 6 tend to be conservatively high, especially for large pipe diameters. Thus, these values can sometimes be used as operating velocities.

A particularly useful feature of nomographic presentation of results is that it gives an immediate indication of the sensitivity of the output to variations in the input. Thus on Figure 6 it is seen that the value of V_{sm} for sandweight solids in a 12-in. (0.30-m) pipe is

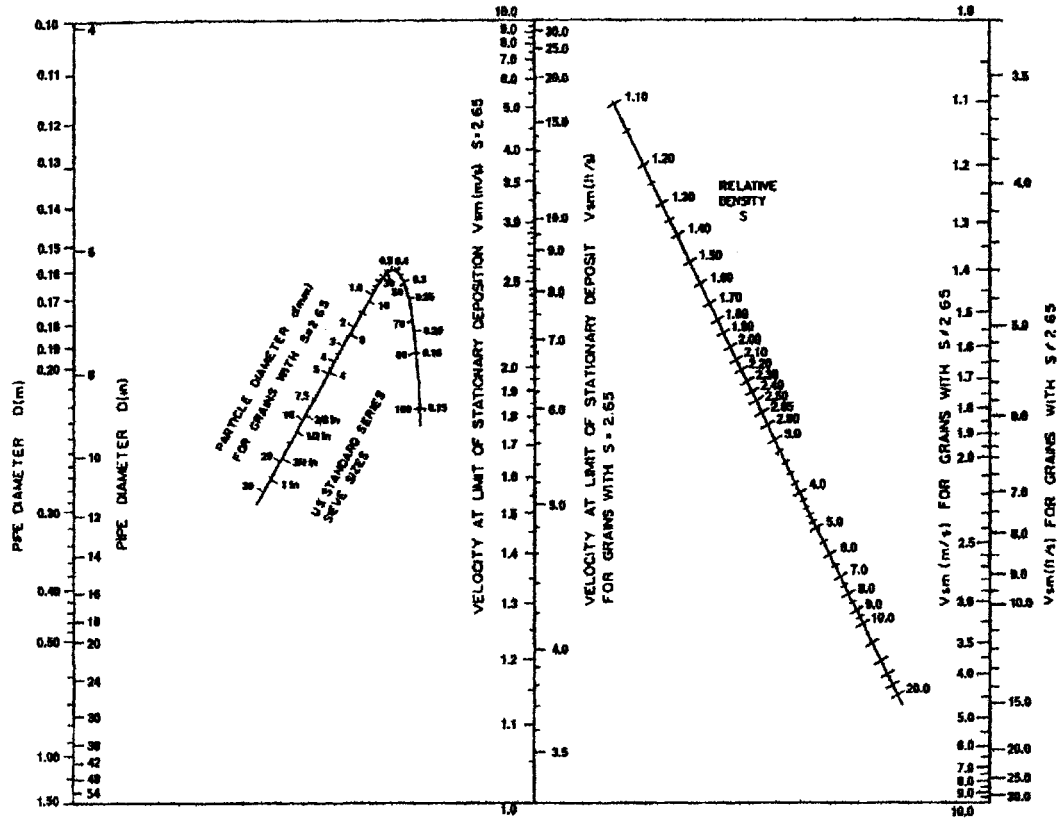


FIGURE 6 Nomographic chart for maximum velocity at limit of stationary deposition (from Wilson, 1979).

virtually unaffected by a variation of the particle diameter between 0.015 and 0.04 in. (0.4 and 1.0 mm). However, the right limb of the particle-diameter curve shows greatly increased sensitivity, so that a change in d from 0.006 to 0.008 in. (0.15 to 0.20 mm) alters V_{sm} by more than 25%.

It is found that a change of the relative density does not influence the form of the relation shown on Figure 6, but does require its "recalibration." This is accomplished graphically by using the inclined relative-density axis on the right-hand side of the figure. To illustrate the use of the right-hand panel of Figure 6, consider again the pipe 12 inches (0.30 m) in diameter, now carrying particles with a diameter of 0.004 in. (1.0 mm) and relative density 1.50. With 0.004 in. (1.0 mm) entered on the particle scale, a straight line joining it to the pipe diameter can be projected to the central axis of the figure, showing that V_{sm} is 11.5 ft/s (3.5 m/s) for the equivalent sand size. In order to correct this for the relative density of 1.50, the latter value is entered on the sloping axis on the right-hand side of the figure and joined by straight-edge to the point just found on the central axis (11.5 ft/s or 3.5 m/s). The projection of this line to the vertical axis on the right of the figure gives the adjusted value of the maximum deposition velocity, approximately 6 ft/s (1.9 m/s).

Particles that differ in density from sands may also have different values of other properties, including the solids fraction in the deposit and the mechanical friction coefficient between the particles and the pipe. The values of these quantities that were employed in the computer program, and hence are reflected in the nomographic chart, apply to sands but not necessarily to other materials. Therefore the values of V_{sm} determined from Figure 6 for materials other than sands must be treated as somewhat less accurate than values for sand. However, both the solids fraction in the bed and the particle-pipe mechanical friction coefficient occur in the computer program only as multiples of the submerged relative density of the particles, and it is found that any change in these quantities merely gives rise to a multiplicative factor which must be applied to the values of V_{sm} obtained from the figure. This permits calibration of the output of the nomographic chart from a few pilot-plant tests with the material of interest.

The particle diameter scale of Figure 6 has not been extended below 0.006 in. (0.15 mm) since smaller particles tend to be influenced by mechanisms not included in the mathematical model on which the nomogram is based. As shown by Thomas (1979) the viscous sublayer can have a significant effect, and all particles which are small enough to be completely embedded in this sublayer will behave in a fashion which is no longer dependent on particle diameter. In this limiting case certain simplifications can be made in the mathematical model. Thomas found that these lead to a simple expression for the shear velocity at the limit of stationary deposition, and on employing a power law approximation for the friction factor, he then obtained a corresponding expression for V_{sm} .

$$V_{sm} = 9.0[g\nu(S_s - S_f)]^{0.37}(D/\nu)^{0.11} \quad (18)$$

Here ν represents the kinematic viscosity of the carrier fluid (i.e., μ/ρ), and the coefficient of 9.0 applies in any consistent system of units. This equation gives the minimum value of deposition velocity for small particles, assuming turbulent flow.

For cases where the interfacial friction sets up a sheared layer several grain diameters in thickness, a new analysis has been developed (Wilson, 1988; Wilson & Nnadi, 1990). For large pipes, and particles near the Murphian size, this new analysis shows that the particle size no longer influences V_{sm} directly. In such cases the value of V_{sm} is less than that predicted by Figure 6, a point which is in accord with experimental evidence. It is then necessary to obtain the value of V_{sm} from the new analysis of Interfacial friction, for comparison with the value from the nomographic chart.

When the parameters in the new analysis are given values typical for sands, it is found that the dimensionless maximum deposition velocity (i.e., the Durand deposition variable) depends only on the fluid friction factor for the portion of the pipe wall above the deposit. A power-law approximation gives an appropriate fit function for this effect, that is,

$$\frac{(V_{sm})_{max}}{\sqrt{2gD(S_s - S_f)}} = \left(\frac{0.018}{f_f}\right)^{0.13} \quad (19)$$

where f_f is the friction factor for fluid alone. If the value of V_{sm} found from the nomographic chart exceeds the value of $(V_{sm})_{max}$ from Eq. 19 the latter value should be used for V_{sm} .

Fully Stratified Coarse-Particle Transport Fully stratified flow occurs where almost all of the particles travel as contact load (i.e., fluid suspension is ineffective). The ratio of particle diameter to pipe diameter is of major importance in determining the presence of this flow type, which does not normally occur for d/D ratios less than 0.015. Fully stratified flow is less likely if the particles are broadly graded, especially if there is a significant homogeneous fraction (i.e., a significant fraction of particles smaller than 200 mesh/75 μm). Calculations made for narrow-graded slurries with water as a carrier fluid indicate fully stratified behavior for values of d/D above 0.018.

Although the actual relationship of the detailed force-balance analysis cannot be expressed in closed form, simple approximating functions can be fitted to match the output of the detailed model. Wilson & Addie (1995) proposed the following expression for approximating fully stratified flow:

$$\frac{i_m - i_w}{(S_m - 1)} = B' \left(\frac{V_m}{0.55V_{sm}} \right)^{-0.25} \quad (20)$$

The ratio on the left-hand side of this equation is the same as that used by Newitt et al. (1955), and the right-hand side shows the decrease of this ratio with increasing V_m/V_{sm} . The coefficient B' has a value of 1.0 for angular particles, and decreases with the degree of rounding. For typical cases the suggested default value is 0.75. As fully stratified flow has higher energy consumption than other flow types, Eq. 20 can be thought of as representing the upper limit of excess pressure gradient (i.e., solids effect). Heterogeneous slurry flows, which are covered in the following section, will generally display smaller values of the solids effect.

Another type of stratified flow that may be encountered is flow above a stationary bed of solids. This type of operation is generally uneconomic, and thus is seldom a deliberate design choice, but it is sometimes found in existing pipelines. Flow over a stationary bed has been studied experimentally, and analyzed by a computer model (Nnadi & Wilson, 1992; Pugh, 1995). For most cases of this type of flow encountered in pipelines the following rough approximation to the computer output may be sufficient. It is based on particles near the "Murphian" size, which tend to be disproportionately represented in stationary deposits:

$$i_m = 0.32(S_s - 1)^{1.05} C_{vd}^{0.6} \left(\frac{V_m}{\sqrt{2gD}} \right)^{-0.1} \quad (21)$$

Pressure Gradients for Partially Stratified Flow As mentioned previously, two methods of support are significant in partially stratified flows: turbulent suspension and granular contact. For turbulent suspension the eddies can carry the particles with no significant additional energy consumption, but granular contacts set up Coulombic friction forces which must be overcome by an additional "solids effect" pressure gradient, represented by $(i_m - i_f)$ where i_f (or i_w) is the hydraulic gradient for fluid only. In many cases this solids effect varies with the submerged weight of solids, i.e., with $(S_s - S_f)C_v$ where S_f is the relative density of the fluid (1.0 for water) and C_v is the volumetric delivered concentration. The quantity $(S_s - S_f)C_v$ can also be expressed as $(S_m - S_f)$ where S_m is the relative density of the mixture. The relative solids effect, written $(i_m - i_f)/(S_m - S_f)$, can be obtained from pipe-loop testing. Typical results for partially stratified flow are shown on Figure 7 for various concentrations of sand with d_{50} of 0.016 in. (0.42 mm). The relative solids effect drops with increasing mean velocity V_m , which in turn produces increased turbulent fluctuating velocities, implying that a larger fraction of particles will be suspended by turbulence. The straight fit line shown on the logarithmic coordinates of Figure 7 indicates that the relation can be approximated by a power law. (Other formulations have also been proposed, see for example Shook & Roco, 1992 and Gillies et al., 1991.)

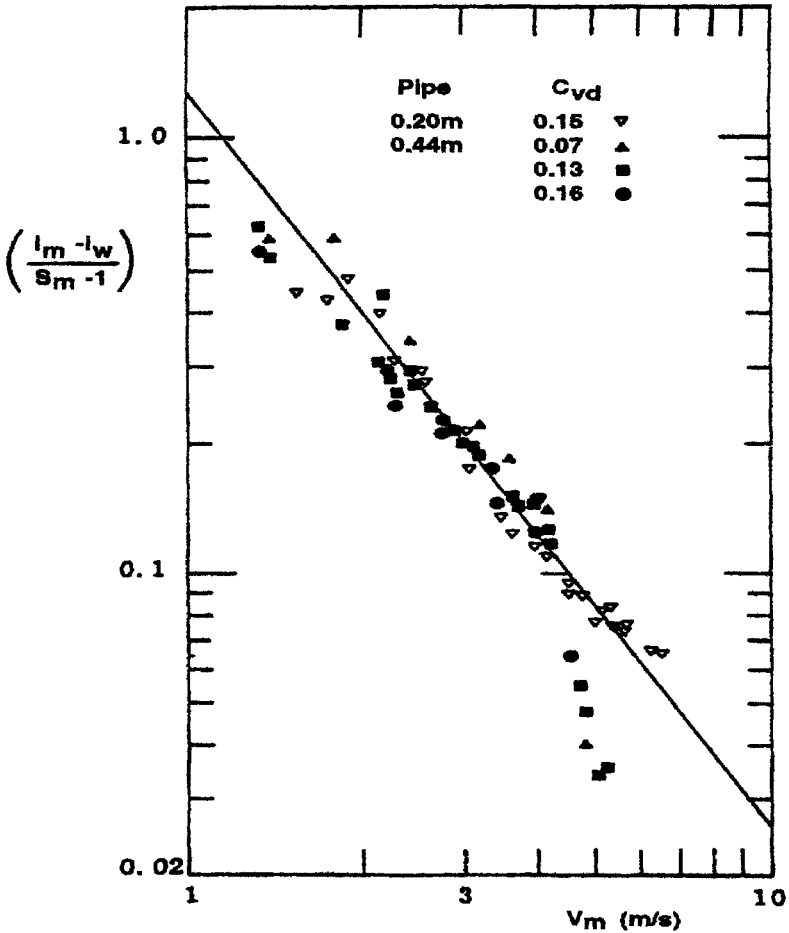


FIGURE 7 Behavior of masonry-sand slurry ($d_{50} = 0.42$ mm) in pipes of two sizes (after Clift et al., 1983).
[m $\times$ 3.28 = ft; m/s $\times$ 3.28 = ft/s]

In addition to the slope parameter (denoted by M) a power law requires a central-value parameter, denoted here by V_{50} . Conceptually, this represents the value of V_m at which half the mass of solids is supported by granular contact and half by fluid suspension. From a practical standpoint it is necessary to match this point with a specific value of $(i_m - i_f)/(S_m - S_f)$. The mechanics of the situation were discussed by Wilson et al. (1997) who proposed a value of 0.22 and showed that this is supported by experimental data. The resulting equation is

$$\frac{(i_m - i_f)}{(S_m - S_f)} \cong 0.22 \left(\frac{V_m}{V_{50}} \right)^{-M} \quad (22)$$

If test data are available, M and V_{50} can be obtained directly from the results. For example consider the data of Figure 7 in the 8 in. (0.20 m) pipe. In calculating the ratio $(i_m - i_w)/(S_m - 1)$, the clear water gradient i_w has been used to approximate i_f , and S_f is S_w ,

that is, 1.0. The value of i_w was calculated from the Darcy-Weisbach formula using the friction factor $f_f = 0.013$ measured previously for flows of water in this pipe. The plotted points fall on an essentially straight line on Figure 7. This behavior corresponds to Eq. 22. The slope of the line gives $M \approx 1.7$, which is typical for slurries with narrow particle grading. The point on the line where $(i_m - i_w)/(S_m - 1) = 0.22$ gives V_{50} , which is approximately 9 ft/s (2.8 m/s).

With V_{50} and M obtained for the 8-in. (0.20-m) pipe, scale-up to a larger pipe diameter can be carried out. The larger diameter of 18 in. (0.44 m) has been selected because data are available with the same sand in this larger pipe (Clift et al., 1982) and thus the scaled-up results can be verified directly. In fact, as seen on Figure 7, the fit line for the larger pipe coincides with that for the smaller pipe in this instance. In this case the clear-water friction factor f_w was found to be the same for both pipes and in both pipes the particle size d was a small fraction of the pipe diameter. It has been found that V_{50} should vary with $(8/f_w)^{1/2}$; it should also depend on the diameter ratio d/D (Wilson & Watt, 1974; Wilson et al., 1997).

These factors can be incorporated in the scale-up procedure, which involves preparing curves of i_m versus V_m for various values of C_{vd} in the larger pipe, the relation for each line of constant C_{vd} being expressed as

$$i_m = \frac{f_w}{2gD} V_m^2 + 0.22(S_s - 1)V_{50}^M C_{vd} V_m^{-M} \quad (23)$$

It is typical to find a moderate decrease in f_w with increasing pipe size, which produces a small increase in V_{50} . Increases in D have the opposite effect, but for small ratios of d/D , as is the case for the data of Figure 7, this effect is not significant.

Other problems arise when pipeline experiments have not been carried out with the particular slurry of interest. In many cases of practical importance, information on the size and grading of the material to be pumped is limited, but estimates of the solids effect must be made. For example, consider the case of an ore that is to be crushed and then transported by pipeline. At the initial stage of the design there may be no adequate sample of the crushed ore, but estimates of the effect of the solids on the head loss must be made for feasibility studies, preliminary designs and cost estimates, and even for justifying the expenses of laboratory or pilot-plant testing.

To estimate the solids effect, two parameters are required: the power M and the velocity V_{50} . The value of M has a lower limit of 0.25 (for fully stratified flow) and approaches 1.7 for slurries with narrow particle grading. If only a rough idea of the grading is available, it may be adequate to use the following approximation, which requires only an estimate of the particle diameter ratio d_{85}/d_{50} (d_{50} is the mass-median particle diameter and d_{85} is the diameter for which 85% by mass of the particles are smaller). Using this evaluation, the approximation for M is written

$$M \approx [\ell n(d_{85}/d_{50})]^{-1} \quad (24)$$

Here ℓn is the natural logarithm. Also, M should not be allowed to exceed 1.7 or fall below 0.25.

The next step is to obtain a commensurate approximation for V_{50} (Wilson et al., 1997). This formula reads

$$V_{50} \approx 3.93 d_{50}^{0.35} [(S_s - 1)/1.65]^{0.45} \quad (25)$$

Here d_{50} is in mm. The coefficient 3.93 applies for velocities in m/s; for velocities in ft/s this coefficient becomes 12.9. With sand-weight solids $(S_s - 1)$ equals 1.65 and the bracketed portion of Eq. 25 equals 1.00. The value of V_{50} obtained from Eq. 25 is substituted into Eq. 22 to obtain the solids effect $(i_m - i_f)$, where i_f is the gradient for an equal flow of water. Note, Eq. 25 is only applicable for $0.006 \text{ in} < d_{50} < 0.055 \text{ in}$ ($0.15 \text{ mm} < d_{50} < 1.4 \text{ mm}$). For larger particles the value of V_{50} given by Eq. 25 should be multiplied by $\cosh(60d_{50}/D)$.

Remarks on Complex Slurry Flows In the introduction to this section, it was mentioned that slurry flows can be divided into three types on the basis of mechanisms of particle support. These types are homogeneous, partially stratified and fully stratified, and

the applicable methods of analysis for these flows have been outlined in the appropriate preceding sections. Quite often, the particle grading curve is sufficiently broad to span two of the flow types, or even all three. This gives rise to complex slurry flows. The larger particles, which would settle readily in water, often receive considerable support from the smaller particles and the carrier fluid, promoting efficient transport. A complete analysis of such flows is not yet available, but it is hoped that the following remarks will aid the design engineer.

Whenever some coarse particles settle, they form contact load. As shown in earlier sections of this chapter, this has an effect on pressure drop which is quite different from that of particles suspended by the fluid. The contact-load effect, analyzed previously for the case of a Newtonian carrier fluid, must eventually be combined with the scaling laws for non-Newtonian fluids presented in an earlier part of this chapter. As laminar flows which have significant particle settling are usually avoided in design, only turbulent flows will be considered here.

Maciejewski et al. (1993) compared large-diameter transportation of coarse particles of about 4 in. (100 mm) in clay suspensions and in oil-sand tailings slurries (particle size below 0.03 in./0.8 mm). They found that the sand slurry was more effective as a transport medium than a viscous, homogeneous clay slurry. The important role of particles with sizes of 0.004 to 0.020 in. (0.1 to 0.5 mm) in reducing friction was further shown in studies by Sundqvist et al. (1996a, 1996b) for products with d_{50} of 0.024 to 0.027 in. (0.6 to 0.7 mm) and various size distributions, with maximum sizes of up to 6 in. (150 mm).

In studying the behavior of complex slurries like these, it is logical to begin by dividing the total concentration of solids C_v into three components, each associated with a support mechanism. Thus C_h stands for homogeneous, C_{mi} for partly stratified (the "middlings") and C_{cl} for the coarse fully stratified particles (the "clunkers"). On the basis outlined previously the particle size of 200 mesh ($75 \mu\text{m}$) separates C_h and C_{mi} and the size $0.018D$ separates C_{mi} and C_{cl} . This point is best illustrated by an example. Take $S_s = 2.65$; and a concentration of 30% by volume ($C_v = 0.30$). From the solids grading curve, suppose that 20% of the total is slimes, 50% middlings and 30% clunkers. Thus, the concentration of slimes in the slurry is $(0.30)(0.20) = 0.06$, and similarly 0.15 and 0.09 for middlings and clunkers, respectively. The equivalent fluid based on the slimes has specific gravity $S_h = 1 + (S_s - 1)C_h = 1 + (1.65)(0.06) = 1.099$ and that for the combined slimes and middlings is $S_{hmi} = 1 + (1.65)(0.21) = 1.347$. Thus, for the middlings the specific gravity difference between solids and carrier fluid is $(2.650 - 1.099) = 1.551$ (rather than 1.650). For the clunkers, the equivalent difference is $(2.650 - 1.347) = 1.303$.

The homogeneous fraction now forms the carrier fluid for the rest of the slurry, and its hydraulic gradient i_h replaces i_w in equations like Eq. 20 and Eq. 22. These are used to determine the solids effect for middlings and clunkers, which may be written Δi_{mi} and Δi_{cl} . The gradient for the mixture i_m represents the sum of i_h and the solids effects for the middlings and the clunkers, i.e.

$$i_m = i_h + \Delta i_{mi} + \Delta i_{cl} \quad (26)$$

The homogeneous gradient i_h is based on appropriate equivalent-fluid or non-Newtonian calculations, as given previously. For the middlings, Δi_{mi} is effectively equivalent to $(i_m - i_p)$ in Eq. 22, when applied to the middlings only, giving

$$\Delta i_{mi} = C_{mi}(S_s - S_h)0.22 \left(\frac{V_m}{V_{50}} \right)^{-M} \quad (27)$$

Here, S_h is the relative density of the homogeneous "carrier fluid" component (1.099 for the example just introduced). The evaluation of M and V_{50} will be mentioned in the following text.

For the clunkers, Δi_{cl} is based on Eq. 20, except that the carrier fluid for the clunkers now includes both the homogeneous portion and the middlings, with a relative density written S_{hmi} (1.347 for the example). The carrier fluid will also have an effect on the coefficient which will now be written B'' instead of B' (the evaluation of B'' will be mentioned next). The relation for Δi_{cl} is

$$\Delta i_{cl} = C_{cl}(S_s - S_{hmi})B'' \left(\frac{V_m}{0.55V_{sm}} \right)^{-0.25} \quad (28)$$

For coarse solids only, it was suggested above that the parameter B' in Eq. 20 should have a default value of 0.75. When considerable fractions of middlings are found, B' must be reduced to give B'' . As it is expected that the reduction is associated with density differences, the appropriate variable would appear to be the weight concentration, C_w , of fines plus middlings, i.e., C_{whmi} (e.g., $C_{whmi} = 0.372$). Thus

$$B'' = B'(1 - \xi C_{whmi}) \quad (29)$$

where ξ is a coefficient with a proposed default value of unity.

Another point concerns the evaluation of V_{50} and M for the middlings. The methods of Eqs. 24 and 25 are basically applicable but d_{50} and d_{85} must now refer to the middlings fraction only, rather than the whole grading curve. For very broad well-distributed gradings the middlings will simply be a slice between 0.003 in. (0.075 mm) and 0.018 D . Here it may be possible to assume that d_{50} is the geometric mean of the two diameter limits (e.g., 0.025 in. for $D = 1$ ft or 0.64 mm for $D = 0.305$ m). For this case a default value of about 1.0 might be appropriate for M . V_{50} will be influenced by the rheological properties of the homogeneous fraction, which will influence the fall velocity of the middling particles, thus enhancing suspension. An initial approach is to consider the homogeneous part of the slurry to act as a Newtonian fluid with effective viscosity significantly higher than that of water. The effect of enhanced viscosity on V_{50} is of the form shown on Figure 8. On this figure the value of V_{50} based on water viscosity is entered on the abscissa (corrected for relative density of solids if different from 2.65). The various curves are for different μ_r , the ratio of viscosity to that of water at 70°F (20°C). The multiplier read from the ordinate can be applied to the V_{50} value for water to estimate V_{50} in the higher-viscosity carrier fluid.

EXAMPLE 2 This example pertains to fully stratified flow of coarse magnetite particles ($d = 1$ in. or 25 mm). Because of their high relative density ($S_s = 4.4$) these particles are to be used as ballast for an offshore drilling rig. The magnetite will be transferred from ore carriers to the rig by a dredge pipeline with internal diameter 19.7 in. (0.500 m).

- a. Find the deposition limit in this line, V_{sm} . Figure 6 is entered with the pipe diameter (on the left axis) and the particle diameter (on the demi McDonald).

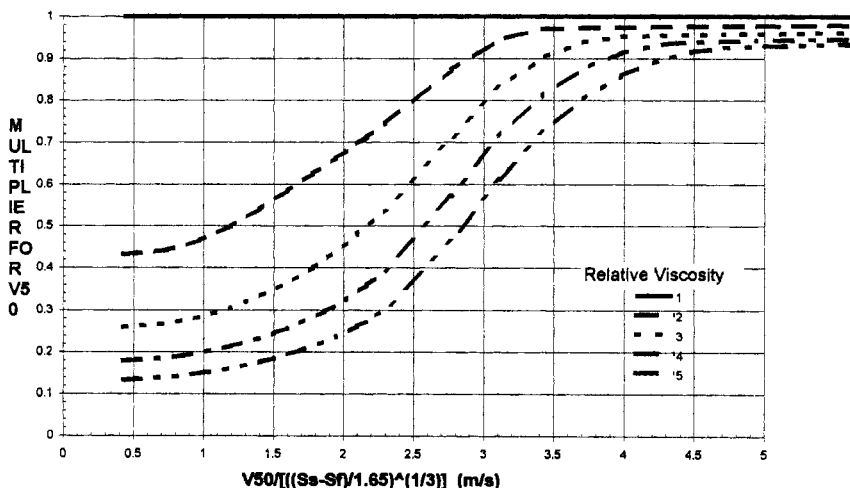


FIGURE 8 Variation of V_{50} with viscosity

Projection to the central axis gives $V_{sm} = 8.7$ ft/s (2.65 m/s) for sand-weight material. This number is joined to the relative density S (i.e., S_s) of 4.4 on the sloping axis and projected to the right-hand axis to give $V_{sm} = 13$ ft/s (4.0 m/s) for the magnetite.

- b. Find the hydraulic gradient i_m for a volumetric solids concentration of 0.10 and a throughput velocity $V_m = 16.4$ ft/s (5.0 m/s). Assume rather angular material with $B' = 0.9$. The first step is to calculate the clear-water friction gradient of i_w . Using $f_w = 0.013$, i_w is given by $0.013V_m^2/2gD$ or 0.033 (height of water per length of pipe). To this must be added $(i_m - i_w)$, which Eq. 20 gives as $(S_m - 1)B'(V_m/0.55V_{sm})^{-0.25}$. The quantity $(S_m - 1)$ equals $C_v(S_s - 1)$ i.e., $(0.10)(3.4) = 0.34$. B' has been taken as 0.9 and the ratio $(V_m/0.55V_{sm})$ equals 2.27, giving $(i_m - i_w) = 0.250$ and $i_m = 0.28$, i.e., 28 ft of water per 100 ft of line or 0.28 m of water per m of line. Very large gradients like this, which have been verified in prototype-scale testing, show why fully stratified flow is only of interest for short-haul applications.

EXAMPLE 3

- a. Suppose that the magnetite of the previous example has been ground to give $d_{50} = 0.008$ in. (0.20 mm) and $d_{85} = 0.012$ in. (0.30 mm). As in the previous example, $D = 1.64$ ft (0.50 m) and $V_m = 16.4$ ft/s (5.0 m/s). In this case the volumetric concentration $C_v = 0.20$. Find the V_{sm} and the hydraulic gradient i_m .

V_{sm} is found as in the previous example. The pipe size is joined to the particle size of 0.008 in. (0.20 mm) and projected to the central axis to give the deposition velocity for sand-weight solids. Solids specific gravity is entered on the sloping line, and projected to the right-hand axis to give $V_{sm} = 14.4$ ft/s (4.4 m/s). As in the previous example this is less than the proposed operating velocity, which is satisfactory.

The flow of this slurry will be partially stratified, and from Eq. 24, $M = [\ell n(0.30/0.20)]^{-1} = 2.5$. However, the maximum limit of M is 1.70, which will be used here. From Eq. 25,

$$V_{50} = 3.93(0.20)^{0.35} \left(\frac{3.40}{1.65} \right)^{0.45} = 3.1 \text{ m/s (10 ft/s)}$$

For this value of V_{50} , with $M = 1.7$ and $V_m = 16.4$ ft/s (5.0 m/s), the right hand side of Eq. 22 becomes 0.098. $(S_m - 1)$ equals $(S_s - 1)(C_v) = (3.40)(0.2) = 0.68$. Hence

$i_m = i_w + (0.68)(0.098)$, and with i_w of 0.033 (found in Example 2) i_m is found to be 0.100 (ft water per ft or m water/m). Note that this gradient is only about one-third that for the coarse particles of the previous example, despite the fact that the solids concentration (and the tonnage transported) is twice as much.

- b. As in part (a) but with $d_{85} = 0.016$ in. (0.40 mm). The power M is reevaluated as $M = [\ell n(0.40/0.20)]^{-1} = 1.44$. For the unchanged values of V_{50} and V_m , the right-hand side of Eq. 22 becomes 0.110 and i_m equals $0.033 + 0.68(0.110)$ or 0.108 (ft water/ft or m water/m).
- c. As in part (a) but assume that there is a fine-particle fraction that increases the viscosity of the homogeneous component to 4 times that of water. From part (a), $M = 1.70$ and $V_{50} = 10$ ft/s (3.1 m/s) in water. Adjusting for the density difference from the sand-water case gives the abscissa of Figure 8 as $3.1/(3.40/1.65)^{0.33}$ i.e., 7.88 ft/s (2.4 m/s). With this abscissa and a relative viscosity of 4, Figure 8 gives a multiple of 0.42, for a revised estimate of V_{50} as $(0.42)(3.1) = 4.3$ ft/s (1.3 m/s). Thus the right-hand side of Eq. 22 becomes 0.022, and $i_m = .033 + 0.015$ or 0.048 (ft water/ft or m water/m). This revised estimate is only about one-half the value of 0.100 obtained in part (a) for pure water as the carrier fluid.

OTHER FACTORS

Vertical Flows For industrial application of vertical transportation of a solid-liquid mixture in a pipe, the operating velocity must be sufficient to maintain a continuous flow of solids at the discharge end. However, unnecessarily high velocity causes excessive pipe wear and energy losses. The appropriate operating velocity depends on the settling conditions of the solids, indicating that size, density, and concentration of particles are key parameters in the hydraulic design of a vertical particle-fluid transportation system.

Many experimental studies have been made of vertical slurry transport. For example, Sellgren (1979) used a pilot-scale facility with a centrifugal pump to investigate important design parameters for ores and industrial minerals taken from in-plant crushing and milling. The results of these experiments are summarized in the following paragraphs.

It is suggested that the allowable minimum mixture velocity be based on the settling velocity of the largest particles in still water multiplied by a factor of 4 or 5. Provided the velocity exceeds this value, then in most industrial applications, with volumetric concentrations of 15% to 30%, the corresponding pressure requirement in the vertical system can be determined by the equivalent-fluid model. As noted in connection with Eq. 5, this model is based on the density of the slurry and the friction factor for water. Applied to vertical flow, it gives

$$p = \rho_w g S_m z \left(1 + \frac{f_w V_m^2}{2gD} \right) \quad (\text{for } V_m > V_{\text{all}}) \quad (30)$$

Here p is the pressure required, S_m is the relative density of the slurry, z is the length of vertical pipe, and V_{all} is the allowable minimum mixture velocity, approximately four times the settling velocity of the largest particle.

The settling velocity in still water of industrially crushed mineral particles is normally reduced significantly compared to smooth spheres of corresponding size. Tests have shown that, on average the settling velocity for particles in the range of 0.04 in. to 1.2 in. (1 mm to 30 mm) is reduced approximately 50%. Therefore, the criterion previously given for the minimum allowable velocity could alternatively be formulated as: V_{all} is twice the settling velocity of a smooth sphere of the same size as the largest particles.

Within the constraints previously discussed, systems operate under conditions where the effect of relative velocity between the components appears to be negligible. The maximum particle sizes considered are in the range of 0.04 in. to 1.2 in. (1 mm to 30 mm) in pipe diameters of 4 to 12 in. (0.1 m to 0.3 m). With larger particle sizes (up to 4 to 6 in./100 to 150 mm) and low concentrations, the relative velocity between the components becomes significant. Boundary-layer transitional effects may also introduce certain instabilities that must be carefully evaluated in long vertical risers. Particles larger than one-fifth the pipe diameter can promote slugging instability in vertical hoisting, and particles larger than one-third the pipe diameter may jam the pipe and should be avoided.

EXAMPLE 4 Centrifugal slurry pumps are used to pump a sand slurry ($d_{50} = 0.06$ in./1.5 mm) out of a quarry. The pipe is vertical with a length of 328 ft. (100 m) and a diameter of 4 in. (0.10 m). Tests have shown that the settling velocity of the largest particles is approximately 1.5 ft/s (0.45 m/s). Select the operating velocity and calculate the head requirement in meters of slurry.

Solution Following the guidelines previously given, the velocity V_{all} is four times 0.45, i.e., 6 ft/s (1.8 m/s). At this velocity, $V_m^2/2g$ is 0.54 ft (0.165 m). With the friction factor f_w taken as 0.016 for smooth-pipe conditions (see Figure 2), the head is obtained from Eq. 30 as

$$328 \left[1 + \frac{0.016(0.541)}{3.94/12} \right] = 336.5 \text{ ft of slurry}$$

in USCS units or 336.5 S_m in ft of water.

In SI units the calculation becomes

$$100 \left[1 + \frac{0.016(0.165)}{0.1} \right] = 102.6 \text{ m of slurry}$$

or $102.6 S_m$ in meters of water.

Inclined Flows Lengths of pipe with an adverse slope often form part of pipelines transporting solids. Compared to the horizontal case, flow up an incline tends to require higher throughput velocities in order to avoid deposition. This is of greatest significance for coarse-particle flow.

In an experimental investigation carried out by Wilson & Tse (1984), four particle sizes between 0.04 and 0.24 in. (1 and 6 mm) were tested in a pipe at angles of inclination up to 40° from the horizontal. It was found that the velocity at the limit of deposition initially increases with the angle of upward inclination, θ , reaching a maximum when this angle is about 30° . For the materials tested this maximum velocity was approximately 50% larger than that required to move a deposit in a horizontal pipe. This large difference is clearly a matter of importance for both design and operation of pipelines with inclined sections.

As indicated schematically on Figure 1, V_{sm} marks the lower end of the range of desirable operating velocities for a pipeline. For purposes of comparison, it is appropriate to represent the deposition limit for inclined flow in terms of the dimensionless velocity, or Durand number, $V_{sm}/[2g(S_s - 1)D]^{1/2}$. The difference between the Durand number for inclined flow and that for horizontal flow, ΔD , is plotted against θ on Figure 9.

For the effect of pipe inclination on friction loss, the following widely used formula by Worster & Denny (1955) may be employed for heterogeneous flows and homogeneous flows in which the hold-up effects are small. Their approach, based on water as the carrier fluid, deals with the extra pressure gradient ($\Delta i(\theta)$, expressed in height of water per length of pipe) beyond that for pumping water alone. For horizontal flow, this extra gradient is simply the solids effect ($i_m - i_w$), which may be written $\Delta i(0)$. Worster and Denny's formula states that

$$\Delta i(\theta) = \Delta i(0) \cos \theta + (S_s - 1) C_{vd} \sin \theta \quad (31)$$

For highly stratified flows, Eq. 31 underestimates the losses. For further information see, for example, Wilson et al. (1997).

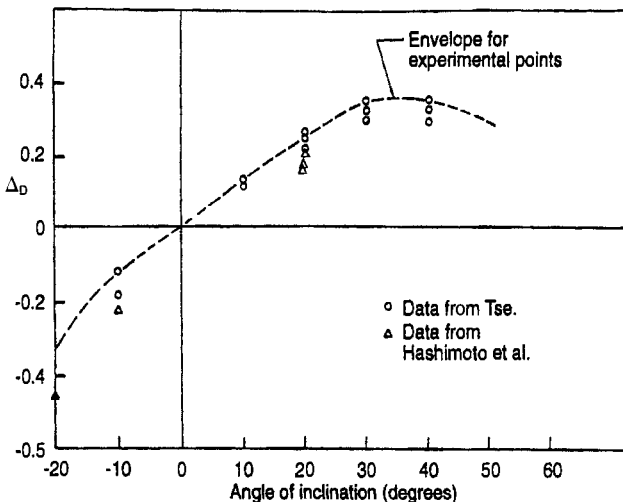


FIGURE 9 Effect of angle of inclination on Durand deposition parameter (after Wilson and Tse, 1984)

EXAMPLE 5 Consider an inclined suction pipe located on a dredge ladder. Pipe length = 59 ft (18 m), $D = 25.6$ in. (0.65 m), angle to the horizontal = 30° . Determine the deposition velocity and the head losses at the proposed operating velocity of 21.3 ft/s (6.5 m/s). Assume $S_s = 2.65$ and $C_v = 0.20$. For a horizontal pipe, it is calculated that $V_{sm} = 15.7$ ft/s (4.8 m/s) and at the proposed operating velocity $i_w = 0.0373$, $(i_m - i_w) = 0.0239$.

Solution From Figure 9, at 30° , $\Delta D = 0.33$, equivalent to an increase in V_{sm} of

$$0.33[(64.4) \cdot 1.65(25.6/12)]^{1/2} = 5.0 \text{ ft/s}$$

in USCS units, or

$$0.33[(19.62)(1.65)(0.65)]^{1/2} = 1.5 \text{ m/s}$$

in SI units.

The resulting limit velocity in the inclined pipe is $15.7 + 5.0 = 20.7$ ft/s, $4.8 + 1.5 = 6.3$ m/s, which is slightly less than the proposed operating velocity. At the operating velocity Eq. 31 gives the excess gradient, $\Delta i(30^\circ)$, as

$$\Delta i(30^\circ) = 0.0239 \cos 30^\circ + (1.65)(0.20) \sin 30^\circ$$

where 0.0239 represents the solids effect in the horizontal pipe. Thus $\Delta i(30^\circ)$ equals $0.0207 + 0.1650$ or 0.1857 . The clear water gradient i_w was 0.0373, giving a total value of 0.223, and on multiplying this by the suction-pipe length of 59 ft (18 m), the drop is found to be 13.2 ft (4.0 m) of water.

SYSTEM DESIGN AND ANALYSIS

Operating and Economic Considerations In order to compare the merits of different transport systems, it is necessary to have a measure of the energy required to move a given quantity of product over a given distance. In slurry transport, the solids are normally the "payload" while the conveying liquid is merely the "vehicle." The specific energy consumption is therefore to be related to the solids transported rather than to the mixture. However, the power which the pump must supply is used to drive the slurry as a whole, and is given by $\rho_w g Q_m H$ where H is the head in height of water and Q_m is the volumetric flow rate of the mixture. The payload of solids is delivered over the line length, L , at a volumetric rate that is the product of the delivered solids concentration C_{vd} and the mixture flow rate. If the application is a hoisting system, with the line directed vertically upwards, the rate at which energy is being added to the solids is $\rho_s g C_{vd} Q_m L$ or $\rho_w g S_s C_{vd} Q_m L$, and the efficiency of the system can be obtained by dividing the power input, giving $S_s C_{vd} L/H$. In this case the specific energy consumption is the inverse of the efficiency i.e.,

$$\text{SEC} = \frac{H}{S_s C_{vd} L} \quad (33)$$

where, as before, H is the head supplied by the pump in height of water.

In the more usual case of an essentially horizontal pipeline, potential energy will not be added to the solids, and thus from the physicist's viewpoint no work will be done. However, in industrial practice transporting solids along the line represents useful work in the economic sense. We can no longer speak of efficiency in the physicist's terminology, but the appropriate economic measure of specific energy consumption is still given by Eq. 33. With H/L now equal to the friction gradient, i_m , the expression is written

$$\text{SEC} = \frac{i_m}{S_s C_{vd}} \quad (34)$$

To obtain a value in horsepower-hour/ton-mile, this ratio is multiplied by 5.33, and for kWh/tonne-km the factor is 2.73. If the power supplied to the pump is required, the expression must be divided by pump efficiency, and the efficiency of the drive train or motor can be taken into account in the same way.

The lower the SEC, the more energy-effective the pipeline is as a means of transport. Since S_s is fixed by the nature of the solids, the quotient i_m/C_{vd} is the basic variable to be considered.

Homogeneous Slurries Figure 10 is from Chap. 15 (written by Clift) of Wilson et al. (1997). The figure shows in simple generalized terms, the effect of changes in solids concentration on the resistance of a piping system to flow of a homogeneous, non settling slurry. At velocities sufficiently high for the slurry to be in turbulent flow, the head lost in flowing through the system (measured as head of a fluid with the slurry density) can be estimated by treating the slurry as a fluid with an effectively constant friction factor, as shown in connection with Eq. 13. However, in laminar flow, to the left of the turbulent curve, the system head varies strongly with concentration, as shown schematically in Figure 10 where $C_1 < C_2 < C_3$. The transition between laminar and turbulent flow is indicated roughly by the intersection of the laminar system characteristic with the turbulent curve. Therefore, as illustrated, the transition velocity increases somewhat with solids concentration.

To examine the operability of such a system with centrifugal pumps, we now superimpose the pump head-discharge characteristic on to the system characteristic. For fine slurries the head delivered by the pump, evaluated in terms of the slurry density, is virtually unaffected by the solids. Therefore the characteristic of a fixed speed pump appears as a single curve in the coordinates of Figure 10, independent of slurry concentration.

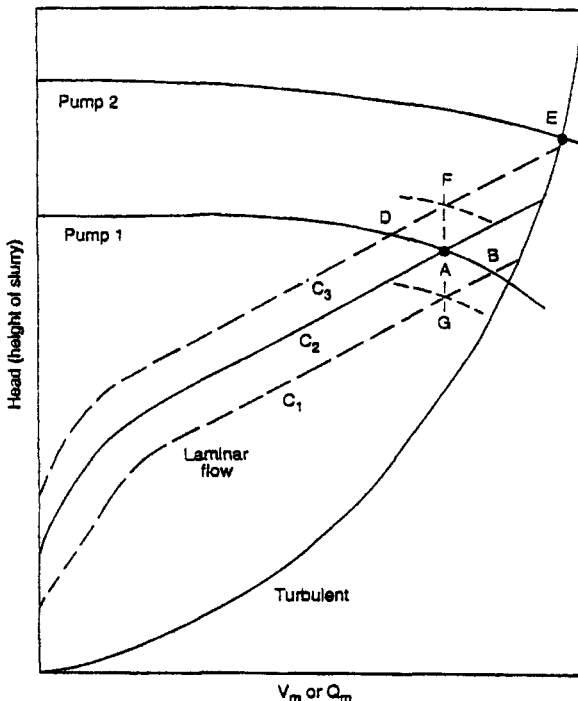


FIGURE 10 Schematic system and pump characteristics for flow of a homogeneous slurry at three concentrations (from Wilson et al., 1997)

Consider first a system which has been designed to operate in the laminar range, corresponding to point A in Figure 10 with slurry concentration C_2 . The curve labeled "Pump 1" represents the characteristic of a pump selected for this operating point.

Figure 10 shows the effect of varying solids concentration. If it decreases to C_1 then the system operation shifts to point B, the new intersection of the system and pump characteristics. Similarly, a concentration increase to C_2 shifts operation to point D. It was shown on Figure 3 that relatively small variations in concentration can have strong effects on the laminar flow curves. Thus the shifts from point A may result from quite small changes in slurry consistency. Nevertheless, as shown by Figure 10, the associated changes in mean velocity (and hence solids throughput) are amplified by the way the system and pump characteristics intersect. Thus steady operation in laminar flow with a fixed-speed centrifugal pump and with no flow control valve is possible only if the slurry consistency is tightly controlled.

For variable solids concentration, two alternatives are available. One is to operate in turbulent flow, at a point like E in Figure 10. "Pump 2" shows the head-discharge characteristics of a centrifugal pump selected for this duty. By operating on the turbulent system characteristic, variations in slurry consistency can be accommodated without significant changes in mixture velocity. This is probably the main reason why many designers prefer to use turbulent flow even for homogeneous slurries. The design point is then selected to be in the turbulent range for the highest solids concentration expected. However, it should be noted that this will not correspond to the lowest specific energy consumption. This is an example of the general point that design for variable operation is usually incompatible with design for minimum energy consumption.

The other option is to use a variable-speed pump. This is illustrated by points F and G in Figure 10. If the solids concentration increases from C_2 to C_3 , then the pump speed is increased to keep a constant flow rate. The corresponding increase in pump speed raises the pump head-discharge characteristic to the broken curve shown passing through F. Similarly, if the concentration falls from C_2 to C_1 , the pump is slowed down to give the characteristic passing through point G. Again, it must be noted that there is an economic penalty for designing for variable concentration, this time represented by the cost of the variable-speed drive and the energy losses in the drive train.

Settling Slurries We now turn to problems of system operability for settling slurries. (For a more detailed treatment see Wilson et al., 1997.) Figure 11 shows typical "system characteristics" for a settling slurry at three delivered concentrations, in terms of head of slurry, j_m . Only the frictional contribution for horizontal transport is considered here. The form of the system characteristics is typical of heterogeneous flow of a settling slurry. The three slurry characteristics are scaled for three notional delivered relative densities, i.e., three values of S_{rd} . The slurry curves all show the minima typical of a heterogeneous slurry flow. As is typical for heterogeneous slurries, the minimum friction gradient occurs at velocities above the deposition point and the position of the minimum moves to somewhat higher velocities as the solids concentration increases.

Figure 11 also shows the point where j_m equals i_w . At this crossing point (but not elsewhere) the heterogeneous slurry behaves like an "equivalent fluid," with a value of 1.0 for the coefficient A' in the homogeneous-flow equation (Eq. 5).

It is now appropriate to consider the behavior of this system with one or more variable-speed centrifugal pumps, with characteristics shown in Figure 11 as "Pump (water)" and "Pump (mixture)." The latter decreases with increasing concentration, due to the effects of solids on pump performance discussed earlier in this section. Therefore, to maintain a constant head and flow rate at point A in Figure 11, the pump rotational speed must be adjusted for the actual solids concentration.

It sometimes happens in industrial practice that the pipeline characteristics for different concentrations cross over at a single point at the water curve and at a suitable operating velocity, as shown schematically in Figure 11. This behavior is expected for closely graded materials with volumetric concentrations less than 20%, but may not apply for broad gradings or higher concentrations. The results with the broadly graded sand in Table 3 show that j_m was approximately constant (equal to 0.055 over a concentration range 20% to 40%) for the selected velocity = 15 ft/s (4.5 m/s). However, the operating point

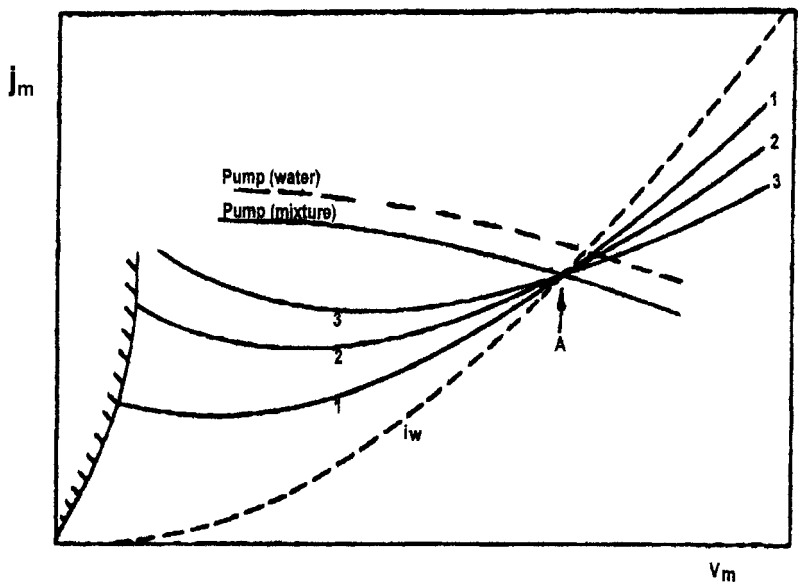


FIGURE 11 Schematic system and pump characteristics for heterogeneous (settling) slurry flow at various delivered concentrations (after Wilson *et al.*, 1997)

was not located on the water curve ($i_w = 0.041$). It follows from Table 3 that the slurry friction losses, j_m , exceeded the corresponding water losses, i_w , by 35%.

The operating data for the sand in Table 3 are of great practical interest since they demonstrate the energy-efficiency of transporting this product at very high solids concentrations. Experiments by Sellgren & Addie (1993) with this sand indicated that the pump solids effect was not influenced greatly by increased solids concentrations. For the highest concentration studied here, the maximum values of R_H and R_η were estimated to be 13 and 15%, respectively. An example including pipeline friction and the effect of solids on the water efficiency of the pump is given in Table 4.

It follows that the energy required to overcome friction losses decreased from 0.039 to 0.024 HPh/ton (0.032 to 0.020 kWh/tonne) when C_{vd} was increased from 20 to 40%. The decrease corresponds to about 37%. The capacity of the pipeline is in this case doubled by the increase in concentration. Because the pump solids effect for this product and pump

TABLE 3 Comparison of slurry friction losses and water friction losses for a broadly graded sand with $d_{80}/d_{50} \approx 7.1$. Data from Sundqvist *et al.* (1996a)

Product		Sand	
Max. particle size, in. (mm)		0.4 (10)	
Average, d_{50} , in. (mm)		0.025 (0.65)	
Portion of fine particles, X_h , %		0	
Pipeline—Diameter, in. (m)		12 (0.3)	
Velocity, ft/s (m/s)		15 (4.5)	
Water friction losses, i_w		0.041	
Conc. by volume, C_v	0.20	0.30	0.40
Slurry friction losses, j_m	0.055	0.055	0.055

TABLE 4 Data for the sand slurry of Table 3 in a 12 in. (0.3 m) diameter pipe using a 14 in. (0.35 m) by 12 in. (0.3 m) conventional heavy-duty pump with an impeller diameter of 36 in. (0.91 m). Pipeline length = 852 ft (260 m), flow rate = 5200 US gpm (328 l/s) and velocity = 15 ft/s (4.5 m/s). Volumetric concentrations 20, 30, and 40% (from Sellgren and Addie, 1996)

	Concentrations by volume (%)		
	20	30	40
Capacity, ton/h (tonne/h)	690 (627)	1035 (941)	1380 (1255)
Energy to overcome pipeline friction, HPh/ton (kWh/tonne)	0.039 (0.032)	0.029 (0.024)	0.024 (0.020)
Pump head reduction, R_H (%)	8	11	13
Required rotary speed (rpm)	333	338	342
Pump water efficiency (%)	79.5	79.4	79.3
Pump efficiency reduction, R_η (%)	6	11	15
Pump efficiency (%)	74.8	70.7	67.4
Total energy, HPh/ton (kWh/tonne)	0.052 (0.043)	0.041 (0.034)	0.037 (0.030)

remained fairly moderate also for the highest concentration, the total energy requirement per tonne decreased from 0.052 to 0.037 (0.043 to 0.030) with an increased C_{vd} value from 20% to 40%. The lowering then corresponds to about 30%.

The attrition of coarse and angular particles during circulation in a test-loop may have a strong influence on friction losses and pump performance. However, the results in Tables 3 and 4 with high solids concentrations and coarse particles embedded in a sand slurry are not expected to have been significantly influenced by circulation effects. In large-scale loop testing special loading and quick measurement procedures are now used to evaluate the interaction of particle attrition, pipeline friction, and pump performance. In general, designing a system to transport a slurry combining broad grading and high concentration requires pilot plant testing using the same material that will be pumped in the prototype line.

The solids concentration used throughout this section (denoted C_v or C_{vd}) refers to delivered concentration, and this is incorporated in the design-oriented equations. Large particles tend to have a lower velocity than small ones, and thus a larger resident (*in situ*) concentration in the pipeline, and this feature should also be considered in designing once-through pipelines on the basis of recirculating test-loop data.

For long-distance freight pipelines it is worthwhile to devote considerable effort to preparing and controlling slurry properties. For such lines solids concentrations range from 0.25 (25%) for ores with S_g of, say, 5.0 to 0.40 (40%) for coal with $S_g = 1.4$. Velocities are usually maintained below 6.6 ft/s (2.0 m/s). For a typical 10 in. (0.25 m) pipe, annual throughput of solids range from about 2.8 M tons (2.8 M tonnes) for ore to 1.4 M tons (1.3 M tonnes) for coal. The corresponding specific energy consumption, including a total pump and drive-train efficiency of 75%, are respectively, 0.13 and 0.17 HPh/ton-mile (0.065 and 0.085 kWh/ton-km). For such cases the energy cost for pumping amounts to roughly half the annual operating cost. This cost is, say, 20% to 30% of the total annual cost, which is seen to be dominated by repayment of capital expenditure.

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SECTION 4.2

APPLICATION AND CONSTRUCTION OF CENTRIFUGAL SOLIDS HANDLING PUMPS

**GRAEME ADDIE
ANDERS SELLGREN**

This section discusses the application of centrifugal pumps to the transport of slurries as introduced in Section 4.1.

Where the distance is large and the size of the solids less than about 100 microns, positive displacement pumps are usually applied. If the solids to be transported are smaller than about 60 mm in size, water is readily available or already part of the process, and the distance is less than 12 mi (20 km), a pipeline driven by centrifugal slurry pumps is usually more cost-effective than belt conveyors, trucks, or railway transportation.

Types of solids transported vary from sag mill feed in a copper mine, iron ore, phosphate matrix, coal, rock salt, tar sands, red mud, various types of waste tailings, and crushed rock and silt. An 18 in (0.45 m) diameter pipeline, for example, is capable of transporting as many as 2200 tons/h (2000 tonnes/h) of sand. This equates to about one hundred truckloads per hour over the distance involved.

The application that involves the largest quantities is in the dredging industry, continually maintaining navigation in harbors and rivers, altering coastlines, and mining material for landfill and construction purposes. Because a single dredge may be required to maintain a throughput of 7000 tons/h of slurry or more, very large centrifugal pumps are used. Figures 1 and 2 show, respectively, an exterior view of a dredge pump on test, and a view of a large slurry pump impeller (Addie and Helmly, 1989).

Because the aim is to transport solids and not water, the higher the concentration of solids, the better for energy consumption and capital cost. This does, however, mean that wear due to the abrasive solids will be significant, requiring a special type of centrifugal pump design and the use of special materials. Slurry pumps may be selected for low-concentration dirty-water service. Most of what follows, however, is about pumps designed and built to give cost-effective operation for heavy-duty slurry service, probably involving significant wear.

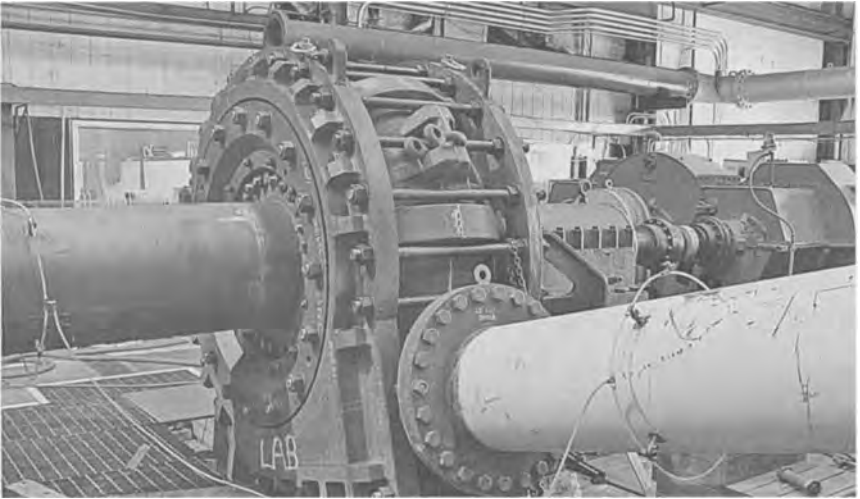


FIGURE 1 Testing a dredge pump at the GIW hydraulic laboratory

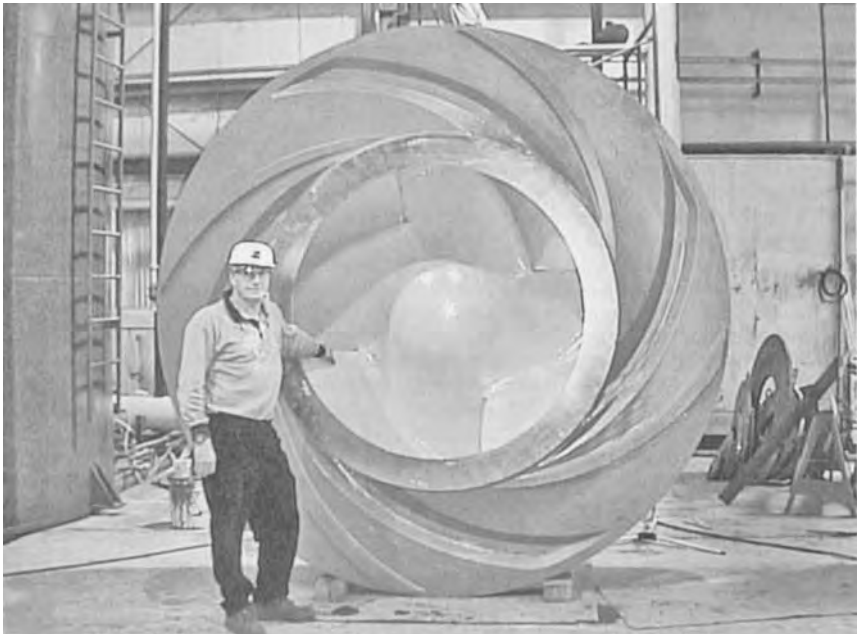


FIGURE 2 Dredge pump impeller, 113 in (2.87 m) diameter

SLURRY PUMP TYPES

The majority of centrifugal slurry pumps that fit the above definition are constructed as the single-stage horizontal radially split type. A sectional arrangement drawing of a single-wall metal type is shown in Figure 3. Where elastomers are used, they must be supported with an outer casing or sideplate. This type is commonly referred to as a double-wall type. An example of a double-wall rubber-lined pump is shown as Figure 4.

Vertical cantilever and vertical submersible centrifugal pumps are also built in significant quantities and serve mostly in general cleanup and other service. These pumps on occasion are called upon to operate as transporting devices and, depending on the design and service, include the special features described later.

CENTRIFUGAL SLURRY PUMP HYDRAULIC DESIGN

In order to reduce wear, the hydraulic design of a slurry pump should be such that the rotating wetted component and fluid velocities must be kept as low as practical while hard metal or elastomer wetted-section thicknesses are increased. Impeller meridional sections are also usually kept close to rectangular and with front sealing on a vertical face to minimize the axial rotating surfaces. Collectors (casings) are usually semi/volute (or even annular) to give better wear over a wider range of best-efficiency-point quantity (flow rate) (BEPQ).

Where larger solids are to be pumped, the inside shroud impeller widths are usually oversized, the number of vanes is reduced to two or three, and the vane overlap may be reduced. All of these distortions modify or reduce the hydraulic performance. Given the variety of types (and severity) of services, the design solution that provides the best compromise between wear life and efficiency and provides the end user with the overall lowest total cost of ownership is not simple to achieve. Although specific to a particular slurry type, Kasztejna et al. (1986) and Cooper et al. (1987) provide a good insight into

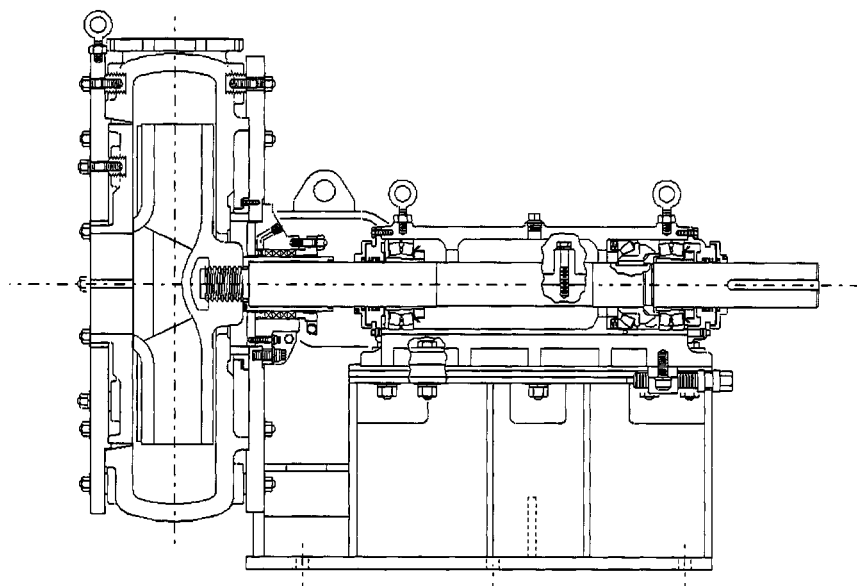


FIGURE 3 Typical single-wall hard iron slurry pump

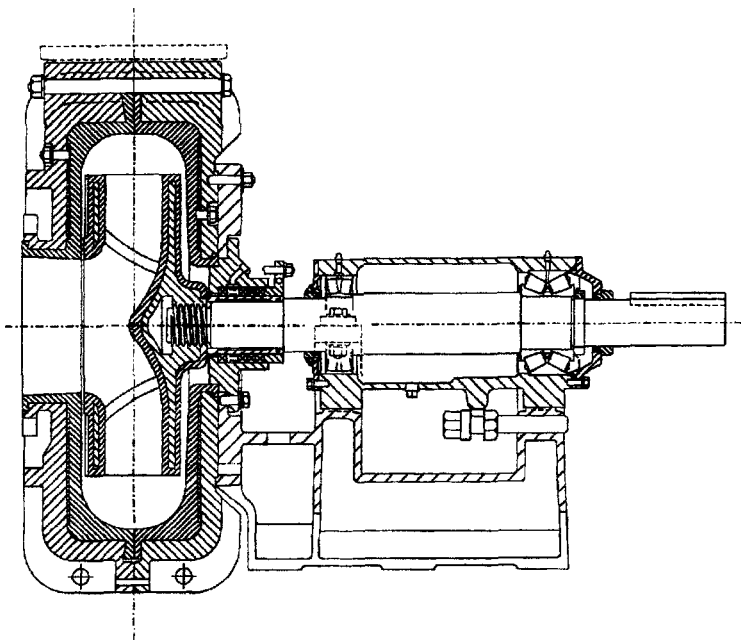


FIGURE 4 Typical double-wall elastomer-lined slurry pump

some of the design tradeoffs and special problems involved in the design of a centrifugal slurry pump.

In general, slurry pumps are of low specific speed (as defined in Section 2.1.1), in the range of 750 to 2000 for N_p (14.5 to 39 for n_q and 0.27 to 0.73 for Ω_p). They employ three to five vanes, have impeller inside shroud widths that are 75 to 200% wider than those a normal water pump, and use impeller vane and shroud thicknesses two to three times that of a water pump. The results of these distortions flatten the constant-speed head-flow curves and reduce efficiency by five or so percent. Typical head coefficients (as defined in Section 2.1.1) and efficiencies are shown in Figures 5 and 6.

In order to minimize wear over different ranges of percent of BEPQ, operation slurry pump casings vary from the true volute water pump T type (shown schematically on Figure 7) through the semivolute C type to the essentially annular A type. There are also a few examples of what is called the OB type with recessed tongue and special extended neck intended for limiting wear in the tongue area due to excessive recirculation (which itself is symptomatic of misapplication). The general applicability of each of these is shown in Figure 8, taken from Addie and Helmly (1989).

Impeller types may be roughly categorized also as being of the old-style square meridional section radial vane RV type (still common in rubber-lined pumps where it is difficult to mold a twisted vane), the most common rounded rectangular meridional section with twisted vanes ME type, and the HE type which is closer to a water pump.

Each combination of the types illustrated in Figure 7 has its own hydraulic performance and wear characteristics. The HE/T combination generally has the highest performance, but is not necessarily the most forgiving for wear, whereas the ME/C combination is capable of respectable efficiency while at the same time having more predictable wear performance. In extremely heavy-duty wear service, operation must be at low discharge flow rates—below BEPQ. The A type shell, or even the OB type, could be best for wear where a pump has been misapplied badly. For additional information on the selection of different hydraulic types, see Addie et al. (1996).

Head coefficient vs Specific speed

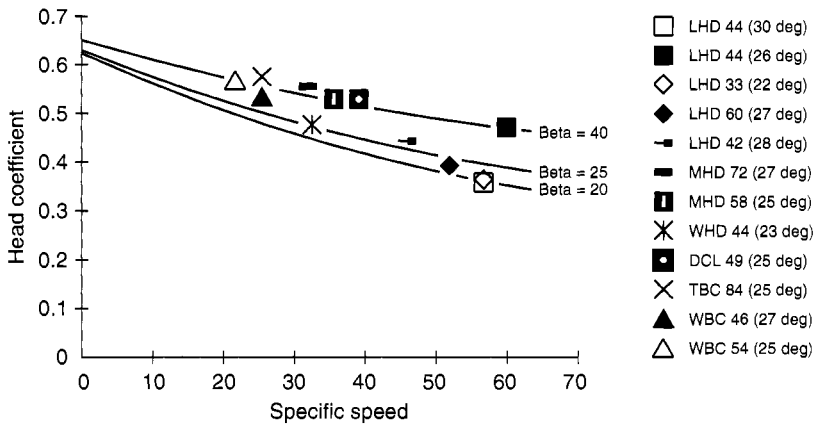


FIGURE 5 Head coefficient versus specific speed n_q , based on m^3/s , m , and rpm (from Addie and Helmly, 1989) [Note: for specific speed N_q based on gpm , ft , and rpm , multiply n_q by 51.65; for universal specific speed I_q , divide n_q by 52.92.]

Efficiency Comparisons with Standards

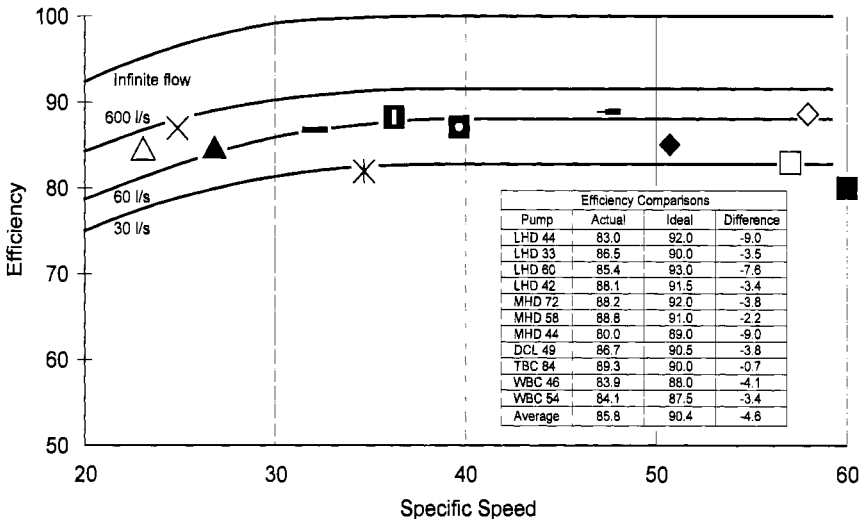
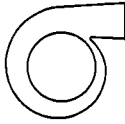


FIGURE 6 Pump efficiency as a function of specific speed n_q , based on m^3/s , m , and rpm (from Addie and Helmly, 1989) [Note: for specific speed N_q based on gpm , ft , and rpm , multiply n_q by 51.65; for universal specific speed I_q , divide n_q by 52.92.]

SHELL TYPES

T - TIGHT CUTWATER
Volute Type



C - CONVENTIONAL
Semi-Volute Type



A - ANNULAR
Annular Type

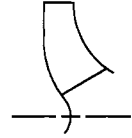


OB - "ODD BALL"
Extended Neck Type

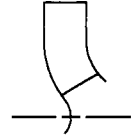


IMPELLER TYPES

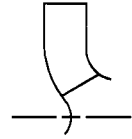
HE - HIGH EFFICIENCY
Twisted vanes in rounded and
narrowed meridional section



ME - MEDIUM EFFICIENCY
Twisted vanes in rounded
meridional section



RV - RADIAL VANE
Radial vanes in rectangular
meridional section



(The ME and RV type impellers are
usually interchangeable and use the
same suction liner and plate)

FIGURE 7 Types of shells (casings) and impellers

Shell Shape

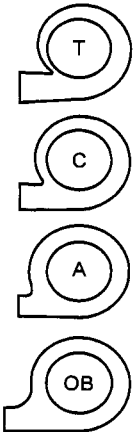
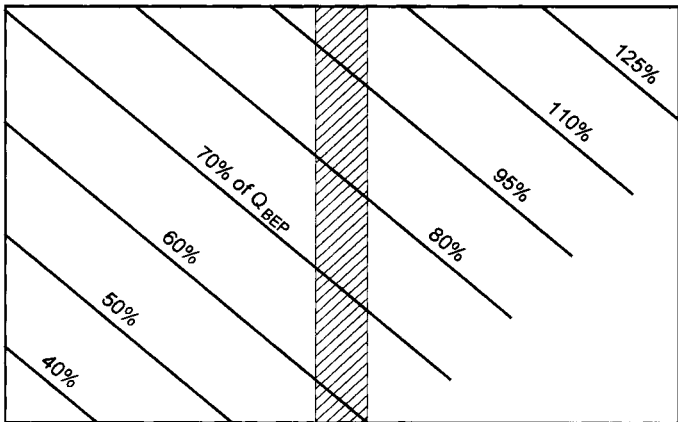
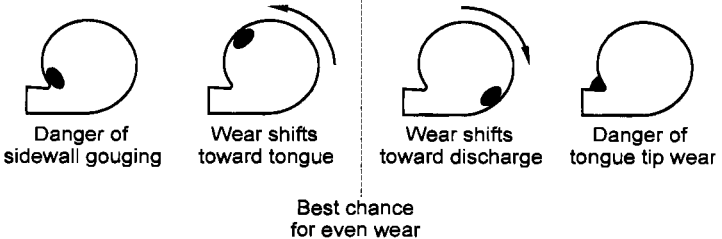


FIGURE 8 Trends in location of casing belly wear

Impeller and shells using elastomers such as rubber, neoprene, and urethane tend to be limited to impeller tip speeds less than about 75 ft/s (23 m/s), although this can rise with stiffer elastomers, at some expense to wear life. Impellers employing elastomers require higher available NPSH because of the thicker impeller vane sections needed, and this condition may limit their use, for example, in pumping flue-gas desulphurization slurries.

MECHANICAL DESIGN OF SLURRY PUMPS

Given the distortions noted earlier the mechanical design of a slurry pump is similar to that of a water pump. Slurry specific gravities (SG) of up to 1.6 and higher may have to be pumped, so the shaft and bearings of necessity must be more robust. Figures 3 and 4 show typical pump bearing housings.

Impeller attachment by an Acme-type screw has generally been found best capable of carrying the heavy loads required, and the connections can be manufactured economically in the hard materials most commonly employed in impellers. The large loads associated with heavy-duty service require roller bearings with separate roller thrust bearings. Designs with ball bearings are suited only for light-service pumps.

The absence of radial sealing in slurry pumps allows shaft deflections larger than those found in water pumps, and these may limit the life of stuffing boxes and mechanical seals. Newer and better designs, however, tend to have shorter shafts and smaller deflections, extending the seal life. A conventional packed stuffing box is still the simplest and most common rotating-assembly seal. Configurations are similar to those used for water pumps, with the lantern-ring supplying a clear-water flush to the center for minimum dilution, or to the product side for maximum life. An expeller-type seal is popular where dilution of the product is unacceptable. These seals are limited to one stage and involve additional efficiency losses, which usually range to 3% or more.

Mechanical seals are now available that take their coolant from the product and operate with no clear-water flush; they are mostly of the single partially balanced type. In some cases, where a pump may run dry, a double type of mechanical seal must be used. At present, mechanical seals are the preferred shaft sealing method for fine-particle light-duty service such as pumps for flue gas desulphurization. These seals are also fairly widely used in the aluminum industry for red-mud pumping service. A mechanical seal described in Maciejewski *et al.* (1993) is used for pumps handling tar sands tailings at pump discharge pressures up to 350 lb/in² (2400 kPa), with d_{50} about 120 microns and average slurry density near 0.0578 lb/in³ (1600 kg/m³). In this case, the process fluid is at an average temperature of about 130°F (55°C) and some 0.16 gpm (0.01 L/s) of water is used for external cooling. In this application, representative of the limits of mechanical seal technology at the present, the average seal life is about 3000 h.

As noted before, pump casings made entirely of elastomeric materials have insufficient strength to withstand the pressure loads, so it is necessary to have an outer casing of some sort, a configuration that is commonly called a double-wall design. For versatility, some manufacturers make these designs for interchangeable wetted internal shell components of hard metal, rubber, or urethane as the service warrants. For very large slurry pumps, the double-wall configuration is heavy and costly. As a cost-effective design for this case, it is worth considering tie-bolt construction using high-tensile outer side plates over a hard-metal plate and liners as shown in Figure 1.

Further, single-wall designs with the main shell casing in hard metal are simple to construct and maintain and thus are very popular. These can be used unlined or provided with liners of bonded rubber or other material. Ceramic materials are now available that are capable of several times the life of both elastomer and hard-metal components. At the moment, the cost-effectiveness of these materials is such that their use tends to be limited to areas of high wear and other selected areas of wetted surfaces.

In a shell of single-wall metal design, the wearing components must carry the water load. Wear tends to progress until leakage of the shell occurs, but the wear is almost always localized so leakage occurs before the structural integrity of the assembly is affected. Single-wall designs may easily be ribbed or thickened locally to increase strength

and wear life in a particular area. In small slurry pumps, minimum casing thicknesses usually produce safe designs of adequate pressure rating.

SELECTION OF SLURRY PUMPS

As noted earlier, service conditions vary significantly, so when selecting a slurry pump, a recommended approach is to select a service class and limit nozzle (branch) and other velocities according to that. Figures 9 and 10 from Wilson et al. (1997) outline an approach that can be used.

For a given service class, Table 1 provides a guide to maximum impeller, nozzle (branch), and other velocities that should give reasonable wear life and keep the cost and size of the pump to a minimum.

Except in the case of dredges (which are usually diesel driven), most slurry pumps are driven by electric motors. Because of the difficulty associated with trimming impeller diameters and variations of system head, it is common for plant slurry pumps up to 250 hp (185 kW) to have V-belt drives. For larger pumps, four pole motors and gearboxes are most common, with one or more variable speed units when there are several (usually up to six) units operating in series.

SOLIDS EFFECT ON CENTRIFUGAL PUMPS

The presence of solid particles in the flow tends to produce adverse effects on pump performance, and detailed information on these effects is needed to achieve reliable and energy-efficient operation. Head-flow curves for centrifugal pumps are often rather flat. Also, the pipeline characteristic for slurry flow usually displays a slow rise with increasing discharge. As a result, the two characteristic curves often intercept at a rather shallow angle. Therefore, even a small diminution in pump head can produce a disproportionately large drop in flow rate. The resulting difficulties in operation can cause large and expensive systems to run inefficiently or not run at all.

The effects on pump characteristics are shown schematically in Figure 11, which is a definition sketch for illustrating the reduction in head and efficiency of a centrifugal pump

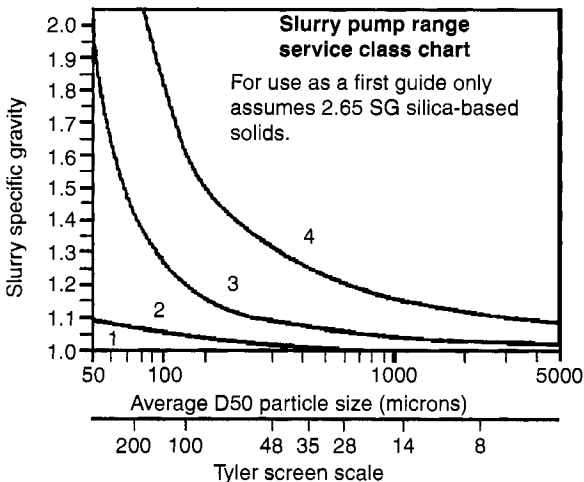


FIGURE 9 Slurry pump range service class chart

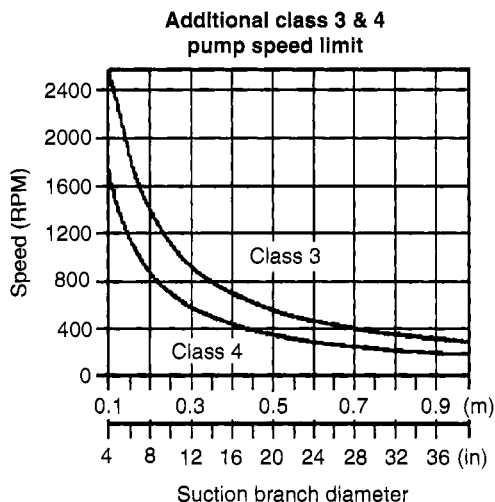


FIGURE 10 Pump speed limit chart for classes 3 and 4

operating at constant rotary speed and handling a solid-water mixture. In this sketch, and the discussion that follows, η_m represents the pump efficiency in slurry service, and η_w is the clear-water equivalent. Likewise, P_m and P_w are the power requirements for slurry service and water service, respectively. The head H_m is developed in slurry service, measured in height of slurry, whereas H_w represents the head developed in water service in height of

TABLE 1 Recommended operating limits for slurry pumps

Operating* Limits	Shell Type	Service Class			
		1	2	3	4
Maximum discharge (R) Velocity		40. (ft/s) 12.2 (m/s)	32. (ft/s) 9.8 (m/s)	27. (ft/s) 8.2 (m/s)	20. (ft/s) 6.1 (m/s)
Maximum throat (R) velocity		50. (ft/s) 15.2 (m/s)	40. (ft/s) 12.2 (m/s)	30. (ft/s) 9.1 (m/s)	20. (ft/s) 6.1 (m/s)
Recommended percent range (C) of BEP flowrate	Annular (A)	20–120%	30–110%	40–100%	50–90%
	Semivolute (C)	30–130%	40–120%	50–110%	60–100%
	Near Volute (T)	50–110%	60–130%	70–120%	80–110%
	Annular/Oblique Neck (OB)	10–110%	20–100%	30–90%	40–80%
All metal pump maximum impeller (R) peripheral speed		8500 (sfpm) 43.2 (m/s)	7500 (sfpm) 38.1 (m/s)	6500 (sfpm) 33.0 (m/s)	5500 (sfpm) 27.9 (m/s)
Rubber lined pump Maximum impeller (R) peripheral speed		5500 (sfpm) 27.9 (m/s)	5000 (sfpm) 25.4 (m/s)	4500 (sfpm) 22.9 (m/s)	4000 (sfpm) 20.3 (m/s)

[Note: sfpm = ft/min; for ft/s, Divide by 60]

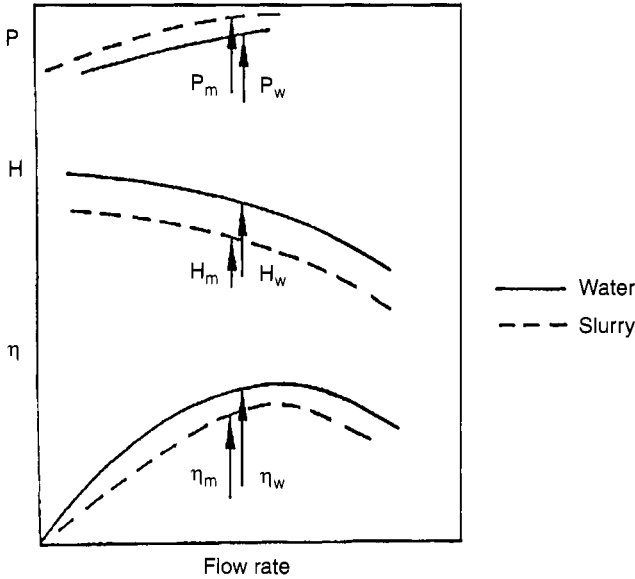


FIGURE 11 Effect of slurry on pump characteristics (schematic)

water. The head ratio H_r and the efficiency ratio η_r are defined as H_m/H_w and η_m/η_w , respectively. The fractional reduction in head (the head reduction factor) is denoted by R_H and defined as $1 - H_r$; for efficiency, the fractional reduction (efficiency reduction factor) is R_η , given by $1 - \eta_r$.

Test loop results for a large variety of solids at the GIW Hydraulic Testing Laboratory have been put together in a generalized design diagram (Figure 12) for larger heavy-duty pumps and smaller pumps having impeller diameters of 8 in (0.2 m) to 16 in (0.4 m). The diagram gives R_H and H_r in terms of pump impeller diameter (D) and solid average size (d_{50}) at a solids concentration by volume (C_v) of 0.15 (15%) with a solids density ratio 2.65 and a negligible amount of fine particles ($X_h = 0$). For example, a pump with an impeller diameter of 32 in (0.81 m), R_H becomes 14% ($H_r = 0.86$) for a d_{50} of 0.39 in (10 mm).

Corrections for various values of C_v , S_s , and X_h will now be given together with estimations of the reduction in efficiency, R_η . Based on experimental results discussed in Wilson et al. (1997), the R_H values obtained from Figure 12 may be multiplied by the following factors when the solids density ratio and the fine particle content are different from 2.65 and zero, respectively.

$$\text{solids density ratio, } S_s: \left[\frac{S_s - 1}{1.65} \right]^{0.65}$$

$$\text{fine particle content, } X_h: (1 - X_h)^2$$

For example, with $S_s = 4.4$ and $X_h = 0.08$ (8%), the factors become 1.6 and 0.85, respectively. The pump previously mentioned with an impeller diameter of 36 in (0.91 m) had R_H of 14%. This value is now corrected to give

$$R_H = 14 (1.6) 0.85 = 19\%$$

R_H can normally be related linearly to C_v for cases with C_v less than 15 to 20%. Thus with $C_v = 6\%$ in the previous example, R_H will be $19 (6/15) = 8\%$ because the reference value for C_v in Figure 12 is 15%.

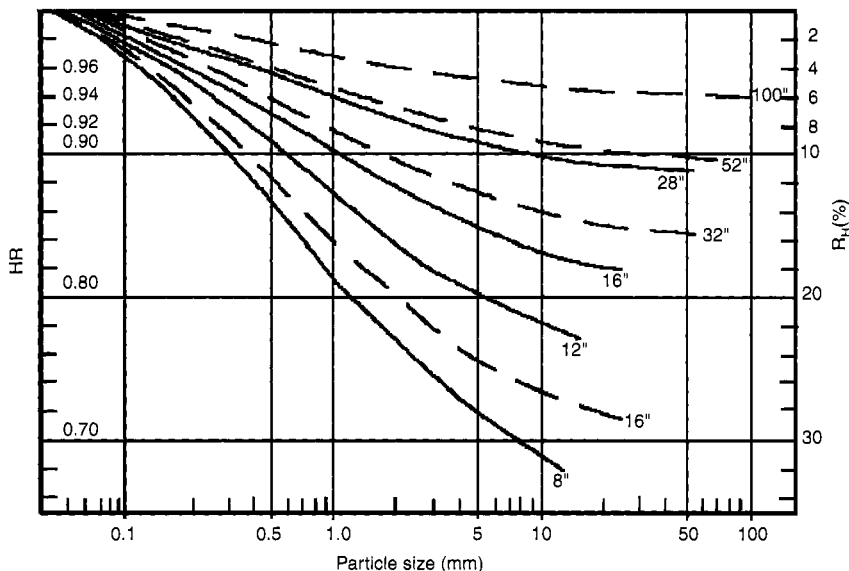


FIGURE 12 Effects of particle size (d_{50} , mm) and impeller diameter (D in) on H_r and R_H . For solids concentration by volume, $C = 15\%$ with solids density ratio, $S_s = 2.65$ and a negligible amount of fine particles ($X_f = 0$). Solid lines represent smaller newer design pumps while dashed lines represent conventional heavy-duty pumps. Adapted from GIW Industries Inc., U.S.A. [Note: for particle size in inches, multiply by 0.039; for impeller diameters in mm, multiply inches by 25.4.]

For the larger pumps in Figure 12, the reduction in efficiency, R_η is normally less than R_H . For the smaller pumps, with $D < 16$ in (0.4 m), R_η usually equals R_H ; however, it may be sensitive to solids properties. Independent of the pump type in Figure 12, R_η may exceed R_H if C_v exceeds about 20%, indicated on Figure 13.

In opposition to these tendencies, there have also been investigations where the reduction in efficiency remains small at very high concentrations, specifically with broad particle-size distributions.

With equivalent reductions, $R_H = R_\eta$, the power consumption increases directly with the relative slurry density.

$$P_m = S_m \cdot P_w \quad (1)$$

with $R_\eta < R_H$ ($H_r > \eta_r$), Eq. 1 becomes

$$P_m > S_m \cdot P_w$$

that is, the power consumption becomes larger than that given by Eq. 1.

Finally, the calculated reductions may be underestimated if the solid particles are coarse and very angular. For example, the solids used in the preceding examples consisted of crushed angular particles and actually gave reductions in head and efficiency well over 10% (Sellgren et al., 1997), compared to $R_H = 8\%$ calculated here based on Figure 12.

On the other hand, with very fine particles, or large X_f -values and high solids concentrations, most slurries are practically nonsettling. In this case, the solid's effect is mainly related to the rheological behavior. When pumping nonsettling slurries in industrial applications, the slurry normally behaves in a non-Newtonian way, giving widely varied pump performance effects. In general, however, small pumps are affected more than large units when pumping highly viscous or non-Newtonian media. Furthermore, the influence on the efficiency is normally larger than on the head.

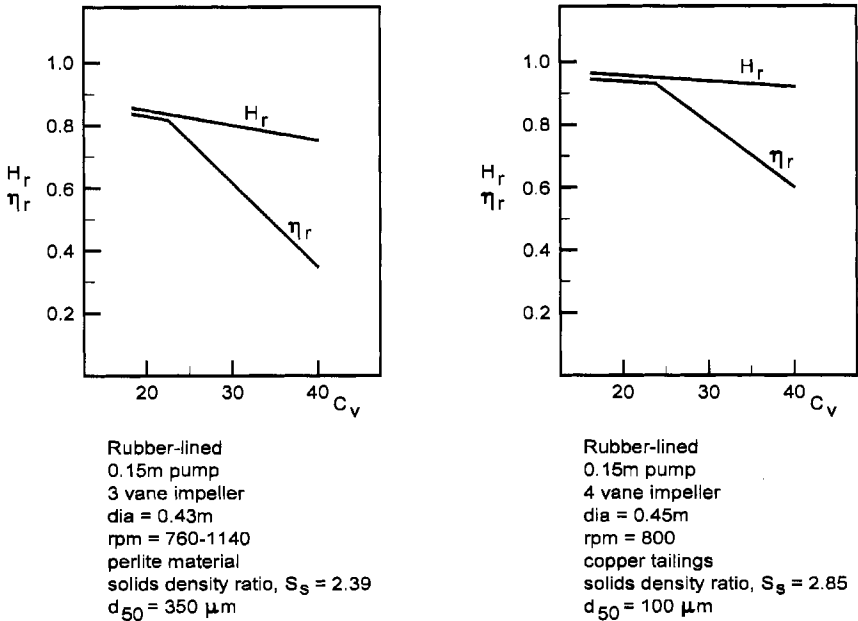


FIGURE 13 Head and efficiency reduction curves for two products (from Sellgren and Vappling, 1986) [Note: for dimensions in inches, divide mm by 25.4 and multiply m by 39.36.]

Occasionally, highly non-Newtonian slurries may be pumped by centrifugal pumps operating at flow rates much lower than the maximum-efficiency value Q_{BEP} . This may cause a dramatic drop in head, which creates an unstable head curve, seen, for example, in Figure 14.

WEAR MECHANISMS AND MATERIALS

As noted earlier, wear in a slurry pump can be severe. It is not possible, therefore, to use normal cast iron or other materials. Ideally, the life of components should be a year or more. In practice, even with the best design and materials, it can sometimes be as low as a month. Understanding wear and using the correct materials is key in the design of a slurry pump. The major erosive mechanisms are sliding abrasion and particle impact. The sliding-abrasion mode of wear typically involves a bed of contact-load particles bearing against a surface and moving tangentially to it, as illustrated on Figure 15. In pipelines, the stress normal to the surface is caused by gravity. For sliding abrasion, the erosion rate depends on the properties of particles and wear surfaces, the normal stress, and the relative velocity.

The normal stress is enhanced when the flow streamlines are curved, as in a pump casing. In this case there is a centrifugal acceleration, equal to u^2/r , where u is the local velocity and r is the radius of curvature of the streamlines. This acceleration can often be much greater than that of gravity, producing a commensurate increase in the normal stress between the moving contact-load solids and the wall material, and hence a greatly increased rate of sliding-abrasion wear. This type of behavior can cause a pump casing to wear through when adjacent straight-pipe sections of the same material have hardly been affected by erosion. Near the tongue of the pump casing, however, the other major wear type is more significant.

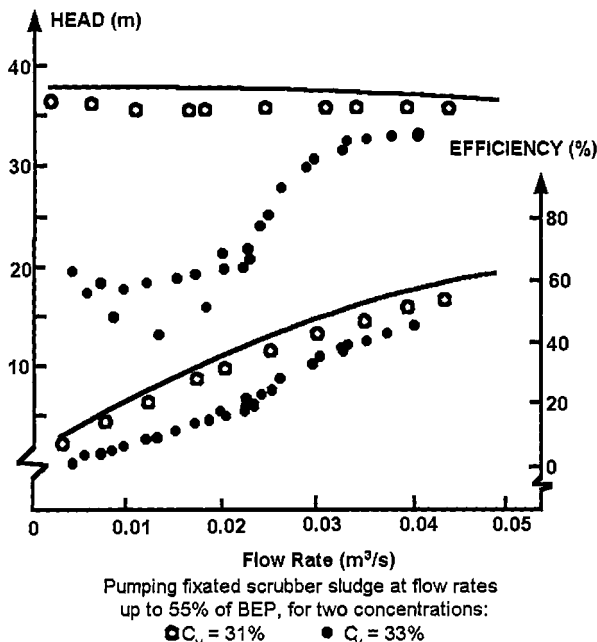


FIGURE 14 Effect of highly non-Newtonian slurries on pump head and efficiency (from Sellgren et al., 1997)
 [Note: for flow rate in gpm, multiply m^3/s by 15,850; for head in ft, multiply m by 3.28.]

The second type of wear is the particle-impact mode, which occurs where individual particles strike the wearing surface at an angle, despite the fact that the fluid component of the slurry is moving along the surface (Figure 16). Removal of material over time occurs through small-scale deformation, cutting, fatigue cracking, or a combination of these, and thus depends on the properties of both the wearing surface and the particles. Ductile materials tend to exhibit erosion primarily by deformation and cutting, with the specific type depending on the angularity of the eroding particles. Brittle or hardened materials tend to exhibit fatigue-cracking erosion under repeated particle impacts. For a given slurry, the erosion rate depends on properties of the wearing surface: hardness, ductility, toughness, and microstructure. The mean impact velocity and mean angle of impact of the solids are also important variables, as are particle characteristics such as size, shape, and hardness, and the concentration of solids near the surface.

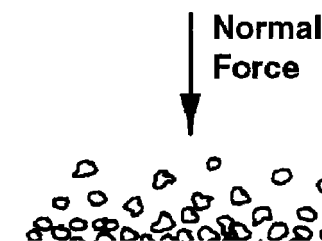


FIGURE 15 Erosion by sliding abrasion

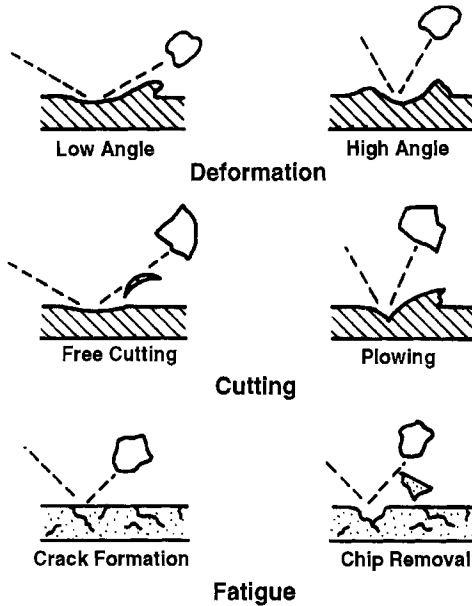


FIGURE 16 Mechanisms of particle-impact erosion

As wear mechanisms vary considerably, so also do the engineering materials that resist them. A survey of all the available wear-resistant materials and their applications would provide subject matter for a volume in itself. Nevertheless, there are a few groups of materials that have found widespread successful application against erosion in slurry systems, and thus merit particular attention. These materials fall into the broad categories of hardened metals, elastomers (rubbers and urethanes), and ceramics. Materials in each of these categories have their advantages and disadvantages, and the final choice of a material must account for wear resistance, strength, ease of maintenance, direct cost, and indirect costs such as plant downtime for maintenance, or failure, safety of operation, and effect on system efficiency.

Within the metals group, superior wear resistance is given by the high-alloy white cast irons, particularly nickel-chrome (e.g., NiHard), chromemoly (e.g., 15% chromium and 2% molybdenum), and high chrome (e.g., 27% chromium) alloys. These materials exhibit higher hardness (600 plus Brinell), but less toughness than steel alloys of similar tensile strength. The high carbon content and resultant massive carbides present in these metals gives them excellent resistance to all forms of particle erosion, especially in sliding or impact at low angles (cases often encountered in slurry applications). For the coarsest slurries, hard metal is common, principally the so-called NiHard, high-chrome, or chrome-molybdenum alloys of 600 to 700 Brinell hardness.

For slurries without coarse particles, elastomers such as rubber, neoprene, and urethane tend to wear better than hard metal, especially for liners and shells. This improvement applies only if there is no "tramp" (that is, extraneous material such as tools, bolts, or pieces of broken castings). Elastomer selection may also be determined by the corrosive

characteristics of the slurry. Natural soft rubber remains an excellent and economical pump lining material to handle fine abrasive and corrosive slurries. Carbon black can be added for additional strength, hardness, and tear resistance, to better withstand the impact of large particles in the slurry. Like all other materials, rubber has limitations that should be considered in the selection process. Wear rates in the presence of coarse material or high tip speeds may make it uneconomical, and tramp material or adverse suction conditions can tear the rubber. High temperature or the presence of oils or chemicals may require the use of one of the higher-cost synthetic materials. For example, neoprene has been found to tolerate higher temperatures and higher impeller tip speeds than can be used with natural rubber.

CALCULATION OF SLURRY PUMP WEAR

The general selection rules noted earlier aim at keeping wear to generally acceptable levels, but are otherwise nonquantitative. Finite element and other methods such as those described by Addie et al. (1996), Bross (2001), or Pagalthivarthi (1991) are now available. They allow particle velocities inside the impeller and in the suction liner nose face area adjacent to the impeller of a pump to be calculated, which, using material wear tests coefficients as described by Pagalthivarthi (1990), can be used to determine wear rates and component lives.

For example, numerical simulations allow calculation of wear around the periphery of a given pump shell for a given set of conditions. Figure 17 is an example of how wear varies with different flows from Roco et al (1983). Addie et al. (1987) shows also how a particular collector can be altered and optimized for the best possible wear.

Although all of these methods cannot predict three-dimensional turbulent wear, they are now opening up the possibility of better designs, better selection, and the calculation and evaluation of the total cost of ownership, Addie et al (1998), and the benefits that could come from that.

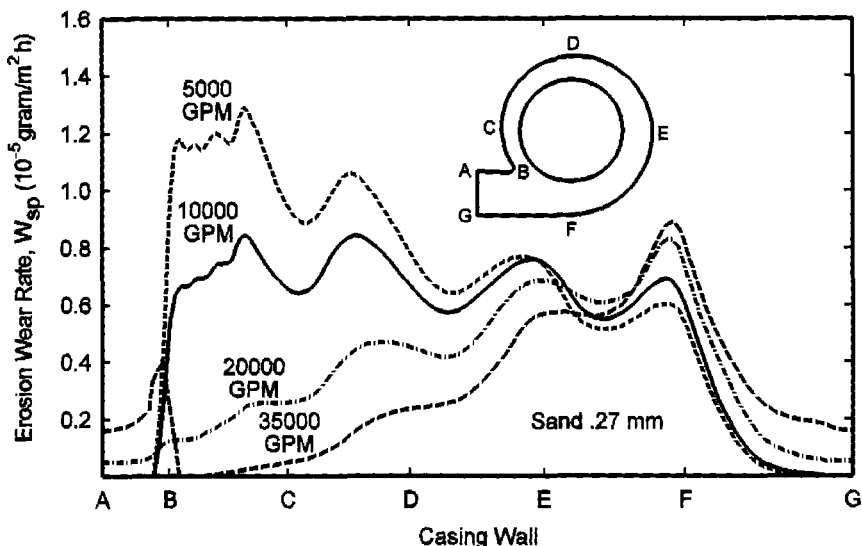


FIGURE 17 Erosion wear distributions calculated at different flow rates [Note: for flow in L/m, multiply gpm by 3.79; for particle size in inches, divide mm by 25.4.]

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SECTION 4.3

CONSTRUCTION OF SOLIDS-HANDLING DISPLACEMENT PUMPS

WILL SMITH

APPLICATIONS

Positive displacement pumps are especially suited for many solids-pumping applications because constant high pressures and good efficiency are achieved over the full range of pump capacities. Low relative velocities between abrasive liquid-solids mixtures and the pump parts minimize erosion. With centrifugal pumps, an increase in system resistance, such as a flow blockage caused by settled solids or the apparent viscosity increase characteristic of many slurries due to momentary capacity reduction, results in a self-defeating reduction in pump flow rate.

Positive displacement pumps are popular for a number of solids-handling applications. Each application presents peculiar demands for pump features.

Solids Transport Large-capacity positive displacement pumps at moderate pressures are used to pump coal and ores over relatively long distances. Solids for transport are usually slurried with water and are pumped at ambient temperatures. Double-acting horizontal piston pumps with large, “mud pump” valves and piston rings (Figure 1) typify transport service.

Most pipelines have multiple pumping stations at intervals along the route, dictated by topography and by pipeline and pumping station first and operating cost balances. Centrifugal booster pumps are employed at the first station to deliver the prepared slurry to the displacement pumps at a sufficient pressure to satisfy their NPSH, including acceleration head, requirement. Subsequent stations are located at points where there is sufficient residual pipeline pressure to fulfill the displacement pump's suction requirements. The capacity of each station is adjusted by pump speed control so the next station's inlet pressure is kept relatively constant.

Most stations muse multiple positive displacement pumps in parallel, with at least one standby pump. Capacity modulation requires but one of the pumps to be operating with

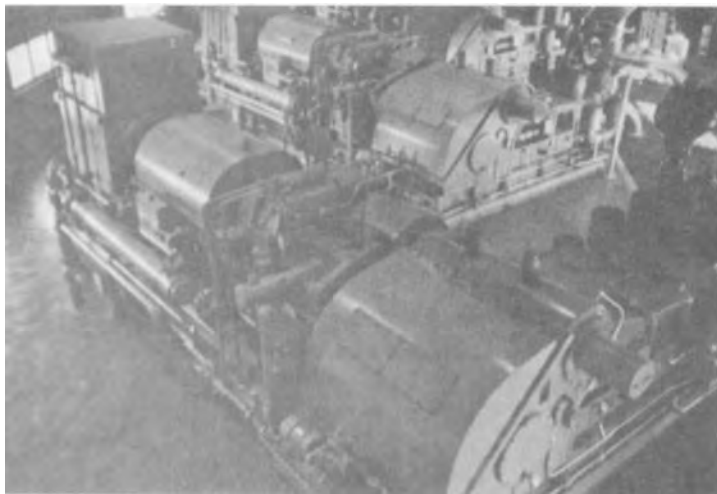


FIGURE 1 Horizontal double-acting duplex coal slurry pipeline pumps (Black Mesa Pipeline, Oil Well Div. of U.S. Steel)

speed control, although all are generally so fitted. Unlike centrifugal pumps, displacement pumps can provide full-rated pressures at all speed-controlled capacities.

Process Pumping Some chemical and petroleum processes require pumping of solids to high pressures for process reactions. Typical of these are bauxite ore in hot caustic for alumina plants, ground coal in water or coal liquids for synthetic fuel production, and grain in water for alcohol production (Figure 2).

Process streams involve a vast variety of liquids and solids. Concentrations may be as high as 70% solids by weight. Temperatures can reach 800°F (427°C). Pressures from several hundred to many thousands of pounds per square inch are accommodated.

Other Specialty Services A number of specialty services employ positive displacement pumps to handle solids. Some are relatively simple applications of standard catalog pumps, and others are single-purpose developments and thus have evolved with unique characteristics.

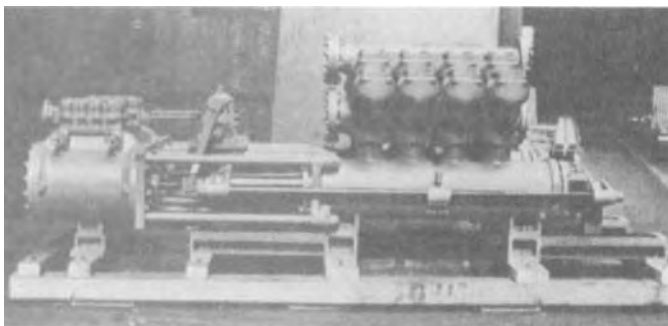


FIGURE 2 Horizontal double-acting plunger pump for slurry of grain mash and water for alcohol production (Flowsolve Corporation)

Mud pumps are used to inject drilling mud, which transports cuttings out of wells and lubricates the drill bit during well drilling. Because of the limited duration of a drilling project, the life of expendable parts (packing, valves, rods) is compromised in favor of size and weight for portability and ease of overhaul. *Sludge disposal pumps* are common in sewerage treatment plants, and *tailings pumps* move solids out of mines.

PUMP CONSTRUCTION

Solids-handling displacement pumps differ from ordinary displacement pumps in the means used to alleviate the deleterious effects of the solids on the packing, the displacement element (whether plunger, piston, or diaphragm), and the suction and discharge check valves. To protect parts from the ravages of the solids, (1) particles are prevented from entering close clearances, (2) operation is at lower speeds to reduce the effects of erosion and abrasion, or (3) the susceptible parts are made of wear-resistant materials. Because sacrificial wear parts must be replaced more often than in clear liquid pumps, special design attention for ease of replacement of these parts is justified.

The designer's choice is between hard materials that resist wear and resilient materials that may accommodate the particles of solids. This choice is often limited by other factors, such as temperature and chemical compatibilities. Hard materials used include

- Tungsten carbide
- Ceramics—chrome oxide and aluminum oxide
- Hardenable stainless steel
- Cobalt-nickel alloys

Some common soft materials are

- Synthetic elastomers
- Softer metals (sacrificial)

Unlike centrifugal pumps, positive displacement pumps are more prone to abrasive than erosive wear. This is due to the much lower relative velocities between the liquid and the pump parts. Centrifugal pumps must generate high relative velocities, typically greater than 100 ft/s (30 m/s), for the dynamic specific energy that is converted to discharge pressure. Positive displacement pump velocities are kept low, limited usually by the settling velocity of the slurry, which is on the order of less than 10 ft/s (3 m/s).

There are four types of positive displacement solids-handling pumps:

1. Reciprocating piston pumps
2. Reciprocating plunger pumps
3. Diaphragm pumps
4. Hydraulic displacement pumps

Valves Common to all types are suction and discharge check valves, the purpose of which is to allow the solids-laden liquid to flow into the pump from the suction line and into the discharge line from the pumping chamber while preventing backflow. The features of slurry valves include

1. Large areas for low flow velocity
2. Smooth, unobstructed passages to avoid trapping and build-up of solids
3. Special designs to enhance sealing against pressure in the closed position
4. Special materials to minimize wear

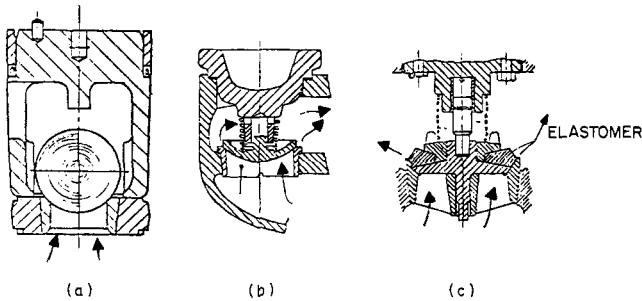


FIGURE 3A through C Three types of slurry valves: (a) free-floating ball valve, (b) spring-loaded spherical (Rollo) valve, (c) spring-loaded elastomer-seal (mud) valve

Figure 3 illustrates some valves developed particularly for solids-handling pumps. Which valve is selected for a particular application depends on the abrasiveness, viscosity, temperature, solids size, and uniformity of the slurry. It is important that the valve lift be sufficient to pass the largest particles of solids expected. Adequate sealing in the closed position is necessary to avoid excessive back leakage, which detracts from the volumetric efficiency of the pump.

Valve materials must meet temperature, corrosion, abrasion, and mechanical strength requirements. Materials for valves and seats range from cast iron and bronze through hardenable stainless steels to solid tungsten carbide. The cost of replacement or refurbishment over the expected life, including labor and downtime, should be balanced against first cost.

Packing Protection A distinguishing feature of solids-handling displacement pumps is the means used to avoid or minimize packing wear. Five different approaches are

1. Use of conventional designs with special materials and operation at less demanding conditions, such as reduced speeds
2. Provision of a clean nonslurry environment for the packing rubbing surfaces
3. Separation of the slurry from the displacement element
4. Separation of the slurry from the pumping element
5. Total isolation of the slurry from the packing

CONVENTIONAL DESIGN WITH MODIFIED OPERATION Piston pumps usually fall into this category. By using larger pumps at speeds that are lower than those normally found with clear fluids, velocity-sensitive wear rates are reduced. *Special piston rings* (Figure 4) are used

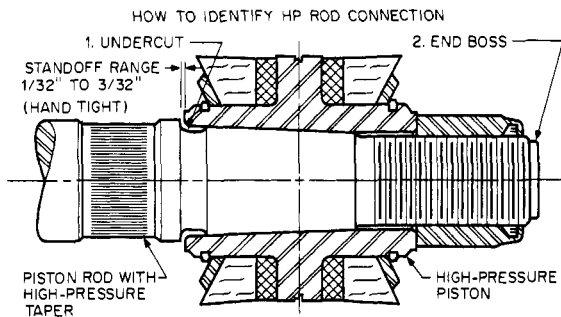


FIGURE 4 Double-acting mud pump piston. Standoff range varies between 0.030 to 0.090 in (0.75 to 2.25 mm).

that tend to resist abrasion because they are made of zero clearance elastomeric materials and designed to scrape clean the cylinder wall. Such piston packing is a normal wear item and must be replaced at regular intervals.

CLEAN NONSLURRY ENVIRONMENT This is accomplished by the injection of clear liquid (Figure 5). Such injection may be either *continuous* or *synchronized* with the stroke of the plunger or piston. When effectively accomplished, flush injection presents essentially clear pumpage to the packing so it attains normal clear liquid packing life. Flush liquid that enters the pumping chamber dilutes the pumpage and decreases the suction volumetric efficiency.

Clear, compatible flush liquid may be scarce or expensive. Therefore special stuffing box plus flush injection systems have been developed. Required flush rates are a function of pump type and size and system employed. Satisfactory operation has been attained with flush rates of about 2 to 5% of pumped flow and even less for very large pumps. If other than a positive displacement flush pump is used, flush flow will increase as packing and throat bushings wear.

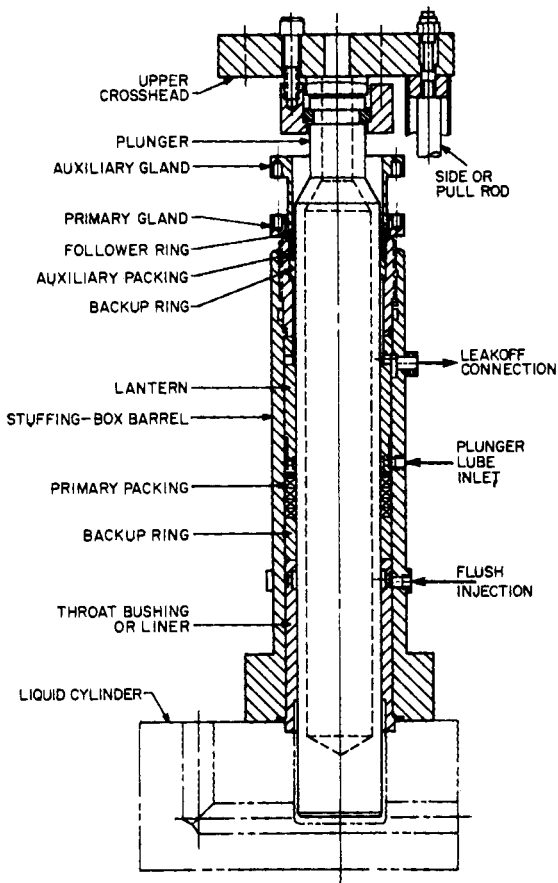


FIGURE 5 Typical flush injection

Flushing prevents incursion of slurry particles by providing a liquid velocity counter to the unwanted slurry flow in clearances and by washing solids particles from the plunger or rod surfaces as they move toward the packing. Injection systems may be as simple as a constant-pressure liquid fed to a point between the packing and the throat bushing or as complex as a sophisticated synchronized injection system involving an additional positive displacement pump driven directly from the slurry pump drive train and independently furnishing a definite quantity of flush fluid to each plunger or piston during each pump stroke (Figure 6).

Combinations of positive, independent, synchronized pumped injection, and auxiliary buffer liquids have been proposed for very special situations.

SEPARATION OF SLURRY FROM DISPLACEMENT ELEMENT One means of reducing the concentration and temperature of slurry at the pump packing is the use of a liquid, or surge, leg (Figure 7). The displacement element—plunger or piston—is separated from the suction and discharge valves by a length of pipe that is filled with clear liquid. This liquid leg communicates with the valve chambers such that, as the liquid ebbs and flows in response to the motion of the displacement element, it acts as a liquid piston, inducing and then expelling slurry through the suction and discharge valves. Make-up liquid is injected into the surge leg, preferably through a throat bushing isolating the displacement chamber from the packing so there is theoretical flow away from the stuffing box. The surge leg thus offers the packing some degree of protection from the solids and from thermal effects.

Surge legs are, however, subject to the complete span of suction-to-discharge pressure change on each stroke. Therefore they must be designed for endurance fatigue loading. When high-temperature isolation is involved, thermal expansion and contraction must also be considered. Although surge leg pumps have been demonstrated to increase piston or plunger packing life, they do not totally prevent solids migration from the pumping

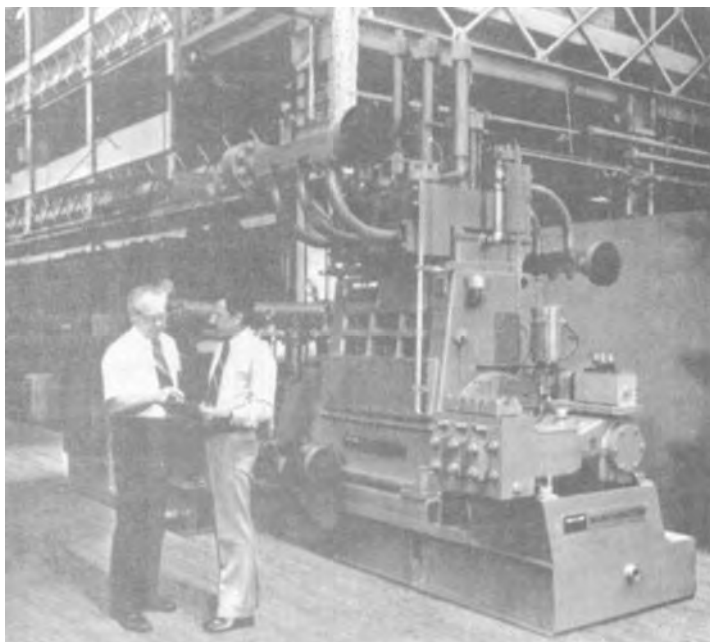
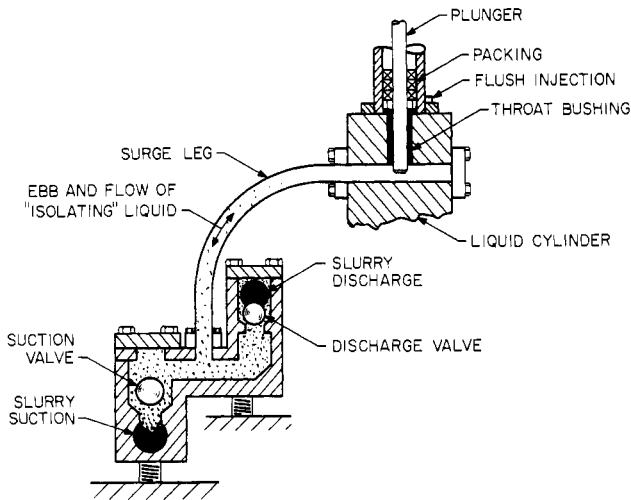
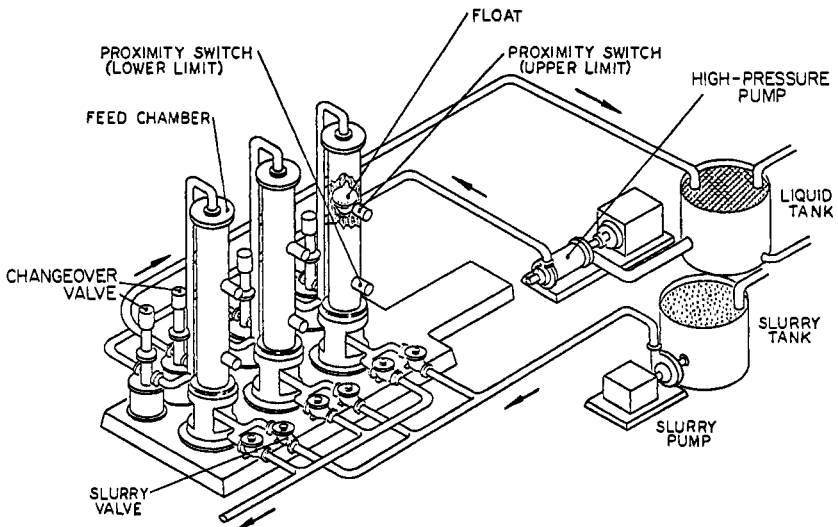


FIGURE 6 Vertical single-acting triplex plunger pump with synchronized flush liquid injection (Flowserve Corporation)

**FIGURE 7** Surge leg plunger pump

chamber. Therefore special construction materials, flush injection, and reduced speeds are still appropriate.

SEPARATION OF SLURRY FROM PUMPING ELEMENT A special case of the liquid leg pump is the large, low-cycle chamber feeder (Figure 8). Although these units may or may not properly qualify as positive displacement pumps (because the motivating liquid may be delivered by a centrifugal pump), they do nevertheless have much in common with surge leg pumps.

**FIGURE 8** Large, low-cycle chamber feeders separate the high-pressure pumping element from the slurry (Hitachi)

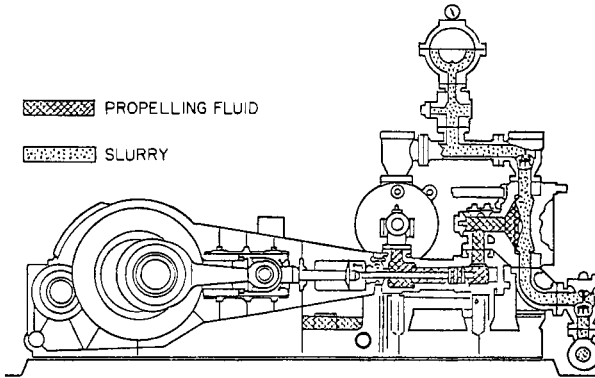


FIGURE 9 Diaphragm pump isolates slurry from packing (Geho)

The motivating liquid is pumped into a chamber and displaces a charge of slurry through suitable check valves. The chamber may have a moving separation barrier between the motivating and motivated liquids. When the slurry is expelled, controls cause valves to reverse the flow, filling the chamber again with slurry while voiding it of the clear liquid. By employing two or more chambers, a virtually continuous flow of slurry is obtained. Advantages of this scheme are the extra-low frequency of check valve operation, low slurry velocities, and high capacities. Centrifugal, positive displacement, or even compressed gas pumps can be used for motivation. Because the motivation pump is independent of the slurry system, it can operate at normal speed.

Two disadvantages of chamber feeders are their dependence on controls and switching valves in addition to the slurry valves and the fact that the systems are large.

TOTAL ISOLATION Many types of diaphragm pumps that completely separate the slurry from the running gear of the pump have been designed (Figure 9). Because the separating diaphragm is wetted on one side by the slurry, it is vulnerable to the mechanical, chemical, and thermal abuse that might otherwise apply to pistons, plungers, packing, and so on. Of course, the diaphragm is also subjected to full reversal loads and must be designed for fatigue life. Both elastomeric and metallic diaphragms have been used.

The severity of diaphragm failure problems has been alleviated somewhat in some designs that employ double diaphragms with intervening barrier liquids. The thesis is that there is a much reduced probability of both diaphragms failing before detection and repair.

DIAGNOSTIC SYSTEMS

Diagnostic monitoring of vibration, pressure pulsations, speed, flow rate, temperatures, and pressures may be applied to these pumps as well as other large machines. Such monitoring is relevant to slurry pumps because of the necessity to replace worn parts more frequently than is the case with clear-liquid pumps. Designers and users of displacement pumps for handling solids should consider using available monitoring equipment as an aid to maintenance and as a means to ensure the safety of personnel by warning of imminent failure. Vibration signatures are readily obtained from sensors placed in critical locations on the pump, and this information can be valuable in assessing the operating integrity of the pump. To this should be added valve wear detection, as would be indicated by a difference between the expected flow rate at the speed involved and the actual flow rate. In addition, proximity probes and vibration sensors could be used to monitor wear of plungers, piston rods, bearings, and crossheads.

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PUMP SEALING

SECTION 5.1

CENTRIFUGAL PUMP PACKING

JAMES P. NETZEL

Packing is used in the stuffing box of a centrifugal pump to control the leakage of the pumped out liquid, or the leakage of air in, where the shaft passes through the casing. This basic form of a seal can be applied in light- to medium-duty services and to those liquids that prove difficult for mechanical seals.

THE DESIGN OF PACKING RINGS

Packing may be referred to as compression, automatic, or floating. Each term describes the type of operation in which the packing will be used.

Automatic and floating packings require no gland adjustments in controlling leakage. Automatic packings are confined to a given space and are activated by the operating pressure. Automatic packing rings are designed in the form of V rings, U cups, and O rings. Floating packing includes piston rings and segmental rings that may be energized by a spring. These types of packing are commonly used in reciprocating applications.

Compression packing is most commonly used on rotating equipment. The seal is formed by the packing being squeezed between the inboard end of the stuffing box and the gland (see Figure 1a). A static seal is formed at the ends of the packing ring and at the inside diameter of the stuffing box. The dynamic seal is formed between the packing and shaft or shaft sleeve. Under a load, the packing deforms down against the shaft, controlling leakage. Some leakage along the shaft is necessary to cool and lubricate the packing. The amount of leakage will depend on the materials of construction for the packing, the operating conditions of the application, and the condition of the equipment.

Packing must be able to withstand equipment variables (see Figure 1b). The design of the packing ring and the materials of construction must be resilient to follow shaft runout and misalignments, as well as to compensate for thermal growth of the equipment without an appreciable increase in leakage.

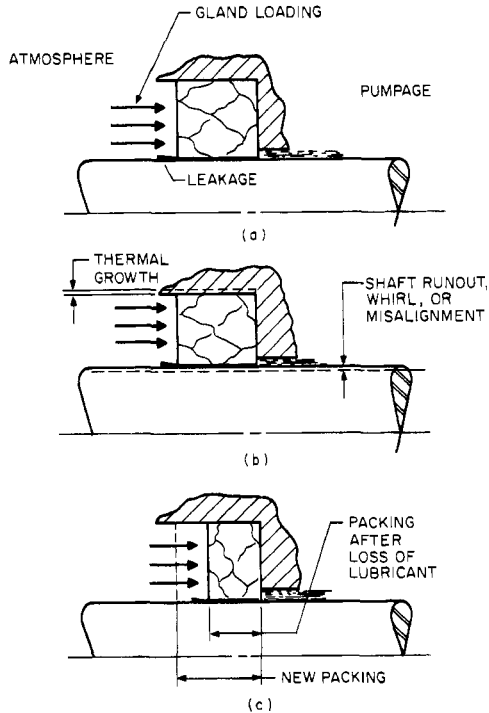


FIGURE 1 Compression packing; (a) new packing installation, (b) installation variables, (c) installation after many adjustments

As rotating equipment is operated, the load on the packing must be adjusted to control leakage (see Figure 1c). Care should be taken not to overtighten the packing. Most compression packings have a lubricant designed into them to prevent overheating of the packing or scoring of the shaft. Repeated adjustments will drive some of the lubricant from the packing, which will result in reduced operating time.

Compression packings are made of twisted, braided, woven, or wrapped elements formed into square or round cross-sections or other configurations. Square cross-sections are more common for rotating equipment.

A selection of the proper materials for the packing must include the chemical resistance to the product being sealed as well as the temperature, pressure, and shaft speed. Complete lists of the construction materials, packing lubricants and binders are given in Tables 1 and 2.

OPERATING FUNDAMENTALS

The Size and Number of Packing Rings The number of packing rings may vary, depending on the objective of the sealing system or the requirements of the rotating equipment. The most common packing arrangement for rotating equipment is illustrated in Figure 2. Three rings of packing are used to seal the process liquid from the packing lubricant. Two rings between the lantern and gland are used to restrict the leakage of the lubricant to the atmosphere. The size of the packing depends on the size of the equipment. Typically, for rotating shafts, the standard square size packings shown in Table 3 may be considered.

TABLE 1 Common materials of construction for packing

Fibers				Metals
Mineral	Animal	Vegetable	Synthetic	Lead
Metal	Wool	Flax	Nylon	Copper
Graphite	Hair	Ramie	Rayon	Brass
	Leather	Jute	PTFE	P-bronze
		Cotton	Carbon	Aluminum
		Paper	Aramid	Iron
			Polyamide	Stainless Steel
				Nickel
				Monel
				Inconel
				Zinc

TABLE 2 Common lubricants and binders for packing

Lubricants		Dry Lubricants	Binders
Mineral	Animal	Graphite	Grease
Lube Oil	Tallow	Moly	Waxes
Paraffin	Glycerol	Mica	Elastomers
Petrolatum	Beeswax	Talc	TFE
Waxes	Lard Oil	Teflon	Other Resins
Greases	Fish Oil	Carbon	
	Soap	Tungsten Disulfide	
Vegetable	Synthetic		
Caster Oil	Oils		
Palm Oil	Waxes		
Cottonseed Oil	Fluorolubes		
Linseed Oil	Silicones		
Carnauba Wax			

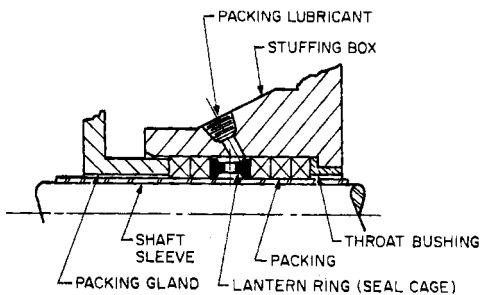
**FIGURE 2** Common packing arrangement

TABLE 3 Packing sizes for rotating shafts

Shaft (or sleeve) Diameter, in (mm)	Packing size, in (mm)
$\frac{5}{8}$ to $1\frac{1}{8}$ (15 to 30)	$\frac{1}{4}$ (6.3)
$1\frac{1}{8}$ to $1\frac{7}{8}$ (30 to 50)	$\frac{5}{16}$ (8)
$1\frac{7}{8}$ to 3 (50 to 75)	$\frac{3}{8}$ (10)
3 to $4\frac{3}{4}$ (75 to 120)	$\frac{1}{2}$ (12.5)
$4\frac{3}{4}$ to 12 (120 to 305)	$\frac{5}{8}$ (16)

The packing size for an existing piece of equipment can be found by using the formula:

$$\text{Packing size} = \frac{\text{box ID} - \text{shaft or sleeve OD}}{2}$$

For packing to operate properly, the finish on the shaft sleeve must be at least $16\text{ }\mu\text{in}$ ($0.4\text{ }\mu\text{m}$) centerline average (CLA) and the finish in the bore should be $63\text{ }\mu\text{in}$ ($1.65\text{ }\mu\text{m}$) CLA. The sleeve *must* be harder than the packing and chemically resistant to the liquid being sealed. If the sleeve has a coated material for a hard-wear surface, the sleeve must also have good thermal shock resistance.

Lantern Rings (Seal Cages) When an application requires that a lubricant be introduced to the packing, a lantern ring is used to distribute the flow (refer to Figure 2). This ring is used at or near the center of the packing installation. For ease of assembly, most lantern rings are axially split. The construction materials range from metal to TFE (tetrafluoroethylene). TFE lantern rings are usually filled with glass or with glass and molybdenum disulfide. They are inherently self-lubricating and will not score the shaft. A throat bushing at the bottom of the stuffing box can be used to provide a closer clearance with the shaft to prevent packing extrusion.

Stuffing Box Gland Plates All mechanical packings are mechanically loaded in the axial direction by the stuffing box gland (refer to Figure 2). In cases where leakage of the process liquid is dangerous or can vaporize and create a hazard to operating personnel, a smothering gland is used to introduce a neutral liquid at lower temperatures (see Figure 3). A sufficient quantity of quenching liquid should be used to eliminate the danger from the

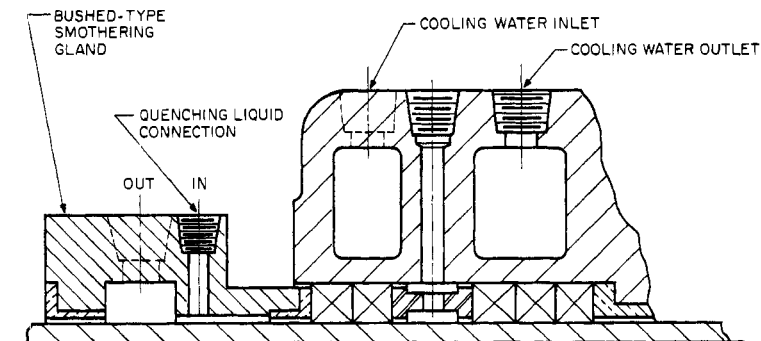


FIGURE 3 A smothering gland and water-cooled stuffing box

TABLE 4 Leakage to prevent packing burning and sleeve scoring

Pressure lb/in ² (bar)	Leakage, drops/min	cc/min
0–60 (0–4.0)	60	4
61–100 (4.1–6.8)		190
101–250 (6.9–17)		470

liquid being pumped. The neutral liquid circulated in the gland mixes with the leakage and carries it to a safe place for disposal. Close clearances in the gland control the leakage of the combined liquids to the atmosphere. This quench can also be used to protect the packing from any wear through abrasion, because the leakage cannot vaporize and leave behind abrasive crystals.

Glands are usually made of bronze, but cast iron or steel can be used for all-iron pumps. When iron or steel glands are used, they are normally bushed with a nonsparking material like bronze or nonmetals like TFE.

Leakage and Power Consumption The basic operating parameters for compression packing are the PV (pressure $\times$ velocity) factor and the projected bearing area of the assembly. Together they determine the rate of heat generation for the system. Some leakage of the product being sealed or of the packing lubricant is necessary to keep the packing from burning up or scoring the shaft sleeve. The minimum values for leakage at different packing pressures are given in Table 4. Flexible graphite and carbon filament compression packings can be used with reduced leakage rates, or in dry, gas-tight pump stuffing boxes, as the developed heat is dissipated through the packing and pump housing.

The exact pressure P between the shaft sleeve and the packing is a function of the pressure distribution over the length of the packing and the axial loading from the gland. For ease of calculation when determining the PV value, the gage pressure of the liquid at the packing is multiplied by $\pi \times$ the packing ID $\times$ rpm. The heat generation at the packing can then be estimated as

$$Q = \frac{f\pi^2 PND^2 L}{CJ}$$

where Q = heat generated, Btu/min (W)

f = coefficient of friction

P = liquid pressure at packing, lb/in² gage (bar)

N = shaft speed, rpm

D = sleeve OD or packing ID, in (m)

L = sleeve length covered by packing, in (m)

C = 12 (60 for SI units)

J = mechanical equivalent of heat = 778 ft $\cdot$ lb/Btu (1 N $\cdot$ m/s $\cdot$ W)

The coefficient of friction for various packings at a pressure of 100 lb/in² (6.8 bar) is given in Table 5. The heat generated by the packing must be removed by the leakage through the packing.

TABLE 5 Typically coefficients of friction

Material	f
Plain cotton	0.22
TFE impregnated fiber	0.17
Grease-lube fiber	0.10
Flexible graphite	0.05



FIGURE 4 Graphite acrylic packing in continuous form (John Crane Inc.)



FIGURE 5 Interlaced construction packing (John Crane Inc.)

APPLICATION INFORMATION AND SEALING ARRANGEMENTS

Materials of Construction Basically, stuffing box packing is a pressure breakdown device. In order for a packed stuffing box to operate properly, the correct packing must be applied and the appropriate design features must be included in the system. Numerous types of packing materials are available, each is suited to a particular service. These may be grouped into three categories:

- 1. Interlaced (braided) packing** Materials used for interlaced (braided) packing include cotton, TFE filament, Aramid fiber, aromatic polyamides, graphite/carbon yarn, and flexible graphite. All of these materials, with the exception of flexible graphite, can be impregnated with various lubricants. An example of graphited acrylic packing is shown in Figure 4. With the exception of flexible graphite, all the materials are of an interlaced construction for greater flexibility (see Figure 5). Packing made with this type of construction remains intact even if individual yarns wear away at the inside diameter of the packing.
- 2. Flexible graphite** These types of packing rings were originally available cut from laminated sheet stock that required the rings to be made with an interference fit. Flexible graphite rings are made today from ribbon tape that is easily compressed to the shaft and bore to affect a better seal. Flexible graphite rings are available in split or endless rings (see Figure 6) or as the ribbon tape itself for ease of maintenance and installation. Operating limits for common packings are found in Table 6.
- 3. Metallic packing** The basic materials of construction are babbitt, aluminum, and copper in either wire or foil form. Metallic packing rings have flexible cores of materials such as twisted glass fiber. The packing is impregnated with graphite grease and/or oil lubricants (see Figures 7 and 8). Babbitt is used in water and oil services at



FIGURE 6 Flexible graphite packing rings (John Crane, Inc.)

TABLE 6 Service limitations of common packing materials^a

Packing Material	Pressure (max.) ^b lb/in ² gage (bar)	PV rating (max.) ^c lb/in ² gage · fpm (bar · m/s)	Temp. (max) ^d °F (°C)	pH Range	Comments
Cotton	100 (6.8)		150 (65.6)	5–7	Nonabrasive material; for cold water and dilute salt solutions
Flax/ramie	100 (6.8)	188,000 (65.8)	150 (65.6)	5–7	High, wet strength and excellent resistance to fungi and rotting; for cold water and dilute salt solutions
Plastic	100 (6.8) 250 (17)	188,000 (65.8) 471,000 (165)	600 (315.5) 150 (65.6)	4–8	Excellent sealing qualities, reacts well to gland adjustments, can extrude at higher pressure if not backed up by braided or metallic packing
Graphited acrylic	250 (17)	471,000 (165)	350 (175)	4–8	For mild chemicals and solvents
Acrylic TFE- impregnated	250 (17)	471,000 (165)	350 (175)	2–10	For mild chemicals and solvents
Babbitt (lead)	250 (17)	471,000 (165)	450 (232.2)	2–10	Shaft sleeve must have a Brinell hardness of 500 or more; for hot oils and boiler feed water
Aluminum or copper	250 (17)	471,000 (165)	750 (398.8)	3–10	Shaft sleeve must have a Brinell hardness of 500 or more; for hot oils and boiler feed water
TFE filament	250 (17)	471,000 (165)	500 (260)	0–14	For corrosive liquids and food service; usually requires slightly higher break-in leakage
Aramid fiber	250 (17)	471,000 (165)	500 (260)	3–11	Strong resilient packing; maximum speed 1,900 fpm (9.6 m/s); good in abrasives and chemicals
Graphite/carbon filament ^e	250 (17)	471,000 (165)	750 (398.8)	0–14	For corrosive liquids and high- temperature applications
Flexible Graphite ^f	250 (17)	471,000 (165)	1000 (540)	0–14	Excellent conductor of heat from the sealing surfaces; operates with minimum leakage; excellent radiation resistance

(a) Continuous lubrication introduced at the lantern ring. This table is only a guide. Consult the packing manufacturer with complete operating conditions for exact recommendations.

(b) Pressure relates to the operating pressure at the stuffing box.

(c) PV data based on a 2 in (5.08 cm) shaft at 1,750 and 3,600 rpm.

(d) Temperature is the product temperature.

(e) Functional temperature. Graphite can be used up to 3000°F (1650°C) in nonoxidizing atmospheres.

(f) Functional temperature. Flexible graphite can be used to 5300°F (2970°C) in nonoxidizing atmospheres.

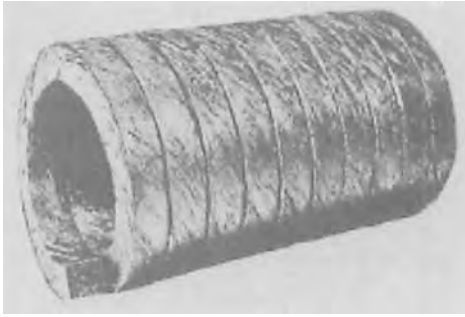


FIGURE 7 Metallic packing in spiral form (John Crane Inc.)

temperatures up to 450°F (229°C) and pressures up to 250 lb/in² (17 bar). Copper foil is used with water and low sulfur oils. Aluminum is used in oil service and heat transfer fluids. Both copper and aluminum have a temperature range up to 1000°F (537°C) and pressures of 250 lb/in² (17 bar). Babbitt is not suitable for running against brass or bronze shaft sleeves, and where copper or aluminum is used, the sleeves should be 550 Brinell (55–60 Rockwell C) or harder.

Additional service limitations for common packing materials can be found in Table 6.

ENVIRONMENTAL LIMITATIONS

Pressure Every pumping application results in either positive or negative pressure at the throat of the pump stuffing box. A positive pressure will force the liquid pumped through the packing to the atmosphere side of the pump. Higher pressure will result in greater leakage from the pump. This results in excessive tightening of the gland, which



FIGURE 8 Metallic packing in ring form (John Crane Inc.)

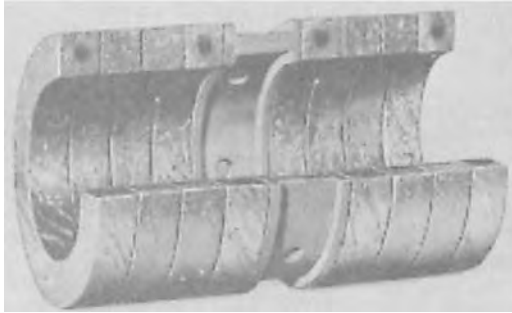


FIGURE 9 Combination of hard and soft packing (John Crane Inc.)

causes accelerated wear of the shaft or shaft sleeve and packing. For pressures at the stuffing box greater than 75 lb/in^2 (5.1 bar), some means of throttling the pressure should be considered. A combination of hard and soft rings die-formed to the exact stuffing box bore and sleeve dimensions can be used (see Figure 9). Harder rings at the inboard end of the box and at the lantern ring and gland break down the pressure and prevent the extrusion of the packing.

If the packing itself cannot be used to break down the pressure, then a throttle bushing must be used (see Figure 10). This is a typical arrangement for a vertical turbine pump where the stuffing box is subject to the discharge pressure of the pump. Here the throttle bushing is used to bring the liquid to almost suction pressure, and most of the leakage through the bushing is bled back to the suction. When the suction pressure is less than atmospheric, as in condensate pump service, an orifice is used in the piping back to the suction in order to maintain pressure above atmospheric pressure in the stuffing box. This prevents air leakage into the pump and excessive flow and wear at the bushing.

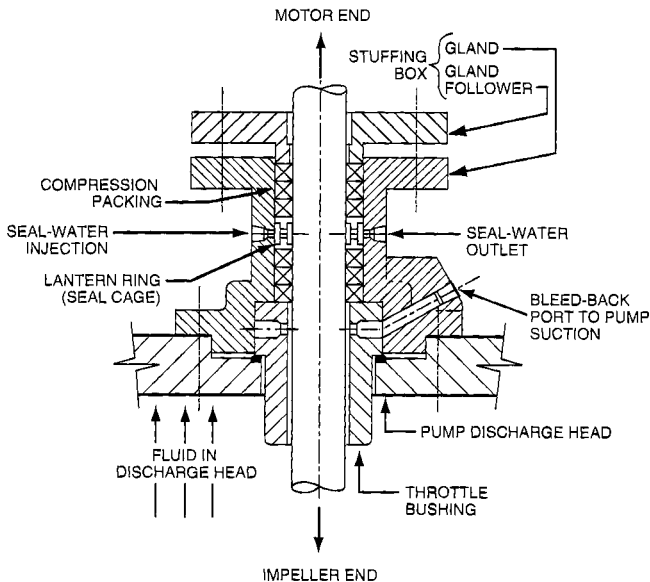


FIGURE 10 Compression packing with throttle bushing for pressure breakdown

When a pump is fitted with a bypass line from the discharge, a valve can also be used to reduce the pressure at the stuffing box. This is another method for reducing pressure for the benefit of the installation.

Air Leakage When the pressure at the stuffing box is atmospheric or just below during normal operation, a bypass from the pump casing discharge through an orifice can be used to inject liquid into the lantern ring. The sealing liquid will flow partly into the pump and partly out to the atmosphere, thereby preventing air from entering at the stuffing box. This arrangement is commonly used to handle clean, cool water. In some pumps, these connections are arranged so that liquid can be introduced into the lantern ring through internally drilled passages.

When the negative pressure is very low, such as when the suction lift is in excess of 15 ft (4.6 m), an independent injection of 10 to 25 lb/in² (4.8 to 11.7 bar) higher than the atmospheric pressure must be used on the pump. Otherwise, priming may be difficult. Hot well pumps or condensate pumps operate with as much as 28 in (0.7 m) of vacuum, and air leakage into the pump would occur even on standby service. Here a continuous injection of clean, cool water is required, or alternatively, a cross connection of sealing water to another operating pump will provide sealing pressure as long as one pump is operating. A lantern ring is provided for this, as shown in Figure 10.

Temperature The control of temperature at the stuffing box is an important factor in promoting the life of the packing. Even though packings are rated for high product temperatures, cooling in most cases is desirable. The heat developed at the packing must also be removed. The rules of thumb are as follows:

- For light service conditions with pressures at 15 lb/in² (1 bar) and temperatures at 200°F (90°C), cooling is desirable.
- For medium service conditions with pressures at 50 lb/in² (3.4 bar) and temperatures at 250°F (118°C), lantern ring cooling is desirable, such as one gpm (3.78 l/min) at 5 lb/in² (0.34 bar) above process pressure.
- For high service conditions with pressures at 100 lb/in² (6.8 bar) and temperatures at 300°F (131°C), lantern ring cooling is desirable, such as one gpm (3.78 l/min) at 5 lb/in² (0.34 bar) above process pressure plus a water-cooled stuffing box (refer to Figure 3).

When cooling water is required, it is circulated through the stuffing box at the lantern ring. For horizontal-shaft pumps, the inlet should be at the bottom, with the outlet at the top of the stuffing box. Some of the coolant may flow to the process liquid and some to the atmosphere. The outboard packing rings seal only the cool liquid.

If the product to be sealed will solidify, then the packing box will have to be heated before the pump is started. This can be accomplished by steam or electric tracing of the pump stuffing box.

Abrasives Liquids that contain abrasives in the form of *suspended solids* such as sand and dirt will shorten the life of the packing. Particles will imbed themselves in the packing and will begin to wear the shaft or shaft sleeve. Abrasives can be eliminated at the sealing surfaces by injecting a clean liquid into the lantern ring. The injection may take the form of a bypass line from the pump discharge through a filter or centrifugal separator. Where necessary, the clean injection may also be from an external source.

To keep the abrasives from the packing in horizontal-shaft pumps, the injection can be made directly to the inboard end of the stuffing box through the lantern ring (see Figure 11). A soft rubber gasket between the lantern ring and the box shoulder can be used to limit the flow of clean liquid and the ingress of abrasives.

When separators and filters cannot be used, an injection from an external source must be considered. Two rings of packing are located between the lantern ring and the inboard end of the stuffing box to keep the product dilution to a minimum. External flushing should be injected into the stuffing box at a pressure 10 to 25 lb/in² (1.7 bar) greater than the pressure at the inboard end of the box from the liquid being pumped. A regulating

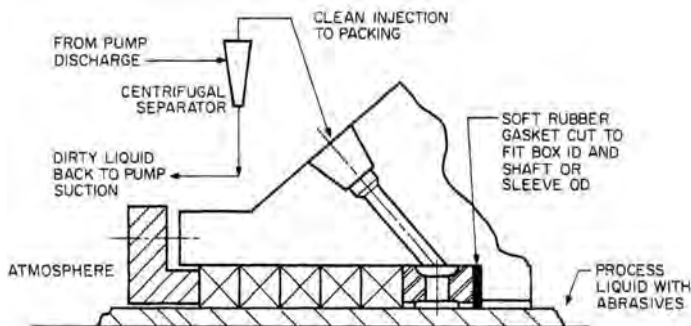


FIGURE 11 A clean injection through a centrifugal separator to keep abrasives out of the stuffing box

valve, illustrated in Figure 12, can be used to control the pressure and flow to the packing installation. Flow to the packing can be regulated to ensure the best operating environment for this seal, while conserving water used for injection.

Dissolved solids in the liquid being pumped can also create a wear problem. Here an increase or decrease in stuffing box temperatures may be necessary to keep solids in solution.

INSTALLING CONTINUOUS COIL PACKING

To install continuous coil packing, perform the following steps:

1. Loosen and remove the gland from the stuffing box.
2. Using a packing puller, begin to remove the old packing rings.
3. Remove the split lantern ring (if present) and then continue removing the packing with the puller.

Packing or Single Seal Flush



FIGURE 12 A regulation valve that controls and monitors the flow of water to the packing (Safematic)

TABLE 7 Packing troubles, causes, and cures

Trouble	Cause	Cure
No liquid delivered by pump	Lack of prime (packing loose or defective, to allowing air leak into suction)	Tighten or replace packing and prime pump.
Not enough liquid delivered by pump	Air leaking into stuffing box	Check for leakage through stuffing box while operating. If no leakage occurs after after reasonable gland adjustment, new packing may be needed. or Lantern ring may be clogged or displaced and may need centering in line with sealing liquid connection. or Sealing liquid line may be clogged. or Shaft or shaft sleeve beneath packing may be badly scored, allowing air to be sucked into pump.
	Defective packing	Replace packing and check the smoothness of the shaft or shaft sleeve.
Not enough pump pressure	Defective packing	As per preceding
Pump works for a while and then quits	Air leaks into stuffing box	As per preceding
Pump takes too much power	Packing too tight	Release gland pressure and retighten reasonably. Keep leakage flowing. If none, check packing, sleeve, or shaft.
Pump leaks excessively at stuffing	Defective packing	Replace worn packing or replace packing damaged by lack of lubrication.
	Wrong type of packing	Replace packing not properly installed or run in. Replace improper packing with correct grade for liquid being handled.
	Shaft or shaft sleeve scored	Put in lathe and machine-true and smooth or replace. Recheck dimensions for correct packing size.
Stuffing box overheats	Packing too tight	Release gland pressure and retighten.
	Packing not lubricated	Release gland pressure and replace all packing if any burnt or damaged.
	Wrong grade of packing	Check with pump or packing manufacturer for correct grade.
	Insufficient cooling water to jacket	Check for open supply line valve or clogged line.
	Stuffing box improperly packed	Repack.
Packing wears too fast	Shaft or shaft sleeve worn or scored	Remachine or replace.
	Insufficient or no lubrication	Repack, making sure packing is loose enough to allow some leakage.
	Packing packed improperly	Repack properly, making sure all old packing is removed and the box is clean.
	Wrong grade of packing	Check with pump or packing manufacturer.
	Pulsating pressure on external seal liquid line	Remove cause of pulsation.

4. After the packing has been removed, check the sleeve for scoring and nicks. If the shaft sleeve or shaft cannot be cleaned up, it must be replaced. Check the size of the stuffing box bore and the shaft sleeve or shaft diameter to determine which size packing should be used.
5. After the size of the packing has been determined, wrap the packing tightly around a mandrel, which should be the same size as the pump shaft or sleeve. The number of coils should be sufficient to fill the stuffing box. Cut the packing along one side to form the individual rings.
6. Before beginning the assembly of any packing material, be sure to read all the instructions from the manufacturer. Assemble the split packing rings on the pump. Each ring should be sealed individually with the split ends staggered 90° and the gland tightened to seal and fully compress the ring. Be sure the lantern ring is reinstalled correctly at the flush connection. Then back off the gland and retighten it, but only finger-tight. The exception to this procedure is that TFE packing should be installed one ring at a time, but not seated because TFE packings have high thermal expansion.
7. Allow excess leakage during break-in to avoid the possibility of rapid expansion of the packing, which could score the shaft sleeve or shaft so that leakage could not be controlled.
8. Leakage should be generous upon startup. If the packing begins to overheat at startup, stop the pump and loosen the packing until leakage is obtained. Restart only if the packing is leaking.

CAUSES FOR A SHORT PACKING LIFE

In order for packing to operate properly, the equipment must be in good condition. Shafts should be checked for runout and eccentricity to be sure they are within the manufacturer's recommended tolerances. Surfaces in contact with the packing should be finished to the correct smoothness and tolerance. Table 7 lists common troubles that affect the packing life. Causes and possible cures are also given.

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SECTION 5.2

CENTRIFUGAL PUMP MECHANICAL SEALS

MICHAEL B. HUEBNER

INTRODUCTION

Mechanical seals can be simply described as devices to minimize or prevent process fluids or gases from leaking out of the pressure casing of a pump along a rotating shaft. In practice, though, a mechanical seal is a highly engineered device capable of operating under a wide variety of fluids, pressures, temperatures, and speeds. Over the decades of their development, mechanical seals have evolved from simple flat rings to very sophisticated sealing surfaces. Through this evolution, they still continue to serve the same purpose: to allow pumps and rotating equipment to operate reliably and efficiently while minimizing process leakage to atmosphere.

Historically, there have been a number of concepts applied to sealing a rotating shaft in a stationary housing. These include mechanical packing, labyrinth seals, close clearance bushings, and split carbon rings. All of these techniques have been used successfully and continue to be applied in specific applications. They all share a common limitation though they all attempt to control leakage through an annular clearance. The most common method for controlling leakage is to reduce the annular clearances between the stationary and rotating components to a minimum. This has a practical limit based on shaft motion, thermal expansion, and wear of components. The result is a reduction in leakage but not to the extent required for most modern sealing applications.

To overcome these limitations, mechanical seals move the sealing surfaces to a plane perpendicular to the shaft rotation. In this location, the sealing surfaces allow for radial motion and thermal expansion without affecting the clearance between the faces. The seal faces are mounted so that they float on a thin fluid film and allow for small axial motions of the shaft caused by pump manufacturing tolerance and alignments. In this orientation, the fluid film between the seal faces can be orders of magnitude smaller than in annular seals while minimizing face wear and frictional drag. Current seal technology can result in reliable seal performance for many years without operator adjustment or repair.

DESIGN FUNDAMENTALS

While the configurations and complexity of mechanical seals vary significantly, they all operate with the same fundamental principles. All seals have one face that is affixed to the stationary pump casing and one face that is affixed to and rotating with the pump shaft. The gap between these faces controls the performance of the seal. The stationary face is generally mounted into a seal gland that interfaces with the pump casing. The rotating face is generally mounted to a sleeve that interfaces with the pump shaft. Appropriate adaptive components and gaskets complete the design. The design and material selection of these components help establish the operating limits for a specific design, which can range from a simple low-duty seal (Figure 1) to a more complex, higher duty seal (Figure 2).

Even though all mechanical seals share many of the same components, the individual component design may vary significantly between different seals. This can be a function of the operating window for the seal, the materials of construction, the size of the installation window in the pump, or other application requirements (such as cleaning or sterilization). It can also depend upon factors as diverse as the presence of abrasive particles or allowances for large shaft motions. For these reasons, there will never be a universal seal design suitable for every application. There will always be a wide range of offerings from seal OEMs.

Mechanical seals can generally be divided into two main design types—pusher seals and bellows seals. Pusher seals have one of the seal faces mounted on a flexible gasket that allows the face to move axially (Figure 3). Bellows seals have one of the seal faces mounted in a bellows assembly that provides a pressure boundary while allowing for axial face motion. Bellows designs vary from elastomeric construction to more complex welded metal designs (Figure 4). Each of these designs has advantages for specific applications.

Mechanical seals can also be provided as a component seal or a fully assembled cartridge seal (Figure 5). Component seals consist of a limited number of critical seal components including the seal faces, springs, and adaptive gaskets. This design relies on internal mating dimensions in the pumps or manual installation to establish the seal setting. Installation of a component seal requires careful practices and a clean environment to prevent contamination or damage of the seal faces. Cartridge seals are fully assembled prior to installation. They are installed by bolting the complete cartridge into the pump seal chamber. This ensures the correct seal setting and minimizes the opportunity for damage to occur. The complete cartridge can also be tested for integrity by the seal OEM before shipping. The cartridge configuration has become more popular due to improved reliability and is required in some seal standards.

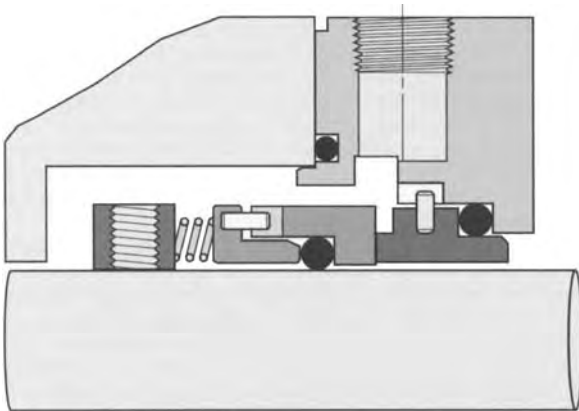


FIGURE 1 Low duty mechanical seal

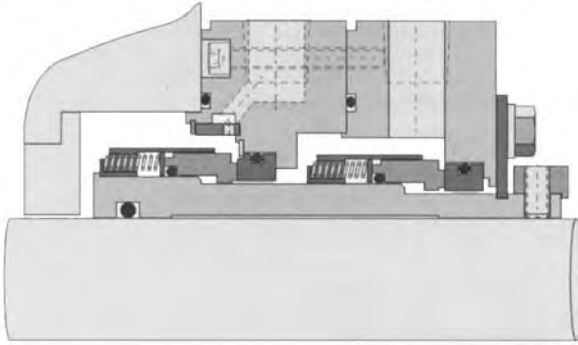


FIGURE 2 General duty mechanical seal

SEAL PERFORMANCE

The faces of a mechanical seal are the most critical design element and largely define the operating window of the seal. The seal designer has great flexibility in the face design to control face deformations and hydraulic loading. These factors control the formation of the fluid film, seal leakage, power requirements, and life of the seal. If the face contact pressure is too high, it will result in too thin of a fluid film, which will create high heat and excessive face wear. If the contact pressure is too low, it will result in a thicker fluid film, which will allow excessive seal leakage.

The primary method for controlling the contact pressure between the faces is balancing the hydraulic closing forces against the hydraulic opening forces. The fluid in the seal chamber exerts an equal pressure on all hydraulically wetted surfaces. Hydraulic closing area is defined as the surfaces that act to increase the face contact pressure. The term "balance ratio" or seal balance is used to describe the ratio of the hydraulic closing area to the face contact area. The seal balance b is defined as:

$$b = a/a_o$$

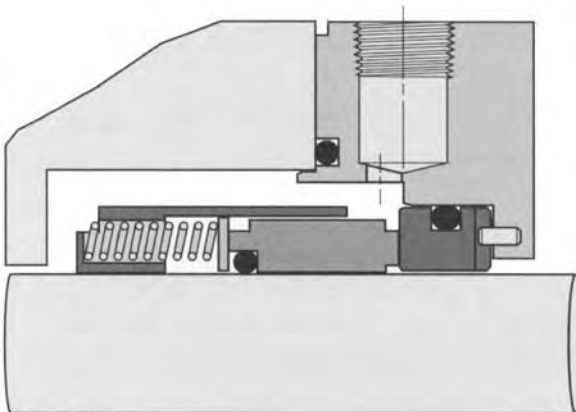


FIGURE 3 Pusher seal

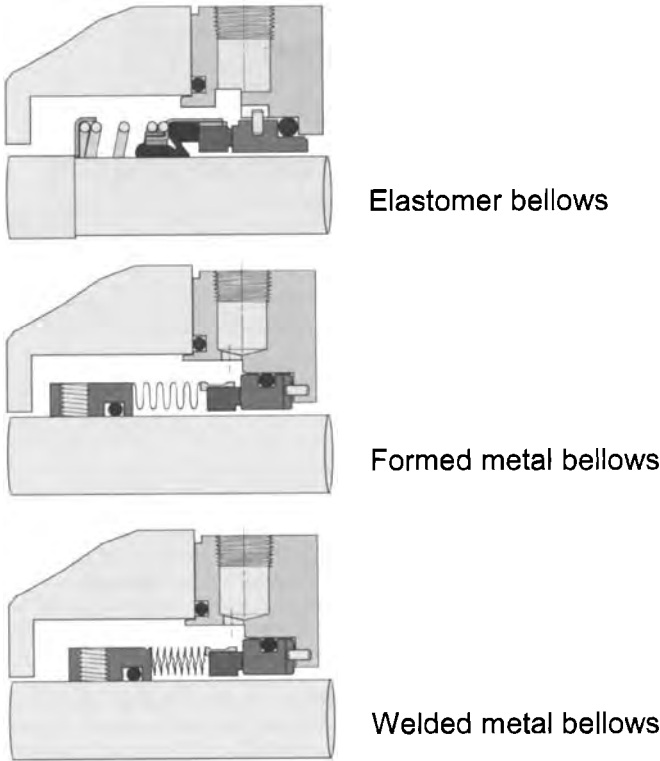


FIGURE 4 Bellows seal—elastomeric, formed, welded metal

where b = balance

a_c = hydraulic closing area, $\text{in}^2(\text{m}^2)$

a_o = hydraulic opening area (or face contact area), $\text{in}^2(\text{m}^2)$

For actual seal designs, the seal balance is defined in terms of the seal face diameters and the seal balance diameter. These are described by the following equations:
OD pressurized seals (Figure 6)

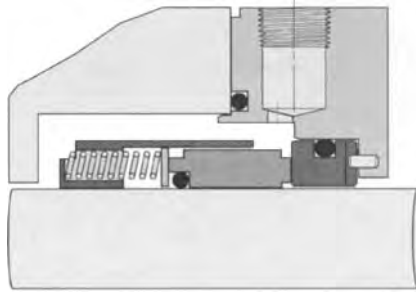
$$b = (\text{OD}^2 - \text{BD}^2) / (\text{OD}^2 / \text{ID}^2)$$

ID pressurized seals (Figure 7)

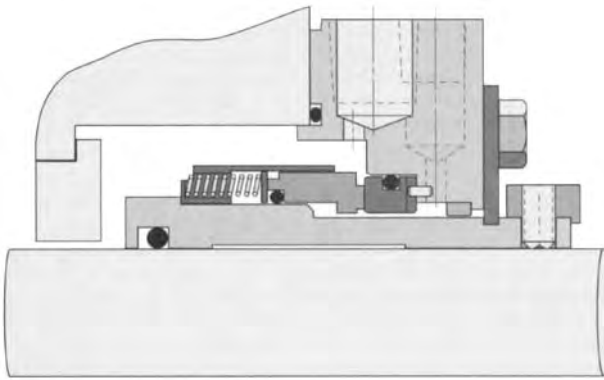
$$b = (\text{BD}^2 - \text{ID}^2) / (\text{OD}^2 / \text{ID}^2)$$

Commonly, contacting liquid seals have balances in the range of 0.65 to 0.85 in operation. Seals with balances less than 1.0 are defined as balanced seals indicating that the seal is designed to hydraulically reduce (or balance) the effects of the seal chamber pressure on the seal faces. Seals with balances of 1.0 or greater are defined as unbalanced seals. Unbalanced seals have a higher face loading and are generally limited to operating pressures less than 10 bar (145 psi). Seal balance can also be expressed as a percentage such as 75% balance.

In addition to the hydraulically generated closing forces, mechanical seals employ a spring device to mechanically load the seal faces. This mechanical loading provides closing



Component Seal



Cartridge Seal

FIGURE 5 Component seal and cartridge seal

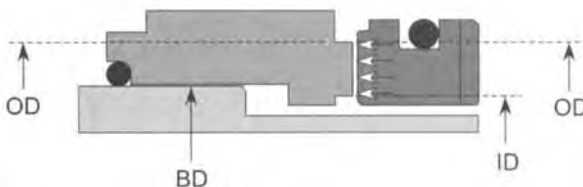
forces during low-pressure operation. It also provides the forces necessary to overcome axial friction forces on the seal face due to O-ring drag and drive friction. Spring pressure exerted on the faces is defined as:

$$P_{sp} = F_{sp} / a_o$$

Where P_{sp} = face contact pressure generated by the springs, psi (N/m²)

F_{sp} = force generated by the springs at working length

a_o = hydraulic opening area (or face contact area), in²(m²)

**FIGURE 6** OD pressurized seal

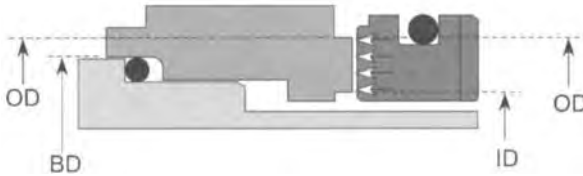


FIGURE 7 ID pressurized seal

In contacting liquid seals, P_{sp} will normally vary between 100 to 240 kPa (15 to 35 psi). For other designs such as dry running seals and gas seals, the value of P_{sp} will generally be approximately 14 to 40 kPa (2 to 6 psi).

The pressure in the fluid film between the seal faces creates an opening force that acts in a direction opposite the hydraulic closing force and the spring force. On a typical single seal, the pressure profile is not constant. It will be equal to the seal chamber pressure at the outer diameter of the seal faces, decreasing to atmospheric pressure at the ID of the seal face. The integration of the pressure across the face area will result in a force, which decreases the contact pressure between the faces. The exact profile of the pressure drop depends upon the flatness (or parallelism) of the seal faces and the state of the fluid film (liquid, gas, or flashing). The integration of the area across the face is described by the pressure gradient factor k and has a numerical value between 0 and 1. For a parallel liquid film, k has a value approximately 0.5. For seal faces which have a converging fluid film (film thicker on the high pressure side), k will be greater than 0.5. For faces with a diverging fluid film (film thinner on the high pressure side), k will be less than 0.5 (Figure 8).

The other factor which affects k is the phase or change of phase of the fluid between the seal faces (Figure 9). A fluid which remains in a liquid state between the seal faces will have a linear pressure drop as it migrates from the high pressure side to the low pressure side. If the fluid film is supported entirely by a gas film, the pressure profile is different due to the compressibility of the gas that results in a larger opening force. A fluid which changes from a liquid phase to vapor phase across the seal face will result in an even larger opening force. This is a normal condition with fluids that will flash under atmospheric conditions such as light hydrocarbons.

The goal of a successful seal design is to create the optimal fluid film thickness by using the seal balance b and the spring pressure P_{sp} to counteract the effects of the pressure drop coefficient k opening the seal faces. The ideal resultant face contact pressure allows for long seal life with minimal leakage and minimum heat generation. The total face contact pressure P_f can be calculated by the following equation:

$$P_f = \Delta P(b - k) + P_{sp} \text{ kPa (psi)}$$

where ΔP = differential pressure across seal faces, kPa (psi)

With the calculation of P_f , other factors can be derived such as PV , starting torque, and power consumption.

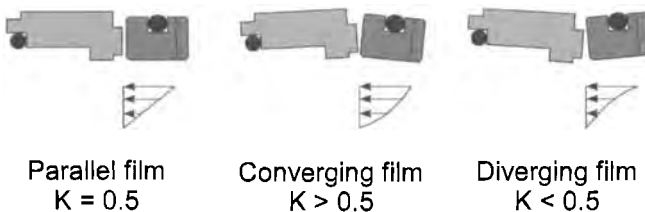


FIGURE 8 Variations in pressure drop coefficient k

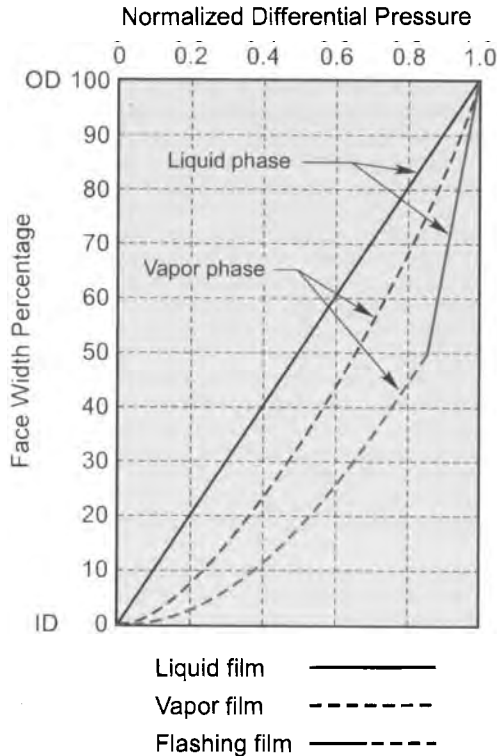


FIGURE 9 Variation in pressure profile with different fluid phases (Flowserve Corporation)

Pressure-Velocity The concept of pressure-velocity has a long history in tribology and material sciences. The product of the pressure times the velocity (or more commonly *PV*) is used to describe a sliding interface between two surfaces. While this is extremely useful in material science, it is less applicable to mechanical seals. Some published data for mechanical seals state *PV* limits considering only the material selection of the seal faces and does not consider the lubricating properties of the fluid film. Other references put *PV* values in terms of seal chamber pressure and not the contact pressure at the seal faces. The conflicting definitions and considerations in *PV* data require the reader to carefully understand the definition used and the considerations made in applying the *PV* limits to a specific seal.

The most accurate method for calculating *PV* is to multiply the total face contact pressure P_f with the velocity at the mean diameter of the seal face V_m .

$$PV = [\Delta P(b - k) + P_{sp}] V_m, \text{ kPa} \times \text{m/s (psi} \times \text{ft/min)}$$

where V_m = velocity at the mean diameter of the seal face, m/s (ft/min)

TABLE 1 Typical values for pressure gradient factors k

Non-flashing fluids	0.50
Flashing fluid	0.70 to 0.75
All vapor of gas	0.66

Seal OEMs may produce *PV* tables for seals that can be of some value in evaluating a seal selection. The user should not extrapolate this data or compare it to *PV* values from other sources.

Overall Coefficient of Friction Two sliding surfaces exhibit resistance to motion, which is classically referred to as friction. This is characterized by the coefficient of friction, which is defined as the ratio of the tangential forces required to create or sustain motion divided by the normal forces between the surfaces. Friction in a nonlubricated interface is a predictable value based on the combination of the mating materials. The coefficient of friction in a mechanical seal face is more complex and less predictable. The resistance to motion is caused not only by classical friction but also by the shearing of the fluid in the interface, which is proportional to the fluid viscosity and inversely proportional to the film thickness. The combination of frictional effects from mechanical contact and fluid shear can be calculated only if all of the variables are known. Even then, it is only possible in sophisticated computer modeling of the seals.

To help provide a more accessible value for resistance to motion, mechanical seals calculations have used the concept of an effective coefficient of friction *f*. The values used for *f* represent the combined effects of all resistances to motion and can be used in standard equations for torque, power consumption, and heat generation. The numerical values for *f* have been derived from seal testing on a variety of fluids under a wide range of operating conditions. Published values of *f* typically range from 0.01 to over 0.15. Typical values for *f* (Table 2) have proven useful for predicting the performance of mechanical seals.

Torque The torque required to spin a mechanical seal is directly proportional to the total face pressure *P_f*, the face area, the coefficient of friction *f*, and the mean radius of the seal face. Torque is most often used to determine the breakout torque required to start a pump. Under these conditions, the value of *f* is generally three to five (3 to 5X) times higher than values found in a running seals:

$$T = P_f A f R_m, \text{ in-lb (N-m)}$$

where *A* = face area, in² (mm²)

R_m = mean radius, in (mm)

Power Consumption Calculating the power requirements for a mechanical seal technically consists of the sum of the power requirements from three areas: seal faces friction, fluid turbulence, and pumping ring circulation. The friction between the seal faces creates a resistance to rotation, which must be overcome by the pump driver. The power required to rotate the seal faces *P_f* can be estimated with the following equation:

$$P_f = (T N) / C, \text{ kW (hp)}$$

where *N* = rotational speed rpm

C = constant for English units = 63,000 (constant for SI units = 9550)

Power is also consumed by windage or turbulence in the seal chamber. The actual power requirement is a function of the diameter of the seal, the seal rotational speed, the fluid viscosity, and the configuration of the seal chamber. Likewise, power is required for a pump-

TABLE 2 Effective coefficient of friction values for various fluids

Low viscosity fluids (e.g., light hydrocarbons)	0.05
Medium viscosity fluids (e.g., water based solutions)	0.07
High viscosity fluids (e.g., lubricating oils)	0.10

Note: The above values are typical for a seal face combination of carbon versus silicon carbide or tungsten carbide.

ing ring (if applicable) and this is a function of the pumping ring and seal gland design, the rotational speed and the fluid viscosity. The seal OEM can estimate these power requirements if all of the factors are known. From lab and field experience though, the seal face generated is the most significant contributor to seal power consumption and actual power requirements can be estimated by considering only P_f .

Temperature Control For stable seal performance, the heat generated by the mechanical seal must be dissipated. This is done to control the temperature of the fluid film between the seal faces, minimize thermal distortion of the faces, and control the fluid properties of the process fluid. In many applications, this temperature rise in the seal chamber is controlled by providing a flow of fluid (generally process fluid) through the seal chamber. The temperature rise in the seal chamber ΔT can be estimated by the following equation:

$$\Delta T = P_f / (C_i \cdot SG \cdot C_p \cdot Q)$$

where ΔT = temperature rise, °C (°F)

P_f = power, kW (Btu/h)

C_i = constant for SI units = 0.06969 (constant for English units = 500)

SG = specific gravity

C_p = specific heat, kcal/kg °C (Btu/h °F)

Q = injection flush flow rate, L/m (gpm)

In most cases, an attempt should be made to keep the temperature rise in the seal chamber less than 10°F (5°C).

Modeling of Seal Faces Modern mechanical seals successfully operate on an effective fluid film of approximately 20 millionths of an inch (0.5 micron). To maintain such a thin film, the seal faces must be manufactured flat and they must remain flat in operation. This is a challenge considering the wide range of speeds, pressures, and fluids found in seal applications. There are many mechanisms that attempt to distort the seal faces. These include drive loads, gasket loads, spring forces, and shrink fits. Seal designers can effectively minimize these distortions through careful seal design. There are two other mechanisms though which are always present and offer the greatest challenge in seal design—thermal distortions and pressure distortions.

When a seal is in operation, heat is generated between the sliding seal faces that results in an increase in temperature in the interface. Heat is conducted through the seal faces and largely dissipated by convection into the process fluid. This results in a temperature gradient along the axial length of the seal face. Since the contacting end of the face is at a higher temperature, it will thermally grow more than the cooler end of the seal face. The sealing surface will no longer be flat but will become convex. This phenomenon is referred to as thermal distortion (Figure 10). Thermal distortion is a function of the heat generated, the

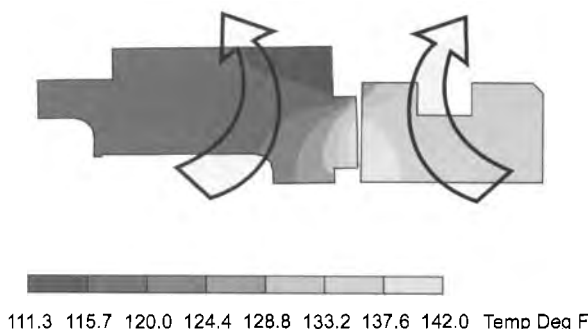


FIGURE 10 Thermal distortion

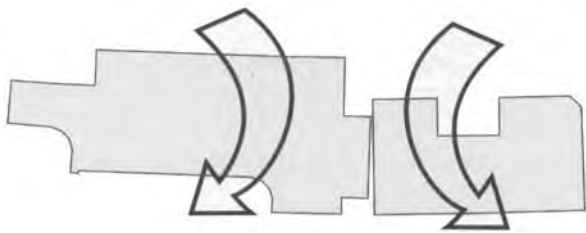


FIGURE 11 Pressure distortion

convective heat losses, the thermal conductivity of the seal face, and the coefficient of thermal expansion of the seal face. This distortion will result in a higher opening force between the faces and greater leakage from the seals.

To counteract thermal distortions, seal designers will design faces that tend to distort in the opposite direction so the sealing surfaces become concave. This phenomenon is referred to as pressure distortion (Figure 11). Pressure distortion is a function of the differential pressure, the shape of the face, and the modulus of the face material. The amount of pressure distortion should ideally be equal to the thermal distortion so the net result is that the seal faces remain flat. This is a formidable task since the fluid film is on the order 0.5 micron (20 millionths of an inch).

Current modeling techniques for modeling mechanical seals have evolved significantly in the past decade. While early attempts at modeling seals involved the simple use of finite element analysis (FEA), current techniques involve a coupling of FEA, computational fluid dynamics, and specialized code for mechanical seal. The specialized code defines the unique effects of contacting and noncontacting seal faces, variable fluid film properties, and multiphase films. It also describes the effects of modified faces design that provide additional fluid film support and hydrodynamic lift. These advances have allowed not only more accurate predictions of seal performance but have also allowed mechanical seals to be applied in services not possible in the past. A typical output from such an analysis is shown below (Figure 12).

Face Technologies The first mechanical seals were designed with flat faces to help establish a thin fluid film. The thin fluid film, at the microscopic level, provides an effectively tortuous path to minimize leakage. This aspect of seal design is still the basis for modern mechanical seals. The concept of flat face sealing was also driven by the available manufacturing methods. Since seal faces must be very flat, lapping has historically been the primary method for preparing the sealing surfaces of a seal face. Relatively large-scale features such as hydropads were introduced to improve film formation with some success. It was not until manufacturing methods for micro features such as grooves and waves were

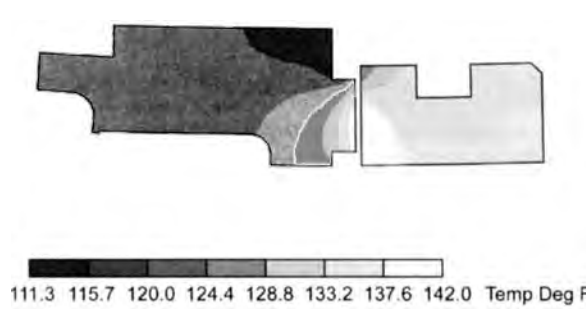


FIGURE 12 Net face distortion



FIGURE 13 Flat seal face

developed that more effective face designs were available. Current laser-based manufacturing technology allows seal manufacturers to manipulate sealing surfaces within millionths of an inch. This allows engineers to create sealing surfaces that are capable of acting as more than simple flow restrictions. Seal faces can now act as engineered surfaces that will lead to more advanced seal designs, expanded operating windows, and improved reliability.

A review of available face technologies can be broken four categories. Each of these technologies has different capabilities and application requirements:

FLAT FACES Flat seal faces (Figure 13) have smooth featureless surfaces and are manufactured to be flat within about 0.5 micron (20×10^{-6} in). This is achieved by lapping with an appropriate lapping compound such as diamond grit. The surface finish (or roughness) of the face is determined by the grit size of the lapping medium that is normally in the range of 3 to 8 microns. In special cases, lapping with coarser compounds, commonly referred to as matte lapping, may be required to improve film formation. Flat faces are measured on an optical flat to insure they are within specification.

FILM ENHANCING FACES There are applications though where the fluid properties and application conditions do not allow for an adequate film formation with flat seal faces. Fortunately, many techniques are available to help promote film formation and improve seal performance (Figure 14). Hydropads are large scallops on the sealing faces that provide a path for fluid to enter into the sealing interface. This can provide better cooling and the formation of a thicker fluid film. Very low amplitude waves are now being used successfully to create a more stable film, reduce heat generation, and reduce friction. This has been especially successful in light hydrocarbons and hot water applications. Face texturing involves creating small features such as dimples into the sealing surface that can improve the film formation of flat faces, but this approach has seen limited use in field applications.

HYDRODYNAMIC FACES Hydrodynamic faces are designed to create high pressures within the contact area of the seal faces. This high pressure increases the opening forces so that the faces achieve total lift-off and operate without any mechanical contact between the seal faces. Since the faces operate on a thick fluid film, leakage will be significantly higher than comparable seals operating with contacting faces. Operating on a thick fluid film allows



FIGURE 14 Film enhancing seal face

for higher speed operation and lower heat generation. Most hydrodynamic seals are used in gas seal designs where the seals are expected to operate continuously on vapor or gas. Special designs have been built which can operate on liquid, gas, or mixed phase processes (e.g., light hydrocarbon services). Typical designs include spiral grooves, T-grooves, or wavy faces (Figure 15).



FIGURE 15 Hydrodynamic seal face



FIGURE 16 Special design seal face

SPECIAL FACE DESIGNS The advent of micromachining capabilities has lead to a fundamental shift in how seal faces are viewed. Rather than a large flat surface, seal faces have the potential to be engineered devices with expanded functions other simple sealing. An example of such a device uses the surface tension properties of the sealed fluid to create a hermetic, noncontacting seal (Figure 16). Fluid in the seal faces collects on a series of small surface tension bands micro machined into the seal surface. This prevents further movement of the fluid. Face separation is maintained by hydrodynamic waves, which operate on air from outside of the process fluid. The combination of hydrodynamic lift and surface tension has proven successful in applications where other technologies have failed. Other examples of new face technology are being developed and will be offered commercially in the near future.

MATERIALS OF CONSTRUCTION

The many components of a mechanical seal have vastly different functions and material requirements and therefore require a wide range of materials. Over the years, seal manufacturers have standardized on materials that cover a broad range of seal designs and applications.

Face Materials The selection of seal face materials is one of the most critical decisions in a seal design. In most applications, a face combination is selected where one face is a soft face (e.g., carbon/graphite) and the mating face is a hard face (e.g., ceramic based material such as silicon carbide, tungsten carbide, or alumina oxide). This combination of a soft and hard face allows for predictable wearing of the surfaces, a low coefficient of friction, and better tolerance of face contact. This combination is the default selection for mechanical seals. In some application, two hard faces are selected which makes the seal more tolerant of abrasive process fluids.

Seal face materials should ideally meet the following criteria:

- Good tribological properties
- High modulus
- High thermal conductivity

- Excellent chemical corrosion resistance
- Resistant to impact damage and breakage
- Easy to manufacture
- Relatively low cost

No single material meets all of these criteria but many come very close. The most common seal face materials are listed below. These descriptions are generic terms and may represent several (or dozens of) grades offered by a specific material supplier. The seal OEMs have qualified specific grades that meet the requirements for their seals.

Resin Impregnated Carbon Graphite This is the most common grade of carbon material used in seal faces. It is a mixture of amorphous carbon and carbon graphite, which has been impregnated with a resin or phenolic impregnant. This grade provides the best combination of low friction, wear resistance, and chemical compatibility of the carbon grades. It is limited in aggressive chemical services by the resistance of the impregnant.

Metal Impregnated Carbon Graphite Metal impregnated carbon grades are also a mixture of amorphous carbon and carbon graphite but these have been impregnated with a low melting point metal. The metal impregnant provides high strength, a higher modulus, and greater resistance to blistering. Currently, antimony is the most common metal used in mechanical seals. This grade is used extensively in light hydrocarbon applications.

Reaction Bonded Silicon Carbide Reaction bonded silicon carbide (RBSiC) is formed by reacting molten silicon metal with a mixture of carbon and SiC powder. This results in the SiC being "reaction bonded" together. The material also contains approximately 10% unreacted silicon metal. This grade has excellent hardness, high thermal conductivity, a high modulus, and good tribological properties. RBSiC is the most common grade of SiC used in the refinery industry. Its use in aggressive chemicals is limited by the chemical compatibility of the silicon metal. It should not be used in caustics and aggressive acids.

Self-Sintered Silicon Carbide Self-sintered silicon carbide (SSSiC) is formed by directly sintering silicon carbide together with the use of minute amounts of a sintering aid. The result is a material composed entirely of SiC, which has excellent hardness, high thermal conductivity, a high modulus, and good tribological properties. In addition, it is chemically resistant to almost any fluid likely to be sealed by a mechanical seal. For this reason, it is the preferred material for chemical applications or highly corrosive services. This grade is also known as direct sintered SiC, alpha sintered SiC, or simply sintered SiC.

Nickel Bound Tungsten Carbide Nickel bound tungsten carbide (Ni-WC) is a cemented ceramic with tungsten carbide particles in a metallic nickel matrix. Ni-WC is a very dense, very tough material, which is used in applications where toughness, clamping, or impact resistance is required. While the material is excellent in most services, it can easily heat check in marginally lubricated applications. Ni-WC is chemically limited by the compatibility its nickel binder.

Alumina Oxide Alumina oxide is a white ceramic material manufactured in various purities from around 94% to over 99%. It shares many of the same properties as other ceramics such as high hardness, high modulus, and excellent chemical resistance. It has lower thermal conductivity though which contributes to its poor thermal shock resistance. Its use in mechanical seals is generally limited to lower duty services in chemical or pharmaceutical applications.

Other Specialized Materials

DRY RUNNING CARBONS In some designs, the seal faces are required to run without a supporting fluid film. Dry running grades of carbon are high graphitic materials designed to

operate in a contacting mode with low friction, low heat generation, and low wear. These grades are specifically designed to run with very low contact pressures.

ACID GRADE CARBONS Acid grade carbons are very pure grades of carbon with a very low concentration of impurities and ash found in standard carbon grades. This allows the materials to be applied in extremely aggressive chemical services. There are trade-offs in performance so this grade is used only where the chemical resistance is required.

METALLIC MATERIALS Metallic materials are used for a variety of components in seals including springs, drive mechanisms, sleeves, glands, and fasteners. The materials in these applications are selected primarily for their corrosion resistance and physical properties. The selection of materials for welded metal bellows introduces additional considerations since it must not only be corrosion resistant but must also be easy to weld. High temperature applications also require that the materials retain their physical properties at extreme temperatures. The most common metals are listed below:

316 AUSTENITIC STAINLESS STEELS (UNS S31600) Austenitic stainless steels (especially AISI 316) are the default materials for most mechanical seals. This grade is also available in other variations such as 316L (UNS S31603), which can be easily welded into bellows, and 316Ti (UNS S31635), which is more available in some locations. AISI 316 is resistant to a wide range of chemicals such as waters, hydrocarbons, and hydrogen sulfide. It is resistant to many acids and bases although this varies with the specific chemical, concentration, and temperature. It should be used with caution in high chloride environments due to pitting corrosion.

ALLOY C-276 (UNS N10276) Alloy C-276 is a nickel-molybdenum-chromium alloy with excellent chemical compatibility in a wide range of aggressive applications including many acids and brines. It is used extensively for all seal parts including springs and welded metal bellows.

ALLOY 400 (UNS N04400) Alloy 400 is a copper-nickel alloy with excellent chemical resistance to aggressive acids and bases. Due its high cost, its use is generally limited to hydrofluoric acid applications. This grade can be used in the construction of welded metal bellows. Related grades such as Alloy K-500 (UNS N05500) are used if high strengths are required for components such as fasteners.

ALLOY 20 (UNS N08020) Alloy 20 is a nickel-chromium-molybdenum alloy which has excellent chemical resistance to hot sulfuric acids and many other aggressive chemicals. It can be easily welded and it is used in the construction of welded metal bellows. It is generally only used when more common alloys (such as 316 or alloy C-276) are not suitable for a specific application.

ALLOW 718 (UNS N07718) Alloy 718 is used almost exclusively in the construction of high temperature bellows. Its combination of high temperature capabilities, resistance to sour environments, and welding properties have made it the standard bellows materials for high temperature refinery services.

Low Expansion Alloys Low expansion alloys are a family of iron-nickels alloys that have a coefficient of thermal expansion similar to most seal face materials. This makes it a suitable material for shrink fit constructions in high temperature bellows applications. These alloys generally have poor corrosion resistance so their use is limited to high temperature services.

Elastomeric and Gasketing Materials Mechanical seals have a number of gaskets (commonly referred to as secondary seals) that prevent leakage of process fluids between the components of the seal. These include gaskets at the interface of the sleeve and shaft, gland and seal chamber, and between internal seal components. The gaskets must allow for thermal expansion of the components, conform to irregular surfaces, and be chemically resistant

to the process fluids. The selection of the gasket materials is generally guided by the chemical and temperature requirements of the application. The selection of gasket materials may also drive the design of seal components due to the gasket installation requirements. Commonly used gasket materials are listed below:

ETHYLENE PROPYLENE (EPR) AND ETHYLENE PROPYLENE DIENE (EPDM) EPR and EPDM, while widely used in general industry, have limited applications in mechanical seals. They possess excellent resistance to hot water, ketones, and alcohols but are incompatible with petroleum-based fluids and lubricants.

NITRILE (NBR) Nitrile is an excellent general duty elastomer used in seal applications. It is suitable for most petroleum-based fluids although it has limited compatibility to many aggressive chemicals. Nitrile has the unique ability to be compounded for low temperature services (down to -54°C or -65°F) so it is often used in cryogenic applications.

FLUOROELASTOMERS (FKM) Fluoroelastomers provide excellent chemical resistance and good high temperature capabilities. As such, they have become the standard elastomer grade for general duty mechanical seals. They have excellent chemical resistance to hydrocarbons, acids, and water. They are not recommended for ketones, amines, and hot water services.

PERFLUOROELASTOMERS (FFKM) Perfluoroelastomers are highly fluorinated elastomers with an excellent chemical resistance and high temperature properties. This includes applications in very aggressive acids, bases, and amines. The actual chemical resistance and temperature limits for FFKMs are very dependant on the specific compound. Some compounds are intended for broad general duty use; others are intended for specific applications. The seal OEM or elastomer manufacturer can provide guidance in selecting specific grades.

FLEXIBLE GRAPHITE Flexible graphite has near universal chemical resistance as a gasket material. It is also stable at extremely high temperatures. This has made it the standard selection for high temperature seal applications. Flexible graphite has almost no resiliency so it must be mechanical clamped or hydraulically loaded against its mating surface.

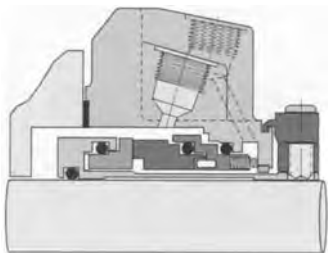
POLYTETRAFLUOROETHYLENE (PTFE) Polytetrafluoroethylene is a fully fluorinated material commonly used in a variety of gaskets. It has near universal chemical compatibility, which makes it useful for aggressive services or batch processes. It is a hard, inflexible material that requires special gasket groove designs and installation practices.

CLASSIFICATION OF MECHANICAL SEALS

With the wide range of seal designs and applications available, it is natural to categorize the options to assist in referring to specific features or designs. There are many different ways to categorize seals and these are fairly consistent among seal OEMs. Common categories include basic design, orientation, and arrangement. Each of the options in these categories has advantages and disadvantages that may help the user in selecting the correct seal.

Classification by API 682 Seal Type API 682, "Pumps—Shaft Sealing Systems for Centrifugal and Rotary Pumps" is a mechanical seal standard published by the American Petroleum Institute. This standard, first published in 1994, has become the most popular standard for mechanical seals. API 682 was written primarily by end users with input from seal and pump OEMs. While the standard was originally intended solely for refinery applications, later revisions expanded the scope to include chemical duty applications. API 682 is also published as ISO 21049.

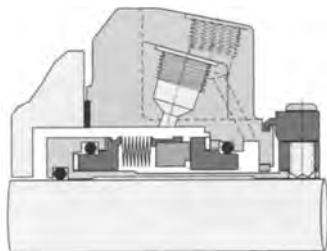
API 682 has introduced several definitions to help categorize seal by design features. Seal Type refers to a seal definition that includes the operating window for the seal, basic design features, and materials of construction. Each Seal Type can be applied to different arrangements and orientations.

TABLE 3 Classification by design and orientation**Pusher seal****FIGURE 17**

Definition: Seal with a flexibly mounted seal face supported by a dynamic gasket, which is allowed to move axially to compensate for shaft motion or face wear

Advantages: Simple design, flexible material selection, can be designed for high pressures

Disadvantages: Requires dynamic gasket, subject to hang-up or fretting under gasket

Bellows seal**FIGURE 18**

Definition: A seal with convolutions of metal or elastomeric material providing sealing of process fluids and mechanical loading of the seal faces.

Advantages: Eliminates dynamic gasket, can be constructed without elastomeric gaskets, can be designed for high temperatures

Disadvantages: Limited material selection for bellows, lower pressure rating than pusher seals, thin metal diaphragms subject to erosive wear

(Continued)

TABLE 3 Classification by design and orientation (*Continued*)

Single spring pusher

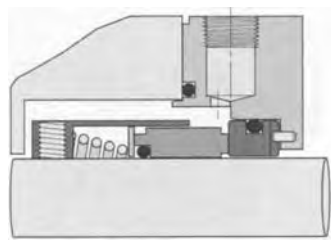


FIGURE 19

Definition: A pusher seal design where the mechanical load is provided by a single coil spring.

Advantages: Large wire cross section more tolerant of corrosion and erosion, low spring rate more tolerant of axial shaft motion

Disadvantages: Requires more axial space, can provide uneven loading of seal faces

Multiple spring pusher

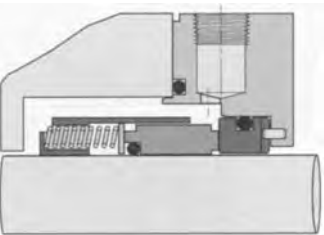


FIGURE 20

Definition: A pusher seal design where the mechanical load is provided by multiple coil springs.

Advantages: Compact package, more uniform face loading

Disadvantages: Small wires subject to erosion and less tolerant of corrosion

Internally mounted seal

Definition: Seal is located inside the seal chamber

Advantages: Pressurized from OD of seal, better circulation of fluid around seal faces, allows for bushing and drain in seal gland to contain leakage

Disadvantages: More components exposed to process fluid

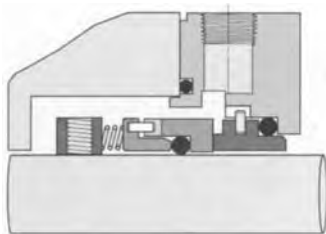


FIGURE 21

Externally mounted seal

Definition: Seal is located outside of seal chamber

Advantages: Few seal components exposed to process fluid, requires no space in seal chamber

Disadvantages: Faces are pressurized on the ID, leakage can not be contained, low pressure rating

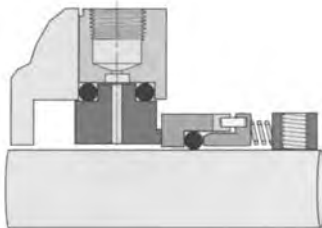


FIGURE 22

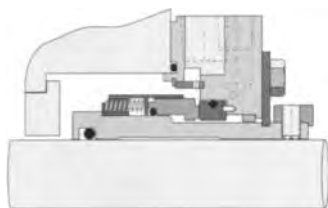
(Continued)

TABLE 3 Classification by design and orientation (*Continued*)**Rotating flexible element**

Definition: The flexible element rotates with the shaft

Advantages: Good circulation around spring components, tends to be self-cleaning

Disadvantages: Potential for erosion of seal components, less tolerant of misalignment

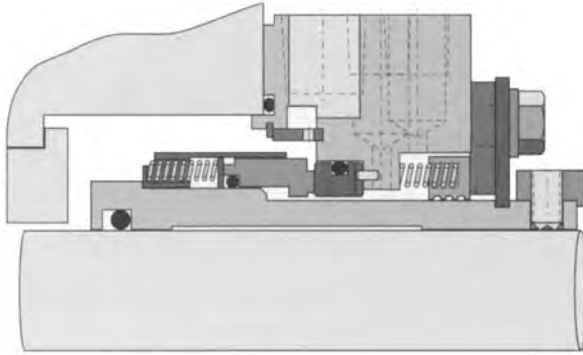
**FIGURE 23****Stationary flexible elements**

Definition: The flexible element is stationary with the gland

Advantages: More tolerant of misalignments, able to operate at higher shaft speeds

Disadvantages: Requires more radial space, may be more difficult to design adequate flush

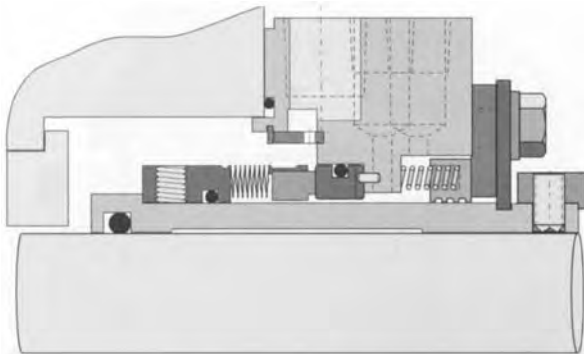
**FIGURE 24**

**FIGURE 25** Type A seal

TYPE A A Type A seal (Figure 25) is an inside mounted, balanced pusher seal. The default design has a rotating flexible element although a stationary flexible element is optional. The default design utilizes multiple springs (in alloy C-276) although there is an option for a single coil spring (316 SS). The materials of construction are carbon versus silicon carbide faces, FKM gaskets, and a 316 SS sleeve and gland. The operating temperature range for the seal is -40°C to 176°C (-40°F to 350°F). The Type A seal is considered the default design for most applications within API 662.

TYPE B A Type B seal (Figure 26) is an inside mounted, balanced general-duty bellows seals. The default design has a rotating bellows assembly although a stationary bellows is optional. The default materials of constructions are alloy C-276 bellows diaphragms, carbon versus silicon carbide faces, FKM gaskets, and a 316 SS sleeve and gland. The operating temperature range for the seal is -40°C to 176°C (-40°F to 350°F).

TYPE C A Type C seal (Figure 27) is an inside mounted, balanced high-temperature bellows seal. The default design has a stationary flexible element although a rotating flexible element is optional. The default materials of constructions are alloy 718 bellows diaphragms, carbon versus silicon carbide faces, flexible graphite gaskets, and a 316 SS sleeve and gland. There is also a requirement for a bronze antickoke device and a floating carbon throttle bushing. The operating temperature range for the seal is -40°C to 400°C (-40°F to 750°F).

**FIGURE 26** Type B seal

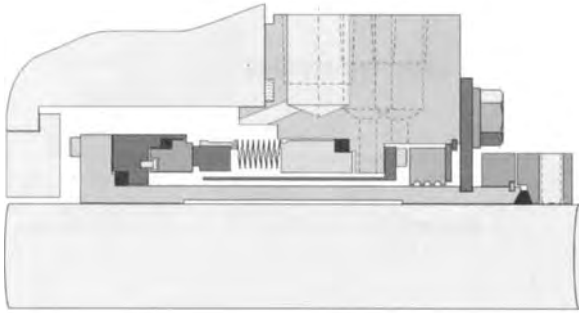


FIGURE 27 Type C seal

Classification by Arrangements and Configurations Mechanical seals are often applied where only one mechanical seal is installed into the seal chamber and the seal operates on the pump process fluid. While this is acceptable for many installations there may be conditions where a single seal cannot be used. This may include applications where the pump fluid is unreliable or is not suitable for a mechanical seal. It can also include applications where leakage from a failed seal would present an unacceptable environmental or safety hazard. To address these issues, arrangements are available where multiple seals are used. The most common seal arrangements are listed below.

SINGLE SEALS Single seals (Figure 28) consist of one set of seal faces. It is the simplest design and, in general, requires minimal outside support systems or auxiliary equipment. The majority of single seals operate on the process fluid although an outside seal flush may be utilized. Leakage past a single seal will go to an atmospheric drain or vaporize and go directly into the atmosphere. If the application contains fluids which are regulated emissions, carcinogens, above the auto-ignition temperature, or present other hazards, the user should consider the consequences of both normal leakage and leakage under failure conditions. This arrangement is defined as an Arrangement 1 seal in API 682.

DUAL UNPRESSURIZED SEALS Dual unpressurized seals (Figure 29) are seal assemblies designed to contain two mechanical seals in series. In this arrangement, the inner (or primary) seal operates on the process fluid under seal chamber process conditions. The seal outer (or secondary) seal is intended to operate as a backup seal under low-pressure conditions. Under normal operation, seal leakage past the inner seal would collect between the two seals and be routed to a flare or recovery system. In the event of a failure of the

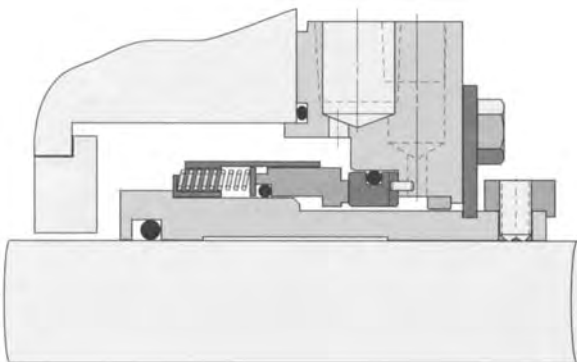


FIGURE 28 Single seal

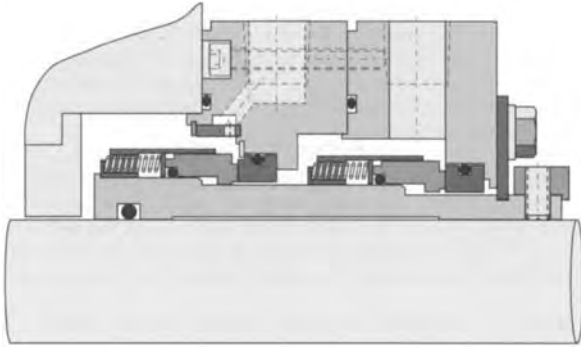


FIGURE 29 Dual unpressurized seal

inner seal, the outer seal could contain the process fluid under full seal chamber pressure. In either scenario, leakage of process fluid to atmosphere would be minimal.

The cavity between the primary and secondary seal may be filled with a liquid or operated dry. If the cavity is filled with a liquid, the arrangement is referred to as a dual unpressurized seal with a liquid buffer fluid. The buffer fluid serves the dual purpose of providing the seal environment for the outer seal and a collection point for inner seal leakage. The outer seal is a standard liquid mechanical seal with the same rating as the inner seal. During operation, the buffer fluid is circulated from the secondary seal cavity to a reservoir that provides cooling and monitoring of the buffer fluid level.

If the cavity between the inner and outer seal is operated without a liquid, the arrangement is referred to as a dual unpressurized seal with a containment seal. A containment seal is a dry-running backup seal specifically designed to operate without liquid lubrication. The cavity between the inner and outer seal is referred to as the containment seal cavity. This cavity must be maintained under low-pressure conditions (less than 0.7 bar or 10 psi). Leakage from the inner seal is routed to the flare or a recovery system. In the event of failure of the inner seal, the outer seal could contain the process fluid under full seal chamber pressure. Leakage of process fluid to atmosphere would be minimal under normal seal operation or in the event of an inner seal failure. This arrangement is defined as an Arrangement 2 seal in API 682.

DUAL PRESSURIZED SEALS Dual pressurized seals (Figure 30) are used when, for some circumstances, it is not desirable to have any process leakage released to atmosphere. There may be other applications where the process fluid is unsuitable or too unreliable for mechanical seals. In these cases, a dual pressurized seal arrangement is used. A dual pressurized

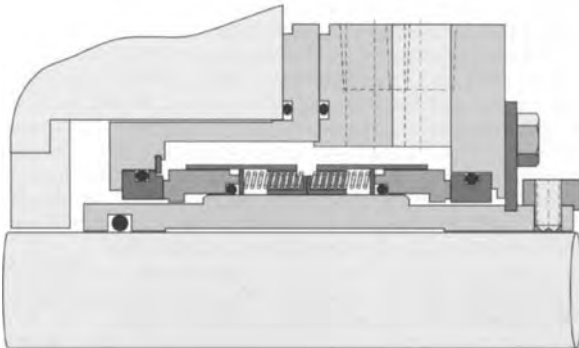


FIGURE 30 Dual pressurized seal

seal consists of two sets of seal faces arranged in a face-to-back, back-to-back, or face-to-face orientation. The cavity between the inner and outer seal is referred to as the barrier fluid cavity. The barrier fluid is maintained at a pressure higher than found in the seal chamber that effectively causes both the inner and outer seal to operate on a fluid film of the barrier fluid. The pressurization also prevents any process fluid from entering the barrier fluid cavity and leaking to atmosphere.

If the barrier fluid is a liquid, it will be circulated from the seal to an external reservoir (or accumulator) and back to the barrier cavity. The support system will not only provide pressurization of the barrier fluid but also cooling and monitoring of the barrier fluid liquid level. If the barrier fluid is a gas, the seal is often referred to as a dual gas seal. The barrier gas is provided by a gas control panel that cleans the gas, regulates the pressure, and monitors the flow rate to the seal. This arrangement is defined as an Arrangement 3 seal in API 682.

Special Applications There are other special applications that do not fit neatly into the above arrangements. These would include dual seals designed to pressure share (or break-down) the process fluid pressure so the application can be more reliably sealed. They may also include seal arrangements with three or more sets of seal faces. While these examples are not common, they do illustrate that special seal arrangements can be developed to meet unusual application requirements.

CONTROL OF SEAL ENVIRONMENT

Mechanical seals operate most reliably when the seal chamber conditions promote a clean, cool fluid film between the faces. In practice, the process environment is often less than ideal. Fortunately, methods of improving the seal environment are available which can circulate, clean, and cool the seal chamber fluid. Seals are often installed in equipment that was designed for packing. This can limit the selection for sealing options or provide poor fluid circulation around the seal. There are seal chamber designs and pump retrofits available which can provide the required space for reliable seal performance. Providing a good environment for the seal is as critical to success as selecting the correct seal.

Seal Chamber Options Historically packing was used to provide the seal in centrifugal and rotary pumps. Packing is generally manufactured in rings with a square cross section ranging from approximately 8 mm (5/16 in) for small chemical duty pumps to 20 mm (3/4 in) for larger heavy-duty pumps. Unfortunately, these small cross sections reflect the small radial clearances found in a typical stuffing box. In addition, the effectiveness of the packing arrangements also depended upon the number of rings used in the installation. On high-pressure pumps, this resulted in many rings of packing and a very deep stuffing box. Packing not only provided sealing for the process fluid but also provided mechanical support for the shaft and was sometimes critical for proper control of shaft deflections. Simply removing packing and installing a mechanical seal often resulted in poor seal performance.

To provide a better environment for mechanical seals, pump manufacturers began offering seal chamber designs specifically to accommodate larger seals and provide better circulation of fluids in the seal chamber. One class of pumps (ASME B73 series) increased the radial clearance from 8 mm to 11 mm (5/16 in to 7/16 in) in their stuffing boxes to 19 mm to 25 mm (3/4 in to 1 in) in their seal chambers. This allowed for the option to install a wider range of seals and provide a better environment. Lab testing has shown that seal chambers provide a better control of mechanical seal temperatures and lower overall temperatures compared to comparable stuffing boxes. Many end users recognize this improvement and specify only seal chambers when new pumps are purchased with mechanical seals. Unfortunately, many end users continue to use stuffing boxes on sealed pumps previously installed in the field even though upgrade options are available.

Seal chamber designs are defined in pump standard such as API 610/ISO 13709 and ASME B73. These designs can be broken into three main categories: cylindrical bore, tapered bore, and tapered bore with flow modifiers (Figure 31).

Cylindrical bore seal chambers maintain the same basic shape as a stuffing box although the radial clearance between the shaft and seal chamber bore has been increased. This design is used for general seal applications and works well with most piping plans. Some

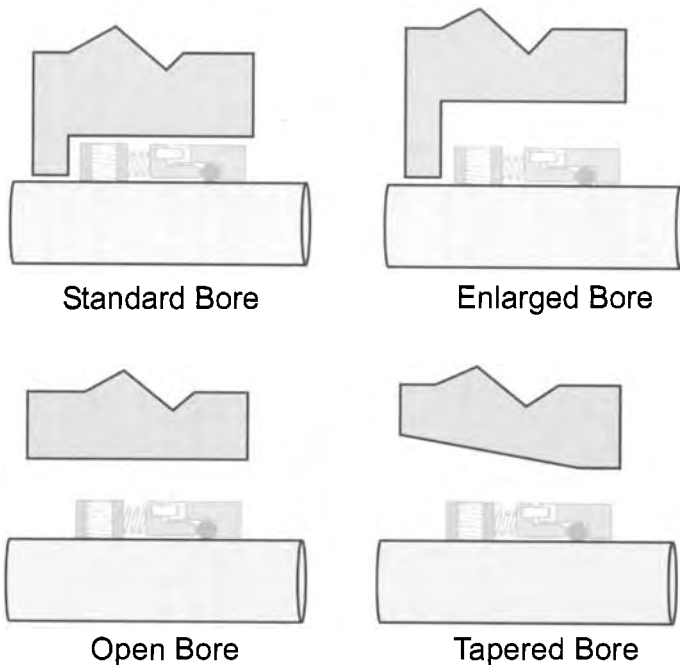


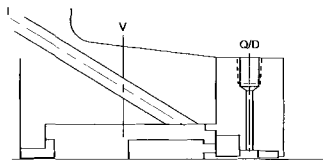
FIGURE 31 Comparison of seal chambers

piping plans require isolation of the seal chamber from the pump process. This may be due to cooling the process fluid (e.g., Plan 21 or 23), cleaning the process fluid (e.g., Plan 31), or introducing an external fluid (e.g., Plan 32). In these cases, a cylindrical bore seal chamber with a throat bushing is required to get the full benefit of the piping plan.

Many pumps are applied where the natural circulation of the fluid in the seal chamber provides all of the cooling requirements for the mechanical seal. To assist in this circulation, seal chambers have been designed which are tapered from the face of the seal chamber back to the impeller. This design is appropriately referred to as a taper bore seal chamber. This design provides good circulation around the mechanical seal and should be used when no external piping plan is used (Plan 02). Tapered bore seal chambers are also inherently self-venting and self-draining.

While tapered bore seal chambers provided good circulation, lab testing and field experience showed that the circulation flows were not as simple as first thought. Areas inside the seal chamber could collect solid particles from the process fluids. To assist in removing these particles from the seal chamber, the bore of the taper was modified to provide protrusion to redirect the rotational flow of the fluid into axial flow. This eliminated any stagnant areas and effectively cleaned the fluid in the seal chamber. These seal chambers are generically referred to as tapered bore seal chambers with flow modifiers and they are available from a number of pump OEMs.

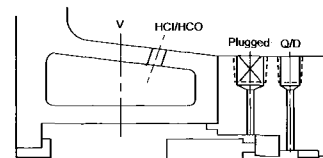
Seal Piping Plans To further improve the environment around the seal, fluid is circulated through the seal chamber. The condition of this fluid may be modified by cleaning and cooling the fluid or by increasing or decreasing the seal chamber pressure. Dual pressurized and dual unpressurized seals have further requirements for handling the barrier fluid or buffer fluid, respectively. There are also considerations for modifying the environment on the atmospheric side of the seal or detecting atmospheric side leakage. All of these methods of improving the seal environment have been captured in standardized seal piping plans. These have been widely published and are generally maintained in API 682 and ASME B73. A list of the current piping is shown below:

TABLE 4 Seal piping plans**Plan 01****FIGURE 32**

Definition: Internal recirculation from a high pressure region of the pump to the seal chamber

Advantages: Built-in from factory

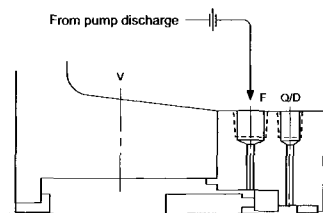
Disadvantages: Flow is not easy to control or modify, injection is generally not directed at the seal faces, difficult to troubleshoot

Plan 02**FIGURE 33**

Definition: Dead-end seal chamber with no flush/circulation. Taper bore seal chambers are generally recommended to improve seal chamber circulation

Advantages: Simple system requiring no external piping or auxiliaries

Disadvantages: Generally limited to lower pressure, lower speed applications

Plan 11**FIGURE 34**

Definition: Circulation of process fluid from the pump discharge (or other high pressure region of the pump) through an orifice into the seal chamber

Advantages: Provides flow to carry away seal generated heat, can be used to increase seal chamber pressure, easy to modify flow rate by changing orifice

Disadvantages: Orifice can become plugged restricting circulation, process fluid must not solidify under atmospheric temperatures or heat tracing may be required

Plan 12

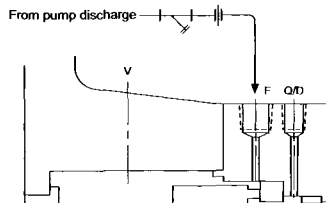


FIGURE 35

Definition: Circulation of process fluid from the pump discharge (or other high pressure region of the pump) through a strainer and an orifice into the seal chamber

Advantages: Provides screening of process fluid to prevent orifice plugging, capable of keep large debris out of seal chamber

Disadvantages: Strainer can become plugged restricting circulation, not recommended for new applications

Plan 13

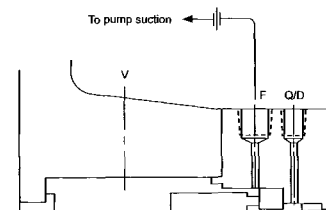


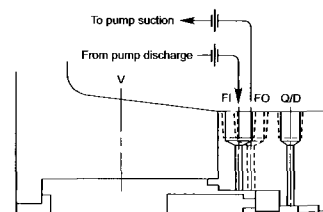
FIGURE 36

Definition: Circulation from seal chamber through an orifice to the pump suction or other low-pressure region of the pump

Advantages: Used in vertical pumps where seal chamber is near discharge pressure, can be used to reduce pressure in seal chamber, can be used to provide clean flush in some pump constructions

Disadvantages: May not be useable in some pump constructions due to lack of pressure differential, orifice can become plugged restricting circulation

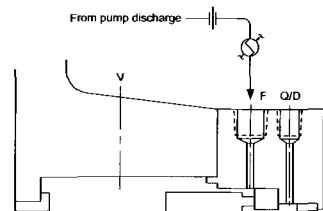
(Continued)

TABLE 4 Seal piping plans (*Continued*)**Plan 14****FIGURE 37**

Definition: Combination of Plan 11 and Plan 13, fluid circulation from pump discharge to seal chamber and from seal chamber to pump suction

Advantages: Provides continual flow into and out of the seal chamber

Disadvantages: Difficult to determine resulting pressure in seal chamber

Plan 21**FIGURE 38**

Definition: Circulation of process fluid from the pump discharge (or other high pressure region of the pump) through an orifice and seal cooler into the seal chamber

Advantages: Reduces the temperature of injected fluid which reduces its vapor pressure and increases its viscosity

Disadvantages: Reinjection of cold fluid back into pump/system, inefficient method of cooling fluid, will cause fouling of water-cooled seal coolers in high temperature applications

Plan 22

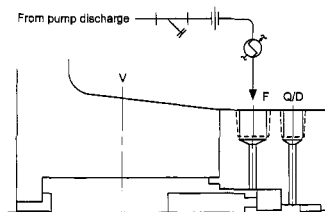


FIGURE 39

Definition: Circulation of process fluid from the pump discharge (or other high pressure region of the pump) through a strainer, an orifice, and seal cooler into the seal chamber

Advantages: Reduces the temperature of injected fluid which reduces its vapor pressure and increases its viscosity, provides screening of process fluid to prevent orifice plugging, capable of keep large debris out of seal chamber

Disadvantages: Reinjection of cold fluid back into pump/system, inefficient method of cooling fluid, will cause fouling of water-cooled seal coolers in high temperature applications, strainer can become plugged restricting circulation, not recommended for new applications

Plan 23

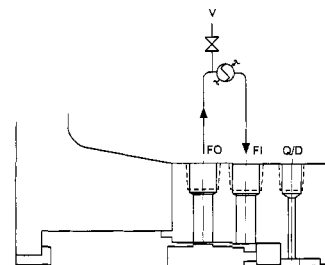


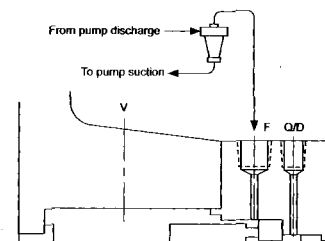
FIGURE 40

Definition: Circulation of process fluid from the seal chamber through a seal cooler and back into the seal chamber

Advantages: Provides efficient cooling with minimum size seal cooler and minimum cooling water requirements, reduces the temperature of circulated fluid which reduces its vapor pressure and increases its viscosity, does not inject cooled process fluid back into pump/system

Disadvantages: Requires careful venting piping prior to start-up

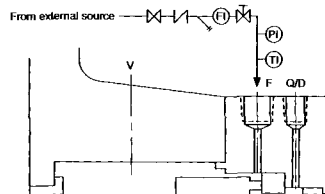
(Continued)

TABLE 4 Seal piping plans (*Continued*)**Plan 31****FIGURE 41**

Definition: Circulation of process fluid from the pump discharge (or other high pressure region of the pump) through a cyclone separator with clean fluid directed into the seal chamber and dirty fluid directed into the pump suction

Advantages: Provides cleaner injection fluid to seal chamber

Disadvantages: Not effective in many applications, may clog or plug in operation, requires similar pressure in seal chamber and cyclone return line to pump suction

Plan 32**FIGURE 42**

Definition: Injection of flush fluid from an outside source into seal chamber

Advantages: Flush fluid is independent of pump process

Disadvantages: Seal reliability depends upon reliability of flush source, flush must be compatible with process fluid and pump conditions (especially temperature), may be expensive due to cost in flush fluid

Plan 41

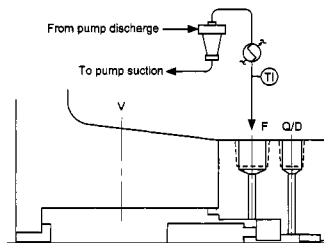


FIGURE 43

Definition: Circulation of process fluid from the pump discharge (or other high pressure region of the pump) through a cyclone separator and seal cooler into the seal chamber

Advantages: Provides cleaner injection fluid to seal chamber, reduces the temperature of injected fluid which reduces its vapor pressure and increases its viscosity

Disadvantages: Not effective in many applications, may clog or plug in operation, requires similar pressure in seal chamber and cyclone return line to pump suction, reinjection of cold fluid back into pump/system, inefficient method of cooling fluid, will cause fouling of water-cooled seal coolers in high temperature applications, strainer can become plugged restricting circulation, not recommended for new applications

Plan 51

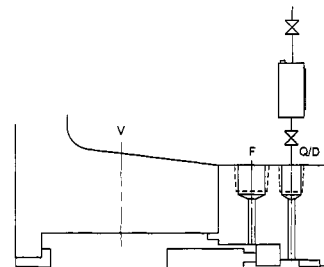


FIGURE 44

Definition: External reservoir providing a dead-end blanket on the atmospheric side of the seal

Advantages: Can modify the atmospheric side of the seal to prevent icing or crystallization

Disadvantages: Useful only in vertical pumps

(Continued)

TABLE 4 Seal piping plans (*Continued*)

Plan 52

Definition: External reservoir supplying buffer fluid to a dual unpressurized seal assembly

Advantages: Captures leakage from inner seal, buffer fluid provides additional cooling to inner seal, operated at low pressure, directs inner seal leakage to flare or vapor recovery system, minimizes process leakage to atmosphere

Disadvantages: Requires external reservoir, buffer fluid, and cooling water (in many cases), requires monitoring of buffer fluid level

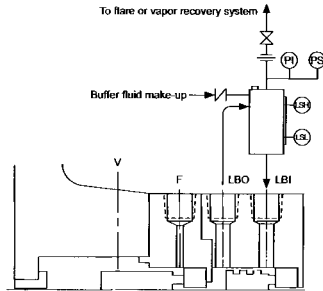


FIGURE 45

Plan 53

Plan 53 represents a series of piping plans that provide a pressurized liquid barrier fluid to a dual pressurized seal assembly. The system not only provides pressurization but also cooling of the barrier fluid as well as instrumentation to monitor system pressure and barrier fluid levels. In the past, a Plan 53 was assumed to consist of a reservoir pressurized with an inert gas. Variations started to appear which were informally designated as a Plan 53 Modified. To help provide a better definition of the variations and allow users to specify them, three versions of Plan 53 was defined

Plan 53A

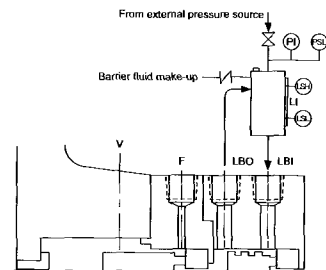


FIGURE 46

Definition: Pressurized external reservoir providing barrier fluid to a dual pressurized seal, pressurization is provided by inert blanket of gas (usually Nitrogen) controlled by a pressure regulator, barrier fluid level is monitored through sight glass or instrumentation.

Advantages: Fairly simple pressurization scheme, easy monitoring of fluid level changes

Disadvantages: Interface between pressurization gas and barrier fluid promotes gas absorption, generally limited to less than 14 bar (200 psi), requires continuous supply of pressurization gas

Plan 53B

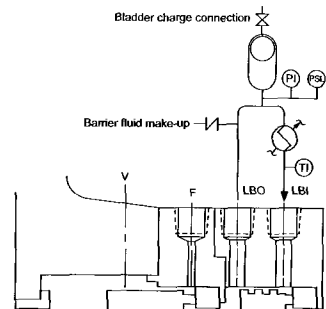


FIGURE 47

Definition: Pressurized external system providing barrier fluid to a dual pressurized seal, pressurization is provided by a pre-charged bladder accumulator, barrier fluid "level" is monitored by system pressure.

Advantages: Suitable of high pressure applications, does not require continuous external pressurization, can be operated in remote areas

Disadvantages: Barrier fluid pressure varies with seal leakage, refill rates determined by size of accumulator

(Continued)

TABLE 4 Seal piping plans (*Continued*)

Plan 53C

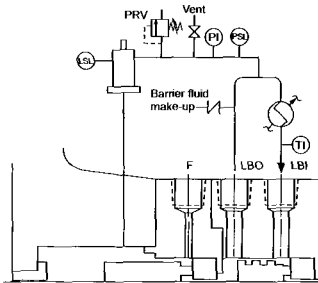


FIGURE 48

Definition: Pressurized external system providing barrier fluid to a dual pressurized seal, pressurization is provided by a piston transmitter which increases the barrier fluid pressure by a fixed ratio, barrier fluid level is monitored by piston extension

Advantages: Automatically provides correct pressurization of barrier system, handles process upsets or pressure spikes, can operate at high pressures

Disadvantages: Limited volume in transmitter, transmitter is exposed to process fluid, limited material selection for transmitters, not suitable for dirty applications or where process fluid may freeze, crystallize, or plate out

Plan 54

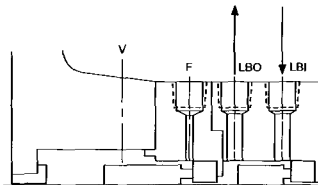


FIGURE 49

Definition: Pressurized barrier fluid provided by an outside source, source may be a closed circulation system or a once-through process stream

Advantages: Easy control of flush flow rate, ability to handle high pressures and high heat loads

Disadvantages: Seal reliability depends upon reliability of barrier fluid system or source, circulation systems may become complex

Plan 61

Definition: Atmospheric side connections (quench and drain ports) plugged for purchaser's use

Advantages: Ports available for future use

Disadvantages: Ports may be left plugged and provide no benefits

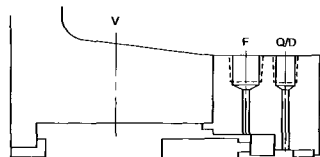


FIGURE 50

Plan 62

Definition: External quench provided to the atmospheric side of seal (usually steam, nitrogen, or water)

Advantages: Modify the environment on the atmospheric side of the seal, minimize coking of hot process fluids, prevent atmospheric crystallization of some process fluids (e.g., caustic)

Disadvantages: Wet steam quenches create problems in hot services, leakage of liquid quench to atmospheric side of seal

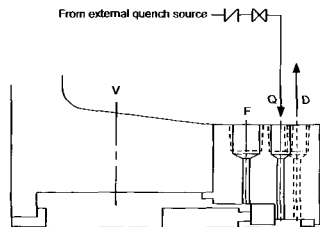


FIGURE 51

(Continued)

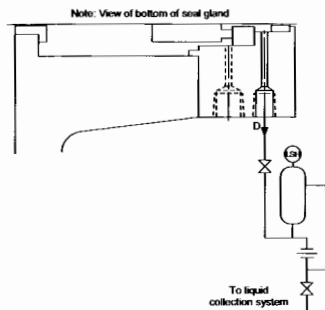
TABLE 4 Seal piping plans (*Continued*)

Plan 65

Definition: External drain from atmospheric side of seal to monitor seal leakage

Advantages: Provide remote alarm of leakage from single seal

Disadvantages: Requires a high leak rate to alarm system, leakage which will solidify or crystallize under atmospheric conditions may require system to be heat traced

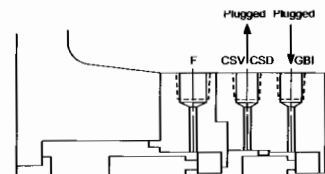
**FIGURE 52**

Plan 71

Definition: Connection to containment seal cavity plugged for purchaser's use

Advantages: Ports available for future use

Disadvantages: Ports may be left plugged and provide no benefits

**FIGURE 53**

Plan 72

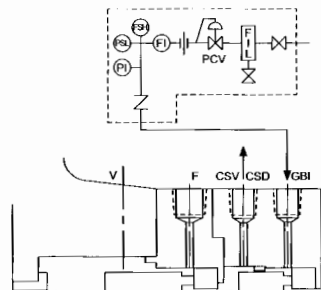


FIGURE 54

Definition: Supply of low-pressure buffer gas to the containment seal cavity of a dual unpressurized seal, requires use of Plan 72 or 74.

Advantages: Dilute and remove process fluid from containment seal cavity, minimize process emissions to atmosphere

Disadvantages: Requires external buffer gas and gas control panel

Plan 74

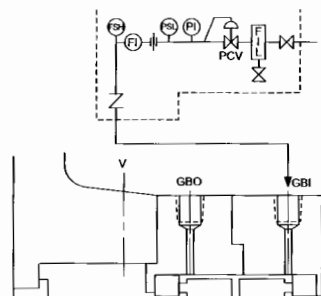


FIGURE 55

Definition: Supply of high-pressure barrier gas to the barrier cavity of a dual pressurized seal

Advantages: Provide environment for dual gas seals, prevent process leakage to atmosphere

Disadvantages: Requires external buffer gas and gas control panel, pressure must be maintained even during standby conditions

(Continued)

TABLE 4 Seal piping plans (Continued)

Plan 75

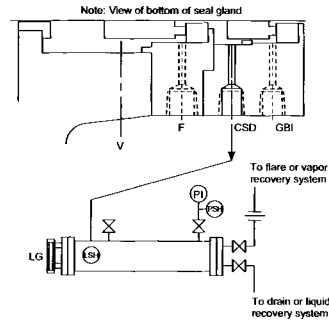


FIGURE 56

Definition: External drain from containment seal cavity to collect and monitor liquid or mixed phase process leakage from inner seal, may be used alone or with a Plan 72

Advantages: Allows monitoring of both liquid and vapor phase leakage, provides alarm for inner seal failure, directs process leakage to flare, drain, or recovery system

Disadvantages: Leakage which will solidify or crystallize under atmospheric conditions may require system to be heat traced

Plan 76

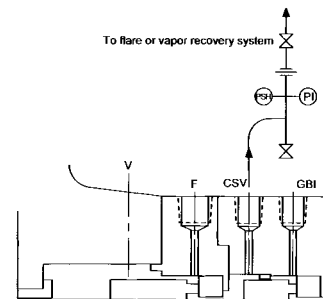


FIGURE 57

Definition: External venting of containment seal cavity to collect and monitor vapor phase process leakage from the inner seal, may be used alone or with a Plan 72

Advantages: Allows monitoring of both liquid and vapor phase leakage, provides alarm for inner seal failure, directs process leakage to flare or vapor recovery system

Disadvantages: Requires a high leak rate to alarm system

Other Piping Plans While the list of defined plans covers most applications, there will always be applications that require modifications to defined plans or require new methods of piping the seal system. Historically, users and seal OEMs would use terms like Plan 53 modified to document that the installed plan is not according to the stand definition. Other users and seal OEMs have come up with additional plan numbers that sound similar to the standard plans. There is great danger in this since there is no accepted definition and different users may use the same description to define different plans. If the plan offered is not in accordance with the standard plans defined in API 682 or ASME B73, they should be accompanied by a complete description of the function and operation of the plan as well as a P&ID to define the offering.

Auxiliary Equipment Several of the standard piping plans require the use of auxiliary equipment. This equipment includes reservoirs, seal coolers, gas control panels, cyclone separators, and other devices, which help contain and condition the fluid in the piping plan. There are no standardized requirements for these auxiliaries other than found in API 682. In many cases, individual end users have company specifications that will need to be followed.

RESERVOIRS Reservoirs (Figure 58) are used in Plan 52 and Plan 53 systems. The purpose of a reservoir is to maintain a supply of buffer or barrier fluid for the seal plan and provide temperature control through the use of internal cooling coils if required. Reservoirs are commonly supplied with capacities from 8 L (2 gal) for small low duty pumps to as large as 76 L (20 gal) for large high temperature applications. The most common sizes have 12 L (3 gal) and 20 L (5 gal) capacities. These are commonly supplied with features such as sight glasses to monitor fluid levels, pressure indicators to locally monitor pressure, and level switches to indicate a change in level.

Reservoirs are constructed from pipe and are available with either fixed or flanged ends. The pipe size and wall thickness dictates the pressure rating for the reservoir. API 682 specifies that 12 L (3 gal) reservoirs be constructed from DN 150 (6 in) schedule 40 pipe and 20 L (5 gal) reservoirs use DN 200 (8 in) schedule 40 pipe. Other sizes and wall thickness may be required for specific applications. API 682 also provides guidance on the sizing and location for all connections and ports to the reservoir.

SEAL COOLERS Seal coolers (Figure 59) are sometimes referred to as heat exchangers. While they do certainly serve the same function, seal coolers are designed to provide effective cooling to low volumes of fluid while offering low resistance to flow as required in many piping plans. Many plants require that heat exchangers comply with standards with such as TEMA, which is inappropriate for the size coolers required for seal systems. Seal coolers are therefore specifically designed for the requirements of mechanical seal systems. Seal coolers are most commonly used in Plan 21 and Plan 23 piping plans.

Seal coolers are required to have a low-pressure drop to insure adequate circulation in Plan 23 systems. Seal coolers should also be designed and mounted to promote thermosiphoning in standby services. The systems must also be ventable and drainable on both the process and cooling water side.

GAS CONTROL PANELS Dual gas seals (Figure 60) require the supply of a barrier gas to pressurize the seal assembly (Plan 74). Some containment seals will also use a buffer gas sweep to improve seal performance (Plan 72). In each of these cases, the gas provided must be conditioned, controlled, and monitored to insure successful operation. Gas seal control panels are designed to serve these functions. A typical panel will include a coalescing filter, pressure regulator, flow meter, flow switch, pressure switch, pressure indicator, and check valve. The list of components may vary depending upon the installation needs.

Piping Practices While piping plans are described in schematics and P&IDs, the success of a plan in an actual application is a function of how the plan is designed and constructed. This includes the size and type of interconnecting piping or tube, the routing of the piping, the use of fittings, and the mounting of auxiliary equipment. In many piping



FIGURE 58 Typical seal reservoir

plans, the flow is generated by the differential pressure in the pump. In other cases, system flows are generated by pumping rings on the seal so the system design becomes critical and must reduce friction losses. There are general guidelines that can help guide the user in creating a successful application.

The most critical piping systems involve the connection of the seal to an auxiliary such as a reservoir or seal cooler. Piping should be designed to minimize the length of the piping runs. This requires that auxiliary be located close to the seal and at the correct elevation. The actual piping or tubing should run directly from the seal to the auxiliary with a minimum of bends, elbows, or fittings. The piping should be continuous sloping upward from the seal to the auxiliary to aid in venting or draining of the system. Piping or tubing sizes should be large to minimize friction losses. An example of good and bad piping systems is shown in Figure 61.

**FIGURE 59** Typical seal cooler

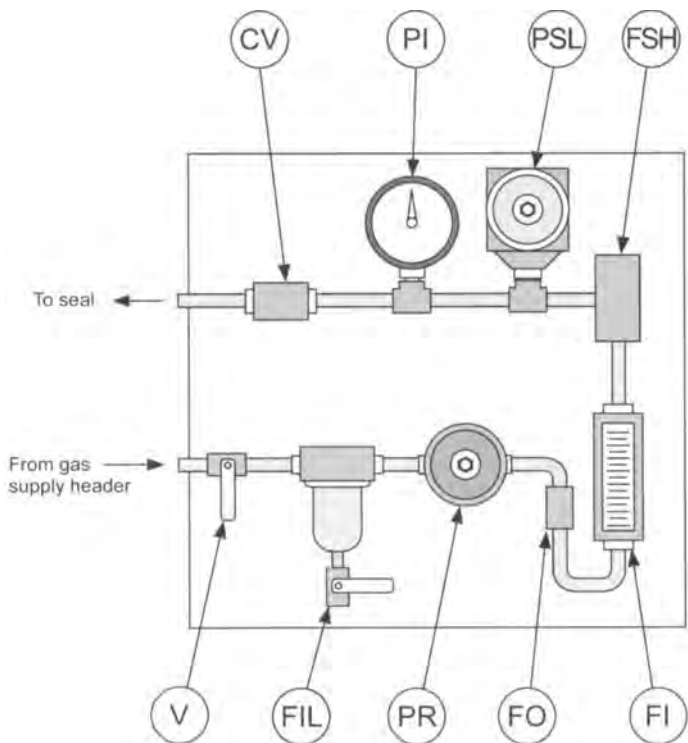
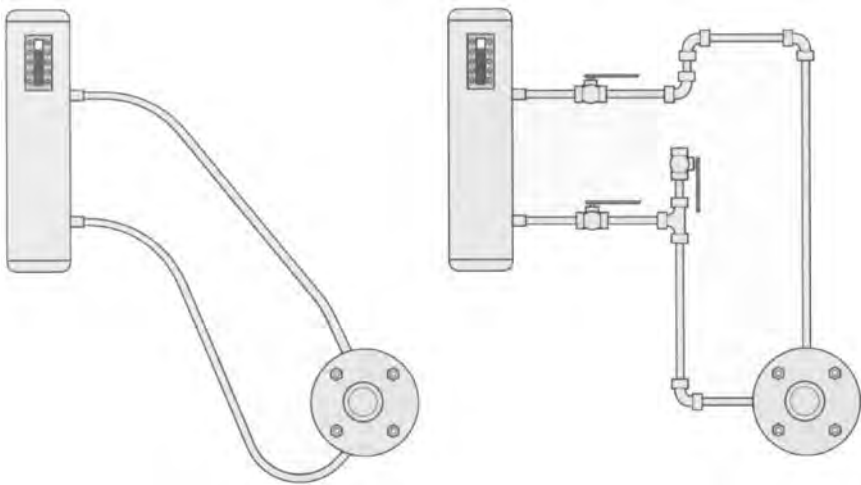


FIGURE 60 Typical gas control panel



Good piping practices

Bad piping practices

FIGURE 61 Examples of good and bad piping practices

SUMMARY

Mechanical seals are a critical component in modern centrifugal pumps. They can provide years of reliable service and meet the most stringent emissions regulations. They are, however, highly engineered devices that require careful selection and operation to achieve these results.

The following cutaway views illustrate some of the available options discussed.

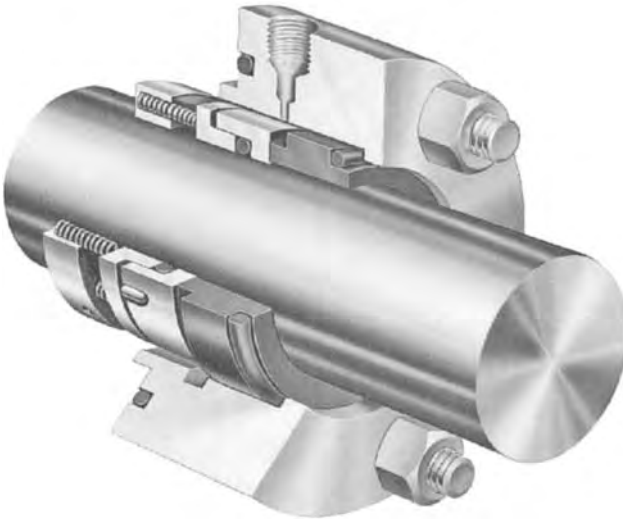


FIGURE 62 Component pusher seal (Flowserve Corporation)

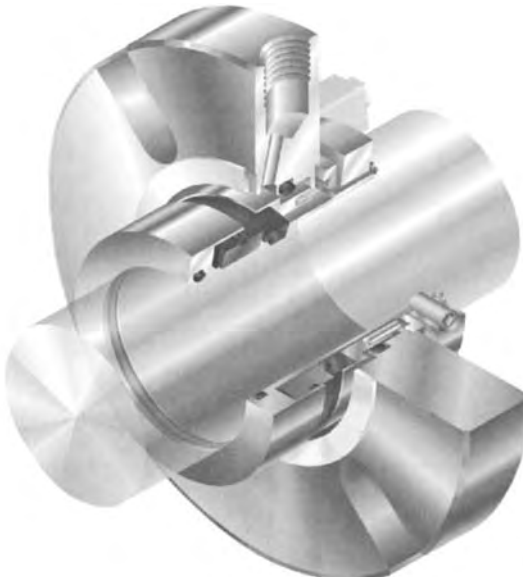


FIGURE 63 Cartridge stationary pusher seal (Flowserve Corporation)

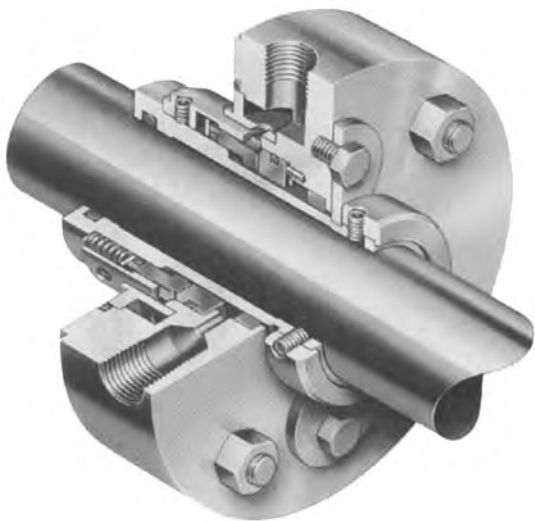


FIGURE 64 Cartridge rotating pusher seal (Flowserve Corporation)

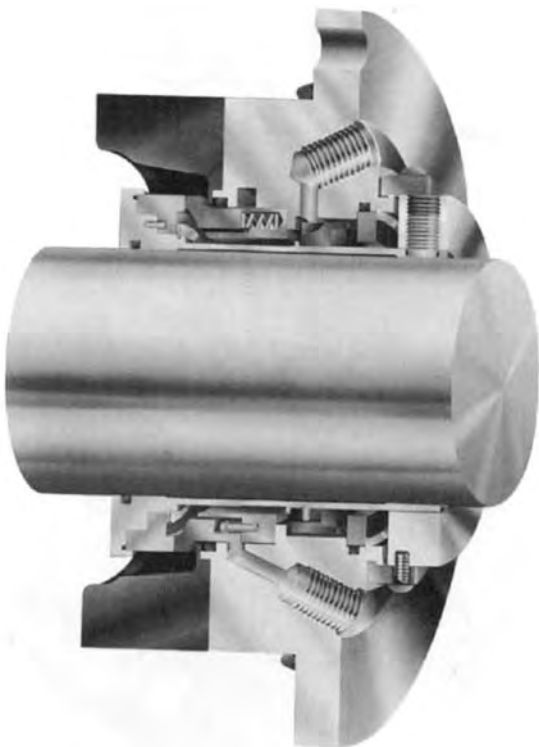


FIGURE 65 Slurry seal (Flowserve Corporation)

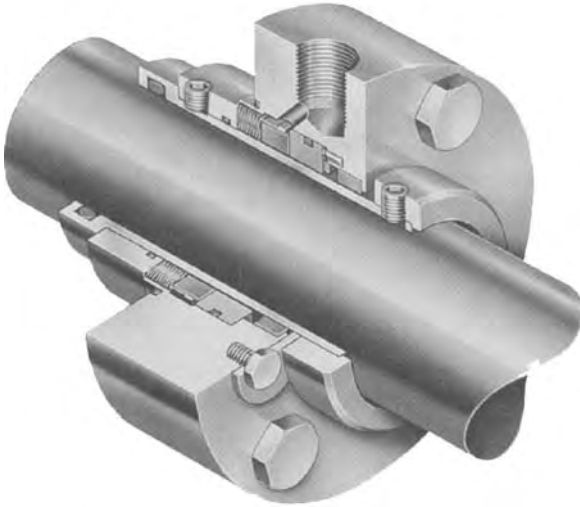


FIGURE 66 Cartridge rotating bellows seal (Flowserve Corporation)

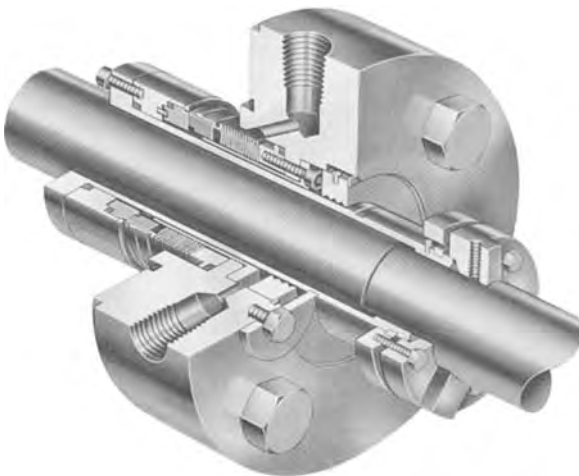


FIGURE 67 Cartridge stationary bellows seal (Flowserve Corporation)

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SECTION 5.3

CENTRIFUGAL PUMP INJECTION-TYPE SHAFT SEALS

ROBERT D. ASHTON
TIMOTHY L. WOTRING

Injection-type shaft seals (sometimes called *packless stuffing boxes*) are designed to control leakage from hot-water pumps. Cool water is injected into each seal to either suppress or regulate the hot leakage, which would otherwise flash up upon reaching the outside of the pump.

Injection-type shaft seals provide high reliability and yet require little maintenance. They are used primarily in power plant boiler-feed and reactor-feed pump applications where shaft peripheral speeds are high (3600 rpm and up) and pumping temperatures are greater than 250°F (120°C). Under these conditions, conventional packing or mechanical seal-type stuffing boxes may not be suitable or desirable.

Injection shaft seals are either *serrated throttle bushings* or *floating ring seal* designs that regulate the flow, temperature, and pressure of the controlled leakage. The flow of the cool injection and of any hot water in the seals depends on operating pressures, but it is restricted by close seal clearances and is regulated by injection control valves.

The operating temperature of the seals is controlled by allowing cool injection water to surround the outside of the seal. Ports in the seal enable the cool injection water access to the shaft to either overcome or mix with hot water in the seal so that the resulting seal leakage is cool.

The seal designs must provide sufficient pressure breakdown between the pump suction or balance device chamber pressures inside the pump and the atmospheric conditions outside the pump. Upon reaching the outside of the seal, the cool leakage is piped away by gravity drain for eventual return to the power plant feedwater system.

SERRATED THROTTLE BUSHINGS

The construction of a serrated (sometimes called *labyrinth* or *grooved*) bushing (see Figure 1) varies from one pump manufacturer to another. However, the serrated designs basically involve a rotating shaft running with a reasonably small clearance, such as 0.002 to

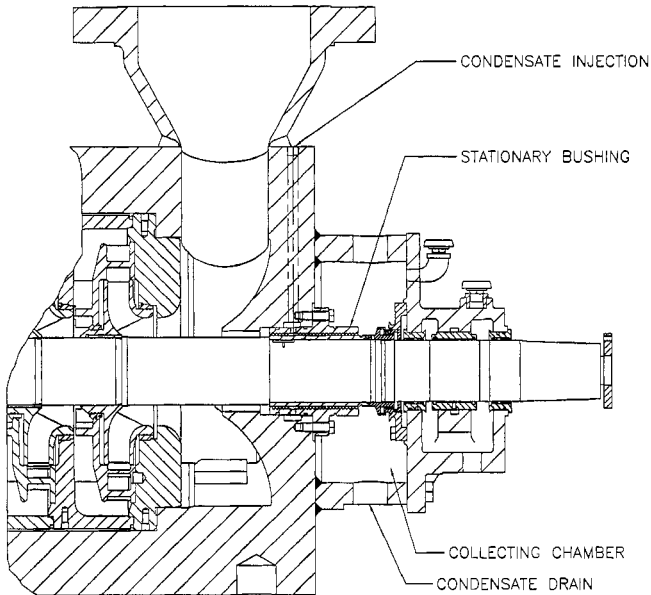


FIGURE 1 A cutaway view of a modern double-case boiler feed pump showing serrated throttle bushing shaft seals (Flowsolve Corporation)

0.003 in/in (0.002 to 0.003 mm/mm) of the shaft diameter within a solid stainless steel (typically 12% chrome) stationary bushing installed in the pump casing end cover. Grooved serrations are applied to the hardened stainless steel rotating surface (usually a shaft sleeve) or to the stationary bushing or to both rotating and stationary surfaces to effect a high throttling action or stuffing box leakage reduction.

The serrations create a more effective pressure breakdown than a smooth axial surface and enable increased running clearances to let foreign particles, such as grit, pass through or flush out in the grooves. The increased clearances also enable more tolerance for any displacement between rotating and stationary components that might result from assembly misalignment or from distortions induced by transient temperatures. The grooved running surface area at clearances is greatly reduced relative to that of a smooth axial surface, and therefore possible metal surface contact between rotating and stationary parts is reduced during transient operations.

Seal leakage could be reduced by decreasing the running clearance in the injection seal, but the smaller clearance would limit the capability of the serrated seal to conform to radial shaft movement. Reduced clearances in any injection seal design may lead to problems with thermal distortions caused by a loss of available cool injection water and may also increase galling by particles trapped in the seal running fits.

FLOATING RING DESIGN

A segmented throttle bushing made of many floating rings will enable conformity with the radial shaft movement and smaller running clearances to reduce seal leakage. The construction of a floating ring stuffing box (see Figure 2) varies from one pump manufacturer to another, but the unit basically consists of a stack of hardened, martensitic stainless steel rings (typically 17% chrome), instead of a stationary bushing. The rings run against a

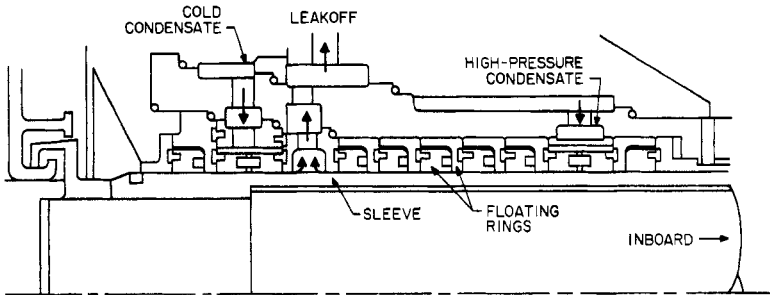


FIGURE 2 Floating ring seal design with shaft sleeve. Pegs prevent rotation of spring-loaded rings but let them float radially (From *Power*, August 1980, McGraw-Hill, New York, copyright 1980)

smooth, hardened, martensitic stainless steel (typically 12 to 17% chromium) rotating shaft sleeve surface.

The separate rings are all contained within a housing installed in the pump casing end cover. Each ring is loaded against an adjacent stainless steel ring spacer so that a stationary seal is produced in the axial direction, but the ring is still able to move radially with the shaft. An axial loading of the rings is provided by hydraulic pressure during operation and by springs during idle pump periods. The rings are locked against rotation usually by pegs and/or a slot arrangement.

A small radial clearance, usually 0.001 to 0.0015 in/in (0.001 to 0.0015 mm/mm) of the shaft diameter, is provided between the rings and the shaft sleeve to provide an adequate throttling of seal leakage across the limited number of seal rings. The length of each ring varies with the diameter of the injection-type seal but is generally about 0.50 in (13 mm). The radial clearance enables a reduced injection flow but increases the sensitivity of the seal to foreign particles in the seal leakage.

The individual seal rings are permitted to move radially (float), finding an equilibrium running position relative to the shaft. The capability to float up to 0.016 in (.4 mm) radially increases the tolerance for shaft misalignment as the shaft passes through the seal area. The multiplicity of seal rings further reduces the effect of any angular displacement between the rotating and stationary components that might arise from errors in the original assembly or from distortions caused by temperature changes during pump load variations.

Some injection-type shaft seal designs use a combination of serrations and floating rings in the same stuffing box. For either type of seal, careful maintenance during the assembly or disassembly requires accurate alignment of the rotating and stationary components as well as careful handling to avoid scratching the components. Cleanliness is of utmost importance with shaft seals.

CONDENSATE INJECTION REQUIREMENTS

Because boiler-feed and reactor-feed pumps normally have high feedwater temperatures, 250 to 500°F (120 to 260°C), with consequent vapor pressures higher than external atmospheric conditions, the seal leakage must be cooled to avoid flashing in the stuffing box or outside the pump. This cooling is usually accomplished by injecting cold condensate from a power plant condensate pump directly into the seal to cool the stuffing box components as well as the seal leakage.

The amount of condensate injection water required will depend on several factors, including: (1) whether the design is a serrated or floating ring; (2) the type of seal control system; (3) the diameter, clearance, and rotating speed of the running fit; and (4) the internal pump pressure and temperature. A typical example of stuffing box flows would be a

5500-rpm, boiler-feed pump with a temperature-controlled seal system (described later) and a serrated 5-in (127-mm) diameter running seal with about 0.015 in (0.38 mm) of diametrical clearance. The approximate flows would be as follows:

1. Total injection per seal: 5 to 15 gpm (0.3 to 1.0 L/s)
2. Leakage from internal pump to seal: 0 to 5 gpm (0 to 0.3 L/s)
3. Drainage from each seal: 8 to 20 gpm (0.5 to 1.3 L/s)

Given a seal with a floating ring design (having typically half the clearance of seals with a serrated bushing design), the previous conditions would require only 50 to 60% of the injection water needed by the serrated bushing design with the larger clearance.

During pump standby periods, higher injection and leakage flows are required because of reduced seal throttling caused by low or zero speed, by high internal pump pressures, or by a pump balance device that induces higher internal leakage during standby. The injection flows to both stuffing boxes on a multistage pump may not be the same as a result of pump balance action at one end of the pump.

Cold condensate, 85 to 110°F (30 to 45°C), is usually available from the power plant condensate pump or secondary condensate booster pump discharge. The variation in condensate supply pressure relative to internal boiler-feed pump pressure makes it necessary to use injection control systems for satisfactory seal operation.

Condensate injection systems vary to accommodate the wide range of pressures and temperatures found in different power plant feedwater system layouts. The four basic types of injection flow systems are: (1) manual, (2) differential pressure-controlled, (3) differential temperature-controlled, and (4) constant drain temperature control.

Manual System Manual flow control requires setting and readjusting a valve by hand for each stuffing box each time a change occurs in pump operating conditions due to varying plant loads. Although it is possible to use such a control system, it is not usually recommended because of the inability to automatically compensate for rapid changing conditions.

Differential Pressure-Controlled System The differential pressure-controlled shaft seals operate as cold condensate maintained at a pressure greater than boiler-feed pump suction is injected into the central portion of the seal. By maintaining an injection pressure above the pump seal chamber pressure, a small portion of this injection water flows into the pump proper, and most of the injected water flows out of the seal into a collection chamber adjacent to the pump bearing bracket. The leakage in the collection chamber, which is vented to the atmosphere, is drained by gravity for an eventual return to the main feedwater system.

The differential pressure-controlled system has a pneumatic injection control valve. This valve in the condensate injection line is governed by an air signal received from a differential pressure control monitor that maintains the preset pressure differential, 10 to 25 lb/in² (70 to 170 kN/m²), between the injection pressure and the internal pump pressure. This system is not always favored, however, because of a feedback instability tendency resulting from operating pressure changes affecting the valve position, which changes pressure, and so on. Another unfavorable factor for hot-water service is the introduction of cold condensate into a hot pump at all times, including pump idle periods, affecting pump prewarning conditions.

Differential Temperature-Controlled System The differential temperature-controlled system operates by maintaining a seal drain temperature differential of 25°F (14°C) above the injection water temperature. The temperature differential ensures a continuous out-leakage of hot boiler feed water. The hot out-leakage is desirable, ensuring that cold water cannot be injected into the pump and thus eliminating thermal distortions that occur with the differential pressure-controlled system. The valve controllers are identical to the controllers used for the differential pressure-controlled system.

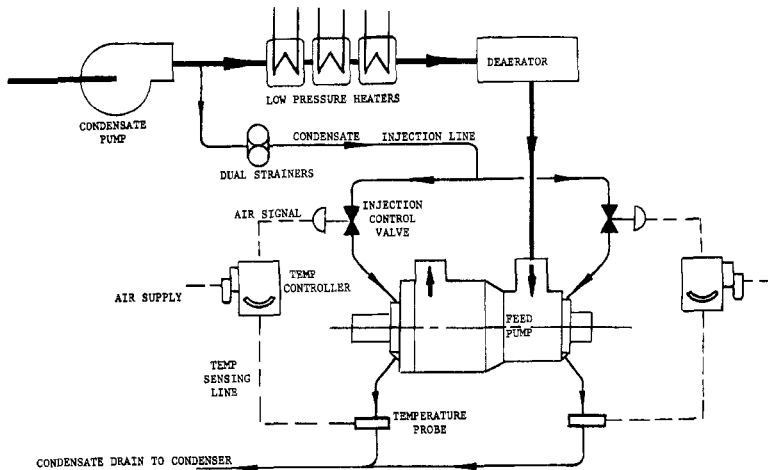


FIGURE 3 Temperature-controlled system regulating condensate injection to feed pump shaft seals

Constant Drain Temperature-Controlled System The constant drain temperature-controlled seal system (see Figure 3) controls hot out-leakage by throttling the injection flow to maintain a preset seal drain temperature. This system has a temperature-sensing probe in each seal drain line. Each probe is connected to an indicating temperature controller, which provides an air signal to a pneumatic control valve in the condensate injection line for control of the seal injection flow rate. Electronic control systems are often used where thermocouples or RTDs sense drain temperatures. The signal output is then processed through an I/P controller that adjusts the position of the pneumatically operated throttle valve as required.

The cold condensate is injected into the stuffing box central portion and allowed to mix with hot water entering the seal from inside the pump. The drain temperature is maintained at a preset 140 to 150°F (60 to 66°C) to preclude flashing in the stuffing box or in the drains. Note that with this system some hot water enters the seal. Therefore, cold condensate does *not* enter the hot pump and does not adversely affect pump warming conditions, especially during extended idle periods. The required condensate injection pressure is at least equal to the internal stuffing box pressure plus interim frictional loss between the condensate supply source and the point of hot-water mixture. Note that this system *may* enable satisfactory operation even when the condensate supply pressure is nearly equal to boiler-feed pump suction pressure. In addition to providing a rapid response to variations in operating pump conditions, this type of control will always supply just enough injection water to maintain the recommended drainage temperature.

Intermediate Leakoff System The intermediate-leakoff shaft seal system (see Figure 4) has many variations but basically uses a bleedoff from a central portion of the stuffing box. This system may be used to reduce internal stuffing box pressure if high boiler-feed pump suction pressure exists. To create a positive leakoff flow, the intermediate bleed-off flow is piped back to a plant feedwater system low-pressure point, such as a plant condenser, heater, or booster pump suction where the pressure is less than the boiler-feed pump suction pressure. However, the back pressure of the leakoff destination must be above the bleedoff vapor pressure to suppress flashing in the leakoff lines. This back pressure may be the leakoff destination pressure or may be created by an orifice or a valve.

Cold condensate injection into the stuffing box is controlled by a pressure differential monitor maintaining a preset pressure above the bleedoff pressure (refer to Figure 4). A stuffing box drain temperature control can also be used. Note that the cold condensate

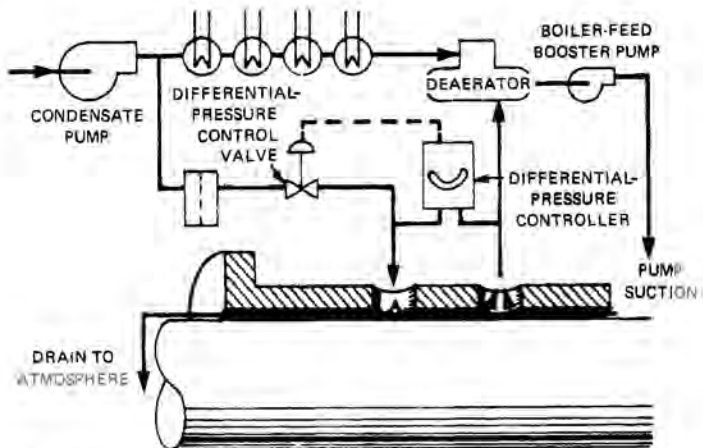


FIGURE 4 Typical intermediate-leakoff shaft seal system (From *Power*, September 1980, McGraw-Hill, New York, copyright 1980)

injection pressure need not equal or overcome a high feed pump suction pressure. The condensate injection temperature must still be 85 to 110°F (30 to 45°C) to keep the drain temperature below the flashing condition. Condensate injection shaft seals without an intermediate bleedoff but that are subject to suction pressures in excess of about 250 lb/in² (1725 kN/m²) must be extremely long for a proper pressure breakdown. Longer shaft seals require thicker pump case end covers, affecting pump cost, and a longer rotating element, which could adversely affect rotor dynamics. The intermediate-leakoff shaft seal is effective where there is high feed pump suction pressure imposed by boiler-feed booster pumps or in a closed feedwater system with no deaerating open heater, wherein a condensate pump discharge can be fully imposed on the feed pump at low plant loads.

In plant systems with feedwater heaters between a booster pump and a feed pump, a high pressure condensate from the cooler booster pump is injected into the seal to enable a cooler intermediate leakoff to help prevent flashing (refer to Figure 2).

INJECTION SOURCES

Many power plant feedwater systems with deaerating direct-contact heaters (open cycle) usually have the boiler-feed pump drawing water directly from the deaerator. These open cycles with constant-speed condensate pumps always have cold condensate supply pressures in excess of boiler-feed pump suction pressure because of the increased available condensate pump head at low system flows and the interim system frictional loss at high system flows.

If a boiler-feed booster pump or a closed feedwater system with no open heaters and resultant higher boiler-feed pump suction pressures is used, the pump manufacturer may elect to require condensate injection seal water booster pumps or may use an intermediateleakoff packless shaft seal with a condensate injection overcoming only the intermediate leakoff pressure. If the condensate pumps are variable-speed units that enable the condensate injection pressure to drop to an equal feed pump suction pressure at low loads with little or no feedwater system frictional drop, then condensate injection seal water pumps are required. In this situation, at least one pump manufacturer offers an optional pumping ring configuration in their packless seal that can increase the seal water pressure in the stuffing box to overcome any pump suction pressure.

AUXILIARY EQUIPMENT

The condensate injection piping should be conservatively sized based on the maximum injection flow requirements to obtain a low-pressure drop between the injection source and the seal injection control valve. These control valves may be equipped with limit stops to prevent full closure and enable a continuous cool injection to the seals under almost all operating conditions. In some installations, isolating lines are furnished around the valves to enable a continued injection flow even during control valve maintenance. The valves can also be designed to remain open during a failure, such as a loss of station air to the pneumatic controls, and to close only with an air supply. Injection control systems that are not properly maintained might result in cold water entering a hot pump. Should this problem occur while a boiler feed pump is in the hot standby mode or when turning gears, thermal gradients will occur, leading to contact among the close running fits within the pump. Proper maintenance and operation of the injection control systems is necessary to ensure reliable operation of the pump itself. The air supply filter regulators for each control must be furnished with relatively dry clean air at station supply pressure.

The condensate injection supply to the seals must be clear and free of foreign matter to prevent damage to stuffing box components. It is therefore necessary to install filters in the injection line prior to the control valve. To keep damaging fine mill scale, oxide particles, abrasives, and other materials from entering the small seal clearances, several pump manufacturers recommend 100-mesh (150-micron) dual strainers. If dual strainers with isolating valves are used, each filter can be cleaned without interrupting injection flow during pump operation. Pressure gages should be installed before and after each filter to permit the operator to monitor filter pressure drop. A differential pressure switch and alarm for each filter are preferable to alert the operator to clean the strainer when pressure drop becomes excessive.

The condensate injection shaft seals should always be filled with cool water before and during pump operation, even during reverse pump rotation. Some pump manufacturers stipulate that condensate injection *must* be continuous without any interruption during all operation modes.

The clearances in the condensate injection shaft seal may double over the service life of the internal wearing parts. With double clearances, the leakage will approximately double. This factor should be considered when sizing the return drain piping back to the plant condenser if frictional losses are to be kept to a minimum. The drain line should be pitched at least a quarter-inch per foot (20 mm/m). The collecting chamber at the pump stuffing box is vented to the atmosphere, and the only head available to evacuate the chamber is the static head between the pump and the point of return. This head must always be well in excess of the frictional losses (even after the leakage is doubled). Otherwise, the drains may back up, the collection chambers may overflow, and the adjacent bearing brackets may flood, with subsequent possible intrusion of water into the pump bearings and lubricating oil.

The seal collection chambers have especially large connections to assure proper drainage, provided no back pressure exists. Two types of condensate drain systems can be used to dispose of the drain coming from the collecting seal chambers. One system uses traps that are piped directly to the plant condenser if sufficient static head exists for positive drain flow. The second system collects the drain in a condensate storage tank into which various other drains (from other pumps shaft seals and so on) are also directed. As this vented storage tank is under atmospheric pressure, it must be set at a reasonable elevation below the pump centerline so that the static elevation difference will overcome frictional losses in the drain piping. A separate condensate transfer pump, under control of the storage tank liquid level control system, can then pump the condensate drains from the storage tank into the plant condenser. The storage tank should have its own overflow protection system that enables outside drainage if, for some reason, proper drainage cannot be achieved. For example, the top of the tank vent pipe should be below the pump centerline to help preclude the possibility of drainage backing up to the level of the pump seal collection chambers. Note that this storage tank should also be large enough for an adequate drainage collection to help prevent backups.

PACKLESS SHAFT SEALS WITHOUT INJECTION

The packless shaft seals that have been so successfully applied to boiler-feed, reactor-feed, and booster pump services are applicable to a number of other services. For instance, they are very suitable for cold-water condensate booster pumps and for high-pressure pumps applied to hydraulic descaling or hydraulic press work. In such services, there is no need to bring in an injection supply water to the breakdown seals (unless the pumped water is not clear and free of gritty material), because the water handled by the pump is already cold with no danger of flashing as it leaves the pump stuffing boxes.

C • H • A • P • T • E • R • 6

PUMP BEARINGS

SECTION 6.1

CENTRIFUGAL PUMP BEARINGS

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PUMP BEARINGS

A pump bearing system consists of bearings to carry radial loads and at least one bearing to carry axial or thrust loads. For horizontally mounted pumps, there are two basic pump rotor arrangements: the pumping element overhangs the bearing system (Figure 1) or the pumping element is between bearings (Figure 2). The function of bearings in centrifugal pumps is to keep the shaft or rotor in correct alignment with the stationary parts under the action of radial and transverse loads. Bearings that give radial positioning to the rotor are known as *radial* or *line bearings*, and those that locate the rotor axially are called *thrust bearings*. In most applications, the thrust bearings actually carry both axial and radial loads.

Loads on Bearings Centrifugal pump bearings are subject to radial and axial loads that result from internal pressures acting on the surface areas of the impeller. The resultant force acting perpendicular to the shaft is considered radial load and the resultant force acting along the shaft axis is considered axial load. Loads in a centrifugal pump vary considerably and are approximately proportional to the developed pressure for a given pump design. Pressure variations occur because of changes in liquid specific gravity, varying operating speed, changes in impeller diameter and at various operating points on the performance curve. Centrifugal pump bearings must be carefully selected to operate reliably when carrying any of these varying loads.

Pump designers utilize various techniques to balance radial and axial loads. Radial thrust loads are described in Section 2.2.1, along with the influence of impeller and casing design on the magnitude of radial unbalance. Axial thrust loads are also described in Section 2.2.1, along with methods of balancing thrust loads to reduce the residual unbalanced axial loads that must be carried by thrust bearings.

Types of Bearings Rolling-element bearings are used for the majority of pump applications. They were originally called “antifriction” bearings because the rolling elements “roll” (and theoretically don’t slide) such that friction is minimized. These bearings can be used

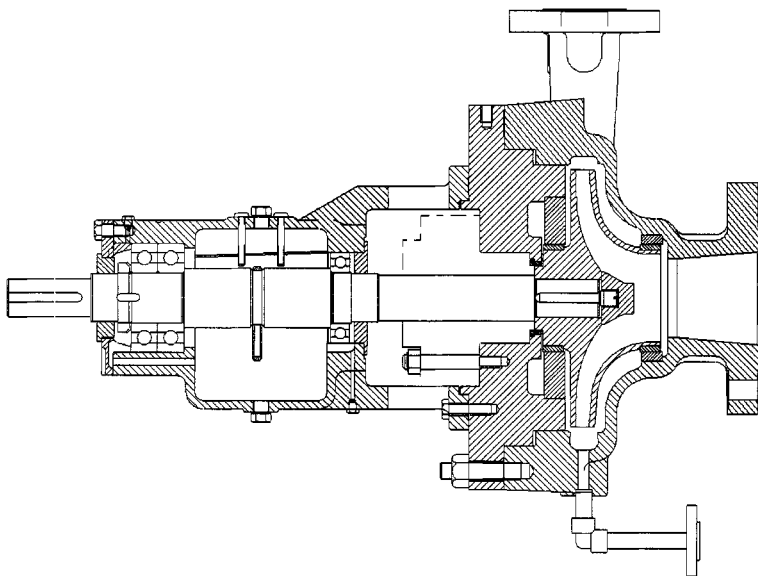


FIGURE 1 Overhung impeller pump and pumping element

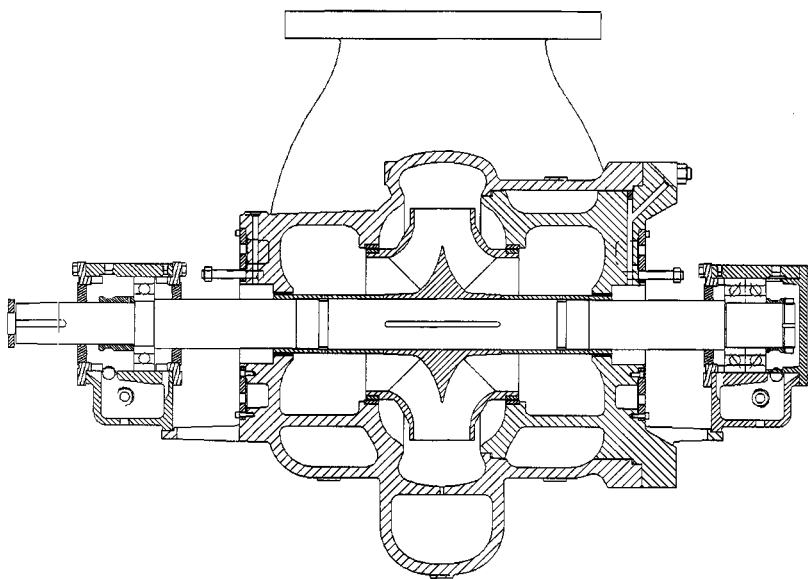


FIGURE 2 Between bearings pump and pumping element

to carry purely radial loads, both radial and axial loads, or purely axial loads, depending on their design and how they are applied or mounted (installed). Ball bearings and various types of roller bearing designs (cylindrical, spherical, and taper roller) are commonly used. Bearing manufacturers' catalogues usually provide all information necessary to properly select and apply these bearings.

Ball bearings come in many configurations, such as single and double row with various contact angles, to handle purely radial loads, combined radial and axial loads, and purely axial loads. Sometimes ball bearings are mounted in pairs (duplex) or triples (triplex) to increase load-handling capability or to provide axial thrust handling capability in both directions, as in the case of an angular contact bearing. The shape of the rolling element is that of a ball. Ball bearings are considered to be a point-contact bearing, meaning that the location where the load is transferred from the bearing race to the rolling element is a small circular area. Ball bearings have a relatively low load rating because the small area results in relatively high contact stress for a given load ($\text{stress} = \text{force/area}$).

Cylindrical roller bearings are used for purely radial loads for applications that require a higher load carrying capability than a ball bearing. These bearings are available only in single rows. The shape of the rolling element is that of a cylinder. Cylindrical roller bearings are considered to be a line-contact bearing because the contact between the bearing race and the rolling element is in the shape of a line. This shape provides a larger contact area that reduces the contact stress for a given load as compared to a ball bearing.

Tapered roller bearings accommodate both a radial and axial load. The shape of the rolling element is that of a truncated cone. These bearings are available only in a single row configuration but must be mounted in pairs since the axial setting or preload is critical to proper operation. They are also a line-contact bearing and, therefore, have a relatively high load carrying capability.

Spherical roller bearings also accommodate both radial and axial loads, though the radial load rating is considerably higher than the axial load rating. These bearings are available only in a double row configuration. The shape of the rolling element is that of a wooden stove barrel. The contact patch is ovalar in shape resulting in high load carrying capability.

When bearing loading or rotating speeds exceed, capabilities of a rolling-element bearing, hydrodynamic bearings are generally selected. Hydrodynamic sleeve (journal) bearings can only accommodate purely radial loads (see Section 6.1). The shaft rides on a film of oil as it rotates inside of a babbitt-lined sleeve. These bearings are a line-contact bearing and have a very high load carrying capability since the bearing contact area is quite large. Contact area is a misnomer since, during operation, the shaft is supported by a layer of oil (oil film) that further increases the "contact area." The bearings are called *hydrodynamic bearings* because, during operation, an oil "wedge" is formed between the shaft and the sleeve, which lifts the shaft away from the bearing metal. Thus "contact" between the shaft and the bearing metal does not actually occur during normal operation.

Tilting pad thrust bearings are also hydrodynamic bearings and can only accommodate purely axial loads. These bearings are normally used in conjunction with hydrodynamic sleeve bearings to accommodate the radial load. Axial loads from the shaft are transferred through a disk mounted on the shaft to the bearing. A compliment of pie shaped pads is mounted in the bearing housing facing each side of the disk to handle axial loads in either direction. These pads are allowed to pivot individually which allows an oil wedge to form on each of the pads, transferring the load to the housing.

Vertically suspended, magnetic drive, canned motor, and certain types of multistage pumps utilize internally mounted product lubricated sleeve bearings. These bearings are often made from rubber, bronze, carbon, or silicon carbide. The fluid flowing between the bearing and the shaft creates a dynamic centering effect, known as the Lomakin effect. Magnetic drive pumps, canned motor pumps, and some multistage designs also utilize product-lubricated thrust bearings or thrust balancing devices for axial thrust, whereas vertically suspended and horizontal multistage pumps normally handle axial thrust with externally mounted rolling-element or hydrodynamic thrust bearings.

Clearance and Preload Single-row deep-groove ball bearings are available with several classes of radial internal clearances. As standard, they are manufactured with Normal

radial internal clearance, but a popular alternative for severe duties, especially applications involving pumping temperatures higher than ambient, is a larger than Normal, or C3, radial internal clearance. Pumps designed for petroleum, petrochemical, and natural gas industries, such as covered by API Standard 610 (ISO 13709) require a C3 internal clearance as standard. To meet this requirement, bearings should be specified to be in accordance with ISO 5753, Group 3. Greater internal clearances may reduce the temperature rise of the bearing lubricant and allow the bearings to run cooler under load. However, there may be a trade-off in that higher pump vibration velocities may occur with greater bearing internal clearances. It is important that the temperature rise of the bearings under maximum load conditions does not exceed the capability of the lubricant to properly perform and maintain the required viscosity, and thus film thickness, within the bearings.

In centrifugal pumps, bearing preload is a term applied to two bearings mounted in a pair to handle both radial and axial thrust loads. Tapered roller bearings and angular contact ball bearings are typically mounted to a predetermined setting or preload. This setting may be necessary to maintain proper contact of the rolling elements with the bearing races during operation to avoid skidding of the unloaded bearing in the pair. For angular contact ball bearings, the setting ranges from various clearances (non preloaded) to various amounts of preload. The determination of clearance or preload is based on a balance between providing sufficient clearance to operate at a low temperature and sufficient contact to avoid skidding of the rolling elements in the lightly loaded bearing in the pair. The pump manufacturer is responsible for determining the amount of preload or clearance necessary to meet the requirements of the standard or specification to which the pump is manufactured and sold.

It should be noted that bearing preload occurs in two other ways: (a) when the bearings are mounted on the shaft with an interference fit (as required by many standards, particularly for dynamic loading applications), and (b) when, for hot liquid pumping applications, the shaft heats more than the bearing races. Also, bearing preload determined by interference fit must be evaluated as the bearings may heat up in service and it may either decrease or increase.

The balance between bearing preload and clearance is not always easy to achieve. To address instances in ball bearings where ball skidding is observed or can be anticipated—in the unloaded bearing of the duplex-mounted pair, at least one bearing manufacturer provides a matched set of bearings consisting of one 40° contact angle bearing paired with one 15° contact angle bearing. Properly selected and mounted, the 15° contact angle bearing should be less susceptible to ball skidding when operating unloaded (when the axial thrust is in a direction such that it is all absorbed by the 40° contact angle mate). Care must be taken with this bearing type to ensure the direction of axial thrust is known and the bearing mounted accordingly. Also, it must be realized that the 15° contact angle bearing has less axial load capability than the 40° contact angle bearing, and bearing selection and sizing must account for this difference in capability. If axial bearing loads in both directions can be of the same magnitude, this option may not be feasible.

Retainers Bearing retainers or “cages” are used to guide and separate the rolling elements as they revolve in the bearing races. Bearing retainers are available in various materials and styles depending on bearing type and application. Pressed steel, pressed brass, machined bronze, and nonmetallic retainers are commonly used. Pressed steel and nonmetallic retainers are successfully used on single and double row ball bearings, cylindrical roller bearings, and tapered roller bearings. Pressed brass is used successfully on low speed angular contact bearings. Machined brass is successfully used on spherical roller and high-speed angular contact bearings. Spherical roller and high-speed angular contact bearings require a strong retainer with closely fitted rolling element pockets to properly guide the rolling elements.

Nonmetallic retainers should not be used for high temperature applications and are not allowed by some heavy-duty pump standards such as API 610 (ISO 13709).

Arrangements Each pump shaft is supported by two radial bearings and a thrust bearing; the thrust bearing is normally a double acting thrust and radial bearing. For pumps

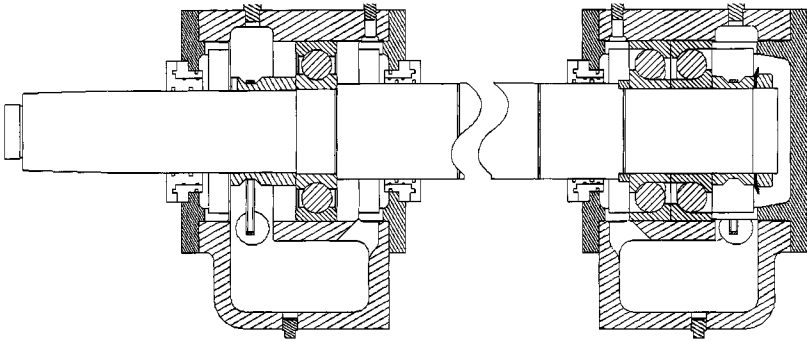


FIGURE 3 Rolling-element radial and thrust bearings

with externally mounted bearings, bearings are usually applied in one of the following arrangements: rolling-element radial and thrust (Figure 3), hydrodynamic radial and rolling-element thrust (Figure 4), or hydrodynamic radial and thrust (Figure 5). Rolling-element bearings are normally provided on simple machines when the application of the bearing is within the speed and load limitations of the bearing. Hydrodynamic bearings are better suited for higher speeds and loads and are, therefore, used on larger, more critical machines. Though both rolling-element and hydrodynamic bearings have proven reliability, rolling-element bearings tend to fail more rapidly once failure has begun. Even when worn, hydrodynamic bearings will seldom fail suddenly, which allows for a planned shutdown to rebuild the pump.

Table 1 shows selection criteria commonly used for the various bearing arrangements (Reference 1).

Lubrication Rolling-element bearings may be lubricated with grease or oil. Sealed bearings are available in certain bearing types that are pregreased from the bearing manufacturer, but most industrial pumps utilizing grease lubrication are regreasable. Grease is an acceptable lubricant for many applications since grease will protect the bearing surfaces from oxidation and regreasing purges contaminants from the bearing. Grease-lubricated bearings tend to operate at a higher temperature than oil-lubricated bearings, particularly until excess grease is purged from the bearing after regreasing.

Oil lubrication is also a widely used method of lubrication. There are various methods of oil lubrication: flood oil, oil splash, and ring oil, each using an oil sump that is integral with the pump bearing housing, and oil mist.

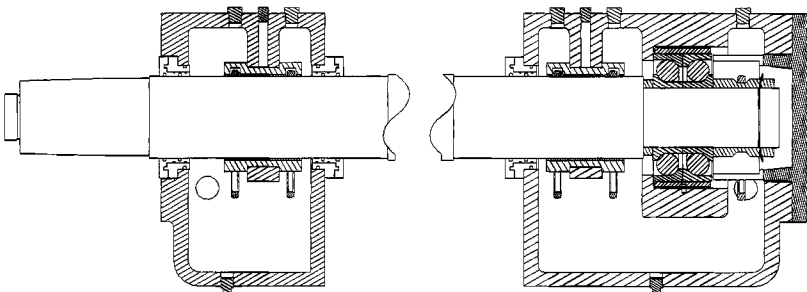


FIGURE 4 Hydrodynamic radial and rolling-element thrust bearings

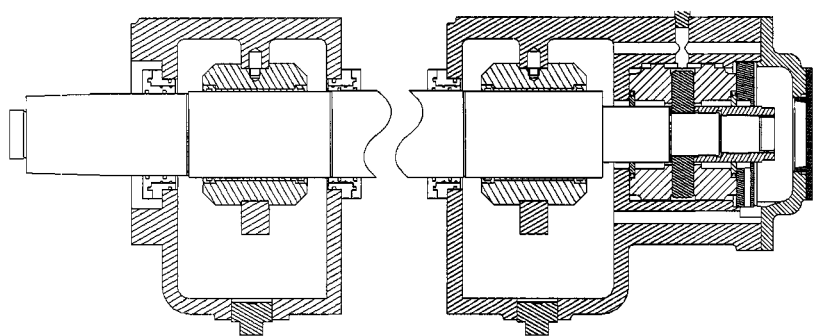


FIGURE 5 Hydrodynamic radial and thrust bearings

The oil level in a flood oil system is high enough to contact the rolling elements with the oil setting normally halfway up the bottom most rolling element. If two bearings are in the same housing such as for an overhung pump, the radial and thrust bearing must be the same size or nearly the same size in a flood oil system. Oil is delivered to the bearing as the rolling elements revolve and dip into the oil.

An oil splash system is similar to a flood oil system except the oil level is below the rolling elements and there is a flinger or thrower, normally a disk, mounted to the shaft inside the bearing housing. The outer portion of the thrower dips into the oil as the shaft rotates and creating a stream of oil and oil droplets that “splash” around inside the bearing housing and into the bearing. Bearings in this arrangement normally operate at a lower temperature because the bearing is not flooded with oil and because there is greater oil circulation thereby increasing cooling area. Since the oil flinger is rotating at the same speed as the shaft, foaming of the oil can be a problem for higher speed applications.

Ring oil systems are similar to splash systems except, instead of an oil flinger, an oil ring is used. The oil ring normally rides in a groove in the shaft and dips into the oil in the sump. As the ring rotates, oil is drawn up and splashes around the inside of the frame and into the bearing. The advantage of the oil ring is that it operates at a slower speed than the shaft and does not tend to cause oil foaming. Some bearing housings utilize troughs or channels that catch the oil stream coming from the oil rings and positively direct the oil to the bearings.

Oil mist systems are widely used and are particularly suitable for high-speed rolling-element bearings. There are two basic types of oil mist systems, pure oil mist and purge

TABLE 1 Bearing arrangement selection

Condition	Type and Arrangement of Bearing
Radial and thrust bearing speed and life are within limits for rolling-element bearings	Rolling-element radial and thrust
Radial bearing speed or life is outside limits for rolling-element bearings and thrust bearing speed and life is within limits	Hydrodynamic radial and rolling-element thrust, or hydrodynamic radial and thrust
Radial and thrust bearing speed or life is outside limits for rolling-element bearings	Hydrodynamic radial and thrust

oil mist. These systems continuously provide fresh oil in the form of a mist from a centralized oil mist generator. Experience has shown that bearing housings equipped with oil mist lubrication systems result in extended bearing life. The bearings tend to run cooler, seldom fail because of lack of lubrication, and inherently provide a bearing environment that is free of the such common contaminants as dirt, dust, and water.

Pure oil mist systems do not utilize the oil sump other than to direct the mist and any condensed oil to a drain once it has passed through the bearing(s). Any internal troughs or passages, such as oil returns for non-oil mist systems, must be plugged to avoid short-circuiting the mist from flowing through the bearing(s). Normally one oil mist reclassifier is used at each pump bearing housing to ensure that a proper oil mist and air mixture is delivered to each bearing, even though there may be more than one inlet fitting on the bearing housing—one for each bearing to be lubricated. Oil mist injectors are normally located between the bearing and bearing housing end cover and the mist is forced to pass through the bearing to either a vent or a drain that is located on the opposite side of the bearing. If a directed oil mist stream is required to ensure that mist circulates through the bearing (e.g., to combat “windage” from the bearing), directional reclassifiers may be installed adjacent to those bearings. This is usually required only for larger or high-speed angular contact bearings because they can act like mini pumps that can push air against the oil mist attempting to enter the bearing.

Purge oil mist systems are used in conjunction with sump oil, splash oil, or ring oil systems. A central oil mist connection is provided in the bearing housing for injecting the oil mist and provide a low-pressure oil-saturated environment for the bearings. A reclassifying fitting is located in this feed line to ensure a proper oil mist. This low pressure created by the oil mist within the bearing housing tends to prevent water, moisture, dirt, dust, and other contaminants from entering the bearing environment. In addition to the oil mist inlet connection, a connection is provided in the bearing housing to draw off excess oil that accumulates from condensed oil mist during operation in order to maintain a constant oil level in the sump. After the oil mist passes through the bearings, the residual mist that is not condensed escapes through the bearing end covers.

Hydrodynamic bearings are lubricated with oil. A hydrodynamic radial and rolling-element thrust bearing arrangement typically utilizes oil ring lubrication. There are grooves in the upper half of the sleeve bearing that allow the oil ring to ride on the shaft. In this case, the oil ring delivers oil to the sleeve bearing directly as it rotates. The rolling-element thrust bearing has a separate oil ring to provide lubrication.

A hydrodynamic radial and hydrodynamic thrust bearing arrangement requires an external lubricating oil system. The additional heat load generated by the oil shear from the hydrodynamic thrust bearing requires a larger oil reservoir and external cooling. This lubrication system uses a separate oil pump to circulate cooled and filtered oil to the pump bearings and back to the reservoir. Most systems have a primary oil pump that is shaft driven from the pump outboard bearing housing with a backup electric motor driven pump on the lubrication system assembly or skid. Often oil for the driver bearings and other drive train components, including gears, is integrated into the same lubrication system.

Cooling A water-cooled oil sump utilizes a plain or finned tube that runs through the oil sump. Cooling water is circulated through the tube, conducting heat from the oil (Figure 6). Some designs use a separate water jacket that bolts to the outside of the oil sump to draw heat from the oil through the bearing housing wall into the water jacket. Tube style heat exchangers are preferable because they are easier to disassemble and clean any mineral deposits, or even replace, than are jackets. Water cooling is not effective nor is it applied on oil mist or grease-lubricated systems.

Air-cooling is becoming more popular as cooling water is in limited supply or of poor quality in many parts of the world. It is also less expensive to operate. In its simplest form, a fan is mounted on the shaft on the outboard side of the bearing housing to direct air over the bearing frame toward the pump, thus creating forced convection of heat from the bearing frame. Sometimes a heat deflector is also utilized which is a finned disc mounted on the shaft between the seal chamber and the bearing housing (Figure 7). This device conducts heat from the shaft before it reaches the bearing housing, absorbs heat radiated

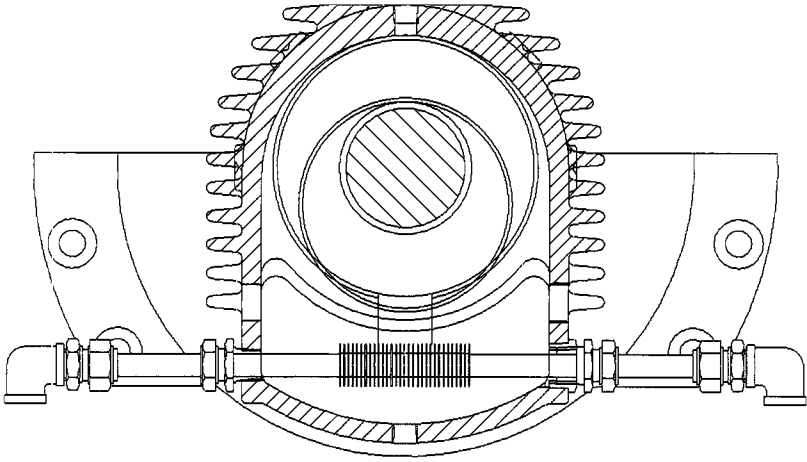


FIGURE 6 Finned cooling tube running through the oil sump

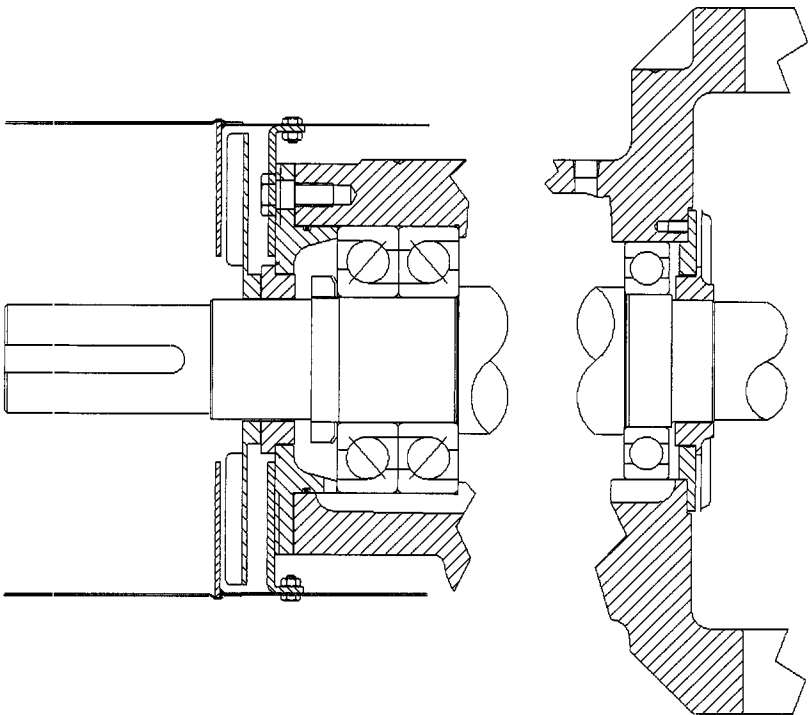


FIGURE 7 Finned disc mounted on the shaft between the seal chamber and bearing housing to deflect and remove heat

from the seal chamber and also creates forced convection in the seal chamber area. Air-cooling can be used on sump oil, oil mist and grease-lubricated bearing systems.

External lubricating oil systems incorporate either an oil-to-water or oil-to-air heat exchanger. The oil-to-water heat exchanger is a tube-type system with the lubricating oil on the tube side and the water on the shell side. Small oil-to-air heat exchangers consist of an electric motor driven fan, forcing convection over a coil of finned tubes containing the circulating oil. Larger systems utilize a large radiator and fan much like you would see on a diesel engine.

Sizing Generally, bearings should be sized for continuous operation under all specified conditions. Many specifications, such as API 610 (ISO 13709), require the thrust bearing be sized to carry loads at both design internal clearances and also at twice design internal clearances. Any thrust loading from gears and any axial force transmitted through flexible couplings also need to be added to thrust from the pump rotor to give a total load requirement for the thrust bearing. Thrust bearings should also provide full-load capabilities if the pump's normal direction of rotation is reversed. Also, if a sleeve-bearing motor (without a thrust bearing) is directly connected to the pump shaft with a coupling, the coupling-transmitted thrust should be assumed to be the maximum motor thrust.

Rolling-element bearing life (basic rating life, L_{10h}) for each bearing or bearing pair) shall be calculated in accordance with ISO 281. Minimum bearing life is specified by various industry standards such as ASME B73, API 610, and ISO 13709. ASME B73 requires 16,000 h and API 610 (ISO 13709) requires 25,000 h with continuous operation at rated conditions and at least 16,000 h at maximum radial and axial loads and rated speed.

It is not uncommon in some municipal specifications to find a requirement of 100,000 h bearing life. This may be suitable and achievable for pumps with low operating speeds, but it could result in severely oversizing bearings for pumps operating at 2 pole motor speeds. Bearing manufacturer catalogues provide maximum operating speeds for rolling-element bearings, but general practice is to select bearings that are below 75% of published values for centrifugal pumps. The higher the operating speed, the more dynamic loads are placed on the bearing-rolling elements, which require near-ideal steady state conditions to operate properly and achieve predicted life. Since pump shafts have variable loads and are subject to deflection and vibration, it is best to select bearings with a speed margin.

Mounting Most rolling-element bearings should be retained on the shaft with a light interference fit and fitted into the housing with a small diametral clearance. Note that some roller bearing designs, such as cylindrical roller types with separable races, require a light interference fit in the housing bore to work properly. Guidelines for the amount of interference fit or clearance recommended for each bearing type and loading are given in ABMA Standard 7. Bearings should be mounted directly on the shaft whenever possible. Bearing carriers that are sometimes used between the bearings and the shaft can be a problem if tolerances and clearances are not carefully controlled and should be avoided. It is preferred that bearings are located axially on the shaft using shoulders, collars, or other positive locating devices such as split rings. Snap rings and spring-type washers are generally not acceptable as they usually fit into grooves in the shaft that contain sharp radii, resulting in severe shaft stress risers. Thrust bearings should always be locked to shafts with a matching bearing nut with a tongue-type lock washer.

Bearings should be heated with an induction-type bearing heater or in a heated oil bath prior to mounting on the shaft. A torch or flame should never be used to heat a bearing. It is difficult to uniformly heat the bearing and to control the metal temperature with a flame, which likely will result in distorting the bearing or changing the metal properties. Heating the bearing to 250°F is sufficient to provide a slip fit of the bearing onto the shaft. If nonmetallic retainers are allowed, extra care must be exercised in the heating process. Thermally insulated gloves must be used when handling hot bearings.

Operation Heat is generated during bearing operation for various reasons. The lubricant is subject to shear forces as it circulates in the bearing. Excess oil in the bearings increases churning and shearing resulting in elevated operating temperatures. Energy from

deflection of the bearing metal occurs on both the bearing races and the rolling element itself, so a highly loaded bearing will run hotter than a lightly loaded bearing. Friction occurs from the rolling element as it revolves in the retainer pocket. Friction occurs from any relative motion or sliding of the rolling element to the bearing race. Bearings whose rolling elements are skidding run very hot. Highly preloaded bearings also tend to run hotter than lightly preloaded or non-preloaded bearings, provided no skidding is occurring.

The operating speed, the type of rolling element, angular operation of a rolling element, the type of bearing retainer, the number of rolling elements, and ambient temperature can also influence operating temperature. Higher speeds result in increased lubricant shear and retainer friction. Roller-type bearings increase lubricant shear, retainer friction, and sliding friction compared to ball-type bearings. Bearings whose rolling elements operate at an angle such as angular contact ball bearings and spherical roller bearings are more prone to sliding and skidding which increases friction. These types of bearings also utilize a heavy-duty retainer that operates in close proximity to the rolling element, resulting in more lubricant shear and friction. Maximum capacity bearings (when allowed) have a higher quantity of rolling elements than standard-capacity bearings, which increases lubricant shear and friction. Bearing operating temperature is approximately proportional to ambient temperature.

As can be surmised from the above discussion, bearing operating temperatures vary widely. Bearing manufacturers generally claim a maximum allowable bearing metal operating temperature of 250°F (121°C), but this is rarely utilized in practice. The practical limit of operation is considered to be 200°F (93°C), though there are many successful applications operating at higher temperatures. 180°F (82°C) is the upper temperature limit of mineral-type oils to achieve a reasonable lubricant life. Mineral oils operating above this temperature oxidize very rapidly, resulting in what has become known in the industry as black oil. Synthetic oils are utilized successfully at higher bearing temperatures. The following operating temperature limits have been established for API Standard 610 (ISO 13709) pumps, based on an ambient temperature of 110°F (43°C):

- For ring-oiled or splash systems, the oil sump temperature should not exceed 180°F (82°C). Bearing outer ring temperatures should not exceed 200°F (93°C).
- For pressurized lubrication systems, the outlet oil temperature should not exceed 160°F (70°C) and bearing metal temperatures should not exceed 200°F (93°C).
- During a shop test with a pressurized system, under the most severe operating conditions specified, the bearing oil temperature rise should not exceed 50°R (28 K).

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SECTION 6.2

OIL FILM JOURNAL BEARINGS

WILBUR SHAPIRO

PRINCIPLES OF OPERATION

A journal bearing is essentially a viscous pump, and it derives load capacity by pumping the lubricant through a small clearance region. In Figure 1, the fluid is dragged along by the rotating journal. To generate pressure, the resistance to pumping must increase in the direction of the flow. In the figure, the journal moves to form a converging tapered clearance in the direction of the rotation or flow.

The eccentricity e is the total displacement of the journal from its concentric position. The attitude angle γ in Figure 1 is the angle between the load direction and the line of centers. Note that, because of the necessity to form a converging wedge, the displacement of the journal is not along a line that is coincident with the load vector. A positive pressure is produced in the converging region of the clearance. Downstream from the minimum film thickness, which occurs along the line of centers, the film becomes divergent. The resistance decreases in the direction of pumping, and either negative pressures occur or the air in the lubricant gasifies or cavitates and a region of atmospheric pressure occurs in the bearing area. This phenomenon is known as *fluid film bearing cavitation*. It should be clearly distinguished from other forms of cavitation that take place in pumps, such as in the impeller, for example. Here the fluid is traveling at a high velocity and the inertia forces on each fluid element dominate. Implosions occur in the impeller and can cause damage.

In a bearing, the viscous forces dominate and each fluid particle moves at a constant velocity in proportion to the net shearing forces on it. Thus, cavitation in a bearing is more of a change of the phase of the lubricant that occurs in a region of lower pressure that permits the release of entrained gases. Generally, bearing cavitation does not cause damage.

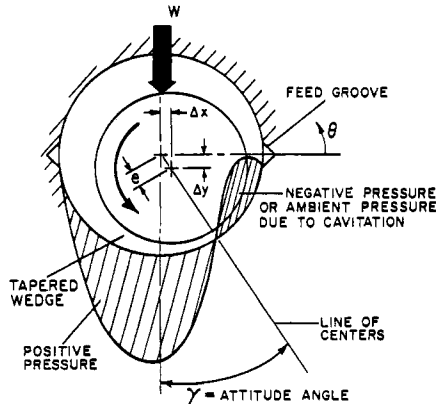


FIGURE 1 Two-groove cylindrical bearing

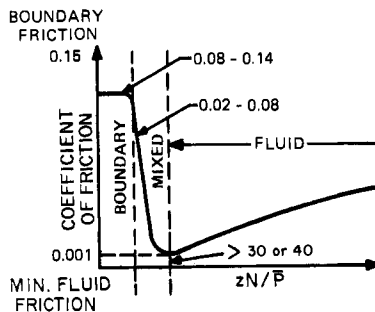
Regimes of Lubrication Whether or not a fluid film can be formed, journal rotation is dependent on several factors including the surface speed, viscosity, and load capacity. A parameter¹ is often used to determine a particular regime of lubrication, $ZN/\bar{P}$, where

Z = viscosity of lubricant, cP (Pa · s)

N = rotating speed, rpm

$\bar{P}$ = average pressure of the bearing, lb/in² (bar)*

A plot of the coefficient of friction versus $ZN/\bar{P}$ generally has the form shown in Figure 2. At low values of $ZN/\bar{P}$, a combination of viscosity, speed, and load places a bearing in a boundary lubricated regime where typical coefficients of friction are 0.08 to 0.14. Boundary lubrication implies intimate contact between the opposed surfaces. As the value of the parameter increases as a result of the increased speed, increased viscosity, or lowered load, there is a dramatic reduction in the coefficient of friction. In this region, there is a mixed film lubrication and the coefficient of friction varies between 0.02 and 0.08. By mixed film lubrication, it is meant that the journal is partly surrounded by a fluid film and is partly

FIGURE 2 Coefficient of friction versus $ZN/\bar{P}$ (Ref. 1)

*1 bar = 10⁵ Pa. For a discussion of bar, see "SI Units: A Commentary" in the front matter

supported by rubbing contact between the opposed members. As $ZN/\bar{P}$ increases further, a situation of full fluid film lubrication prevails.

A general rule of thumb is that $ZN/\bar{P}$ should be 30 (0.44) or greater for a fluid film to be generated. Note that as $ZN/\bar{P}$ continues to increase beyond the full film demarcation, the coefficient of friction rises, but at a relatively low rate and generally remains in regions of low coefficients of friction.

EXAMPLE

$$\begin{aligned} Z &= 30 \text{ cP (0.03 Pa} \cdot \text{s)} \\ N &= 150 \text{ rpm} \\ \bar{P} &= 200 \text{ lb/in}^2 \text{ (13.6 bar)} \\ \frac{ZN}{\bar{P}} &= \frac{30 \times 150}{200} = 22.5 \end{aligned}$$

The bearing is not fluidborne and is operating in the mixed film regime. At what speed will the bearing become hydrodynamic? For hydrodynamic operations, $ZN/\bar{P} = 30$. Therefore,

$$N = \frac{30\bar{P}}{Z} = \frac{30 \times 200}{30} = 200 \text{ rpm}$$

THEORETICAL FOUNDATIONS

The foundation of a fluid film-bearing analysis emanates from the boundary layer theory of fluid mechanics. The governing differential equation was first formulated by Osborne Reynolds in 1886 and is known as Reynolds' equation in his honor. It has been only in the last 30 years or so that general solutions have been obtained, and this has been primarily due to the use of numerical methods applied to the digital computer. References 2 and 3 go into the details of contemporary numerical solutions and are recommended for those interested in the analytical aspects of lubrication.

Principal Assumptions Reynolds' equation can be derived from the Navier-Stokes equation of fluid mechanics, and a number of textbooks are available that comprehensively describe the derivation.⁴ The primary assumptions are as follows:

- Laminar flow conditions prevail, and the fluids obey a Newtonian shear stress distribution where the shear stress is proportional to the velocity gradient.
- Inertial forces, resulting from acceleration of the liquid, are small relative to the viscous shear forces and may be neglected.
- The pressure across the film is constant since the fluid films are so thin.
- The height of the fluid film is small relative to other geometric dimensions, and so the curvature of the fluid film can be ignored.
- The viscosity of the liquid remains constant. In most cases, this is a reasonable assumption since it has been repeatedly demonstrated that, if the average viscosity is used, little error is introduced and the complexity of the analysis is considerably reduced.

Derivation of Reynold's Equation of Lubrication Assume that the rotating journal has a peripheral velocity U . Consider an elemental volume in the clearance space of the bearing and establish equilibrium (see Figure 3). Note that since inertial forces are neglected, the volume is in equilibrium by the pressure and shear forces acting upon it, so there is no acceleration. As shown in Figure 4, p is the pressure and τ is the shear stress acting upon the volume. Summing forces in the x direction

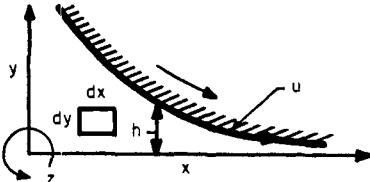


FIGURE 3 Fluid control volume

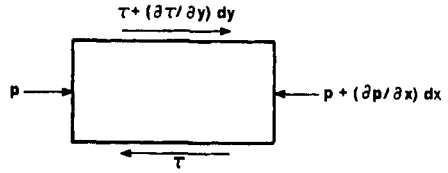


FIGURE 4 Force equilibrium on fluid element

$$p \, dy \, dz - \left(p + \frac{\partial p}{\partial x} dx \right) dy \, dz - \tau \, dx \, dz + \left(\tau + \frac{\partial \tau}{\partial y} dy \right) dx \, dz = 0 \quad (1)$$

$$-\frac{\partial p}{\partial x} dx \, dy \, dz + \frac{\partial \tau}{\partial y} dx \, dy \, dz = 0$$

or

$$\frac{\partial p}{\partial x} = \frac{\partial \tau}{\partial y} \quad (2)$$

For a Newtonian fluid in a laminar flow, the shear stress is directly related to the velocity gradient with the proportionality constant being the absolute viscosity μ (see Figure 5):

$$\tau = \mu \frac{dv}{dy} \quad (3)$$

$$\frac{\partial \tau}{\partial y} = \mu \frac{\partial^2 v}{\partial y^2}$$

where v is the fluid velocity. Substituting Eq. 3 into Eq. 2, we obtain

$$\frac{\partial p}{\partial x} = \mu \frac{\partial^2 v}{\partial y^2} \quad (4)$$

Integrating with respect to y twice produces the following equation:

$$v = \frac{1}{\mu} \frac{\partial p}{\partial x} \frac{y^2}{2} + C_1 y + C_2 \quad (5)$$

where C_1 and C_2 are constants of integration. The boundary conditions are

$$v = 0, y = 0 \text{ and} \quad (6)$$

$$v = u, y = h$$

Substituting the boundary conditions of Eq. 6 into Eq. 5 results in the following expression for v :

$$v = \frac{1}{2\mu} \frac{\partial p}{\partial x} (y^2 - h^2 y) + \frac{uy}{2} \quad (7)$$

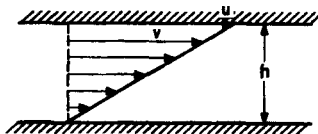


FIGURE 5 Laminar velocity distribution across film

The velocity in the z direction would be similar, except that the surface velocity term would be omitted because no surface velocity exists in the z direction. In addition, the pressure gradient would be with respect to z .

Now let us consider the flow across the film due to this velocity. Note that Eq. 7 is the velocity computed in the x direction, which is in the direction of rotation of the journal:

$$q_x = \int_0^h v_x dy = \int_0^h \left[\frac{1}{2\mu} \frac{\partial p}{\partial x} (y^2 - h^2 y) + \frac{uy}{2} \right] dy \quad (8)$$

After integrating,

$$q_x = -\frac{1}{12\mu} \frac{\partial p}{\partial x} h^3 + \frac{1}{2} uh \quad (9)$$

Note that q_x is the flow per unit width across the film. The flow in the axial direction is

$$q_z = -\frac{1}{12\mu} \frac{\partial p}{\partial z} h^3 \quad (10)$$

Now let us consider a flow balance through an elemental volume across the film (see Figure 6). The net outflow through the volume equals the net reduction in volume per unit time:

$$\left(q_x + \frac{\partial q_x}{\partial x} dx \right) dz - q_x dz + \left(q_z + \frac{\partial q_z}{\partial z} dz \right) dx - q_z dx = -\frac{\partial h}{\partial t} dx dz \quad (11)$$

Thus, this gives us

$$\frac{\partial q_x}{\partial x} + \frac{\partial q_z}{\partial z} = -\frac{\partial h}{\partial t} \quad (12)$$

Substituting Eqs. 9 and 10 into Eq. 12, we obtain

$$-\frac{\partial}{\partial x} \left(\frac{1}{12\mu} h^3 \frac{\partial p}{\partial x} \right) - \frac{\partial}{\partial z} \left(\frac{1}{12\mu} h^3 \frac{\partial p}{\partial z} \right) = -\frac{\partial h}{\partial t} - \frac{u}{2} \frac{\partial h}{\partial x} \quad (13)$$

and the final equation becomes

$$\frac{\partial}{\partial x} \left(\frac{h^3}{\mu} \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial z} \left(\frac{h^3}{\mu} \frac{\partial p}{\partial z} \right) = 6\mu \frac{\partial h}{\partial x} + 12 \frac{\partial h}{\partial t} \quad (14)$$

Equation 14 is the general form of Reynolds' equation used for laminar, two-dimensional lubrication problems.

Reynolds' equation is a flow balance equation. The left-hand side represents pressure-induced flows in the x and z directions through the differential element. The first term on the right-hand side represents the shear flow of the fluid induced by the surface velocity of the journal u . Note that this term contains the derivative of clearance with respect to distance. If this term is zero, then there is zero pressure produced by hydrodynamic action,

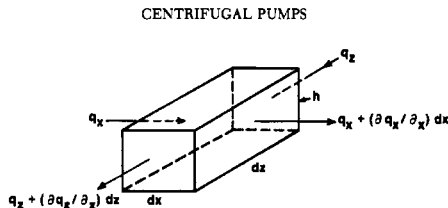


FIGURE 6 Flow balance across control volume

and the term $\partial h/\partial x$ is the mathematical representation of the tapered wedge. The second term on the right-hand side refers to a time rate of change of the film thickness, which can be translated to a normal velocity of the center of the journal. It produces pressure by a fluid velocity normal to the bearing surfaces that attempts to squeeze fluid out of a restricted clearance space. This phenomenon is called the squeeze film effect in bearing terminology. Since it is proportional to the velocity of the center of the journal, it is the phenomenon that produces viscous damping in a bearing.

The solution to Reynolds' equation (refer to Eq. 14) provides the pressure at all points in the bearing. The application of the digital computer has enabled a rapid solution of Reynolds' equation over a grid network representing the bearing area.^{2,3}

Once the pressures have been obtained, a numerical integration is applied to determine the performance parameters (in other words, the load capacity):

$$w = \iint p r d\theta dr \quad (15)$$

The flow across any circumferential line is

$$q_\theta = \oint \left(-\frac{1}{12\mu} \frac{\partial p}{r \partial \theta} h^3 + \frac{1}{2} u h \right) dz \quad (16)$$

The flow across any axial line is

$$q_z = \oint \left(-\frac{1}{12\mu} \frac{\partial p}{\partial z} h^3 \right) r d\theta \quad (17)$$

The viscous frictional moment is obtained by integrating the shear stress over the area and can be shown to be

$$M_f = \iint \left[\frac{1}{r} \frac{\partial p}{\partial \theta} \frac{h}{2} + \frac{\mu r \omega}{h} \right] r^2 d\theta dz \quad (18)$$

where ω = journal surface speed, rad/s.

Typical computer program output includes the following:

- Pressure distribution throughout the grid network
- Load capacity
- Side leakage and carryover flows
- Viscous power losses
- Righting moments due to misalignments
- Attitude angles
- Cross-coupled spring and damping coefficients due to displacements and velocity perturbations of the journal center
- Clearance distribution

Turbulence Equation 14 is for laminar conditions. For very high speed bearings, operations beyond the turbulent regime may occur and Reynolds' equation must be modified. The turbulent theory has been developed, and the literature on this topic can enable performance predictions for turbulent bearings.^{5,6}

The onset of turbulence is determined by examining the bearing's Reynolds number, which is the ratio of inertia to viscous forces and is defined as

$$Re = \frac{\rho u h}{\mu} \quad (19)$$

where Re = Reynolds number

ρ = fluid density, $\text{lb} \cdot \text{s}^2/\text{in}^4$ (kg/m^3)

u = surface velocity, in/s (m/s)

h = local film thickness, in (m)

μ = viscosity, $\text{lb} \cdot \text{s}/\text{in}^2$ ($\text{Pa} \cdot \text{s}$)

A reasonable approximation is to use the concentric clearance c for h . In terms of N rpm and journal diameter D , the Reynolds number is

$$Re = \rho \frac{\pi DN}{60} \frac{c}{\mu} \quad (20)$$

The criterion for turbulence in journal bearings is that $Re \geq 1000$.

EXAMPLE As an example, consider the following:

Journal diameter $D = 5$ in (127 mm)

Bearing length $L = 5$ in (127 mm)

Radial clearance $c = 0.0025$ in (0.064 mm)

Operating speed $N = 5000$ rpm

Lubricant viscosity $\mu = 2 \times 10^{-6}$ $\text{lb} \cdot \text{s}/\text{in}^2$ (14×10^{-3} $\text{Pa} \cdot \text{s}$)

Lubricant density $\rho = 7.95 \times 10^{-5}$ $\text{lb} \cdot \text{s}^2/\text{in}^4$ (8.66×10^{-11} $\text{kg} \cdot \text{s}^2/\text{mm}^4$)

$$\begin{aligned} Re &= \rho \frac{\pi DN}{60} \frac{c}{\mu} \\ &= 7.95 \times 10^{-5} \times \frac{\pi \times 5 \times 5000}{60} \times \frac{0.0025}{2 \times 10^{-6}} = 130 \end{aligned}$$

Thus, the bearing is operating in the laminar regime. To become turbulent (assuming constant viscosity), the operating speed would have to increase to approximately 38,500 rpm.

The example cited is for a relatively high-speed, oil-lubricated bearing for pump applications. In general, most pump bearings operate in the laminar regime. Exceptions might occur when water is used as the lubricant because it is much less viscous than oil.

Evaluation of Frictional Losses It is often desirable to obtain a quick estimate of viscous drag losses that the journal bearings produce. If we consider shear forces again, we return to the laminar flow equation:

$$F = \mu A \frac{u}{h} \quad (21)$$

where F = viscous shear force, lb (N)

μ = viscosity, $\text{lb} \cdot \text{s}/\text{in}^2$ ($\text{Pa} \cdot \text{s}$)

A = surface area, in^2 (m^2)

u = journal velocity, in/s (m/s)

h = film thickness, in (m)

To obtain friction, we multiply both sides of Eq. 21 by the journal radius R . Then the viscous frictional moment is

$$M = \mu AR \frac{u}{h}$$

The frictional horsepower loss is

$$FHP = \frac{NM}{63,000}$$

where N = rotating speed, rpm

M = moment, lb · in ($N \cdot m$)

Also

A = surface area of bearing = πDL , in² (m²)

u = surface speed = $\frac{\pi DN}{60}$ in/s (m/s)

Substituting, we obtain

$$FHP = \frac{\mu L D^3 N^2}{766,000h} \quad (22)$$

In obtaining approximate losses for estimation purposes, the concentric clearance c is substituted for the local film thickness h .

If we consider the previous example where $D = 5$ in (127 mm), $L = 5$ in (127 mm), $c = 0.0025$ in (0.064 mm), $N = 5000$ rpm, and $\mu = 2 \times 10^{-6}$ lb · s/in² (14×10^{-3} Pa · s), the horsepower loss is

$$FHP = \frac{(2 \times 10^{-6})(5)(5)^3(5000)^2}{(766,000)(0.0025)} = 16.32 \text{ hp}$$

Note that for thrust bearings, the frictional horsepower loss is

$$FHP = \frac{\mu N^2}{h} \frac{OD^4 - ID^4}{6.127 \times 10^6} \quad (23)$$

where OD = outside diameter

ID = inside diameter

A general rule of thumb is that the frictional horsepower in a thrust bearing is approximately twice that in a journal bearing.

BEARING TYPES

Cylindrical Bearing The most common type of journal bearing is the plain cylindrical bushing shown schematically in Figure 1. It can be split and have lubricating feed grooves at the parting line. A ramification is to incorporate axial grooves to enable better cooling and to improve whirl stability (described in more detail below in the discussion of cylindrical bearings with axial grooves). The principle advantages of cylindrical bearings are (1) simple construction and (2) a high-load capacity relative to other bearing configurations.

This type of bearing also has several disadvantages:

- **Whirl Instability:** This is prone to subsynchronous whirling at high speeds and also at low loads. Whirling is an orbiting of the journal (shaft) center in the bearing, a motion that is superimposed upon the normal journal rotation. The orbital frequency is approximately half the rotating speed of the shaft. The expression *half-frequency whirl* is commonly used. The reason for the occurrence of this whirl and more details concerning bearing dynamics are presented in the section on bearing dynamics.
- **Viscous Heat Generation:** Because of the generally large and uninterrupted surface area of this bearing, it generates more viscous power loss than some other types.

- **Contamination:** The cylindrical bearing is more susceptible to contamination problems than other types because contaminants that are dragged in at the leading edge of the bearing cannot easily dislodge because of the absence of grooves or other escape paths.

The advantages of simplicity and load capacity make the plain journal a leading candidate for most applications, but performance should be carefully investigated for whirl instability and potential thermal problems. Cylindrical bearings are generally used for medium-speed (500 in/s [200 mm/sec] surface speed) and medium- to heavy-load applications (250 to 400 lb/in² [17 to 28 bar] on a projected area).

Cylindrical Bearing with Axial Grooves A typical configuration of this type of bearing is a plain cylindrical bearing with four equally spaced longitudinal grooves extending most of the way through the bearing. Usually, a slight land area exists at either end of the groove to force the inlet flow to each groove into the bearing clearance region (see Figure 7), rather than out the groove ends. This configuration is a little less simple than the plain cylindrical bearing, and because the grooves consume some land area, this configuration has less load capacity than the plain bushing. Since oil is fed into each of the axial grooves, this bearing requires more inlet flow but also will run cooler than the plain bushing. The grooves act as convenient outlets for any contaminants in the lubricant, and thus the grooved bearing can tolerate more contamination than the plain cylindrical bearing.

In general, this bearing can be considered as an alternate to a plain bearing if the former can correct a whirl or overheating problem.

Elliptical and Lobe Bearings Elliptical and lobe bearings have noncircular geometries. Figure 8 shows two types of three-lobe bearings with the clearance distribution exaggerated so that the lobe geometry is easily discernible. An elliptical bearing is simply a two-lobe bearing with the major axis along the horizontal axis.

The lobe bearing shown in Figure 8a is a symmetric lobe bearing where the minimum concentric clearance occurs in the center of each lobed region. Thus, at the leading edge region, a converging clearance produces positive pressure, but downstream from the minimum film thickness, a divergent film thickness distribution can be found with resulting negative, or cavitation, pressures.

The canted lobe in Figure 8b, on the other hand, generally develop positive pressure throughout the lobe because the bearing is constructed with a completely converging film thickness in each lobed region. This design has excellent whirl resistance (superior to that of the symmetric lobe bearing) and a reasonably good load capability. A 2:1 ratio between leading and trailing edge concentric clearance is generally a reasonable compromise with respect to performance.

Elliptical and lobe bearings are often used because they provide better resistance to whirls than cylindrical configurations. They do so because they have multiple load-producing pads that assist in preventing large-attitude angles and cross-coupling (see the section on bearing dynamics). Elliptical and lobe bearings are generally used for high-speed, low-load applications where whirls might be a problem.

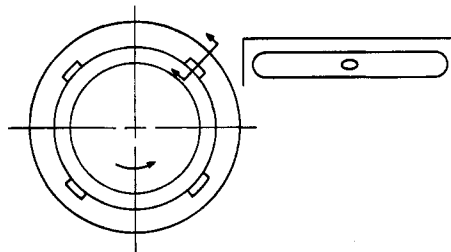


FIGURE 7 Cylindrical bearing with axial grooving

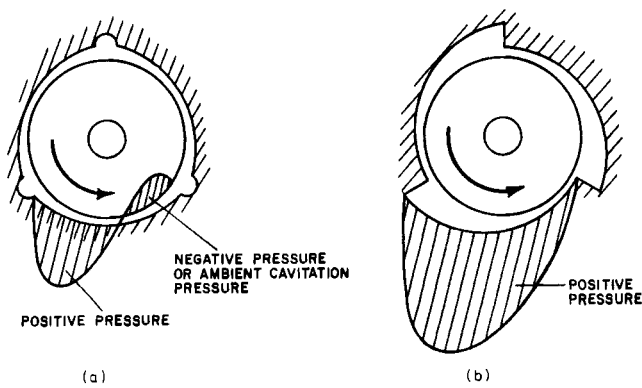


FIGURE 8 (a) symmetric lobe bearing and (b) canted lobe bearing

Elliptical, or two-lobe, bearings generally have poor horizontal stiffness because of the large clearances along the major diameter of the ellipse. The split elliptical configuration, however, is easier to manufacture than the other types because it is two cylindrical bearing halves with material removed along the parting line. Lobe bearings are usually clearance- and tolerance-sensitive. The other types of lobe bearings are complicated to manufacture.

Tilting-Pad Bearings Tilting-pad bearings are used extensively, especially in high-speed applications, because of their whirl-free characteristics. They are the most whirl-free of all bearing configurations.

An important geometric variable for tilting-pad bearings is the preload ratio, defined as shown in Figure 9.

The preload ratio equals

$$\text{Preload ratio} = PR = \frac{c - c'}{c} = 1 - \frac{c'}{c} \tag{24}$$

where c = machined clearance
 c' = concentric pivot film thickness

The variable c' is an installed clearance and is dependent upon the radial position of the pivot. Figure 10 displays two pads. Pad 1 has been installed such that the preload ratio

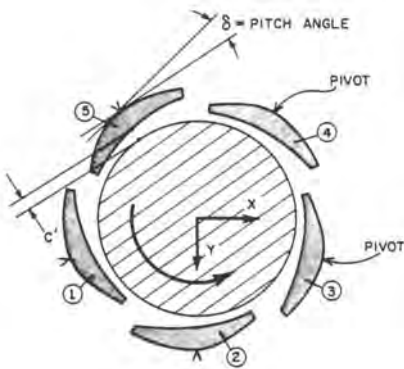


FIGURE 9 Five-pad tilting pad bearing

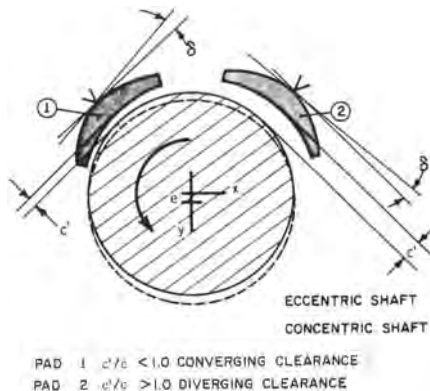


FIGURE 10 Tilting-pad bearing preload

is less than one. For pad 2, the preload ratio is one. The solid line represents the position of the journal in the concentric position. The dashed portion of the journal represents its position when a load is applied to the bottom pads (not shown). Pad 1 is operating with a good converging wedge, even though the journal is moving away from it. Pad 2, on the other hand, is operating with a completely diverging film, which means that it is totally unloaded. Thus, bearings with installed pad preload ratios of one or greater will operate with unloaded pads, which reduces overall stiffness of the bearing and results in a deterioration of stability because the unloaded pads do not aid in resisting cross-coupling influences. In the unloaded position, they are also subject to flutter instability and to a phenomenon known as *leading edge lockup*, where the leading edge is forced against the shaft and is maintained in that position by the frictional interaction of the shaft and the pad. This is especially prevalent in bearings that operate with low-viscosity lubricants, such as gas or water bearings. Thus, it is important to design bearings with preload, although for manufacturing reasons it is common practice to produce bearings without preload.

Tilting-pad bearings have some other characteristics that are both positive and negative:

- They are not as clearance-sensitive as most other bearings.
- Because the pads can move, they can operate safely at a lower minimum film thickness than other bearings.
- They do not provide as much squeeze film damping as rigid configurations.
- Generally, they are more expensive than other bearings.
- For high-speed applications, their pivot contacts can be subjected to fretting corrosion.

Hybrid Bearings A hybrid bearing, schematically shown in Figure 11, derives a load capacity from two sources: (1) the normal hydrodynamic pressure generation and (2) an external high-pressure supply that introduces oil into recesses machined into the bearing surface via restrictors (orifices or capillaries upstream of the recesses). External pressure significantly enhances load capacity. Also, these bearings have excellent low- or zero-speed load capabilities. They are sometimes used as startup devices to lift off the rotor. When self-sustaining hydrodynamic speeds are attained, the external pressure is shut off. The characteristics of externally pressurized, or hybrid, bearings include the following:

- High load and stiffness capabilities
- An external flow that assists in cooling

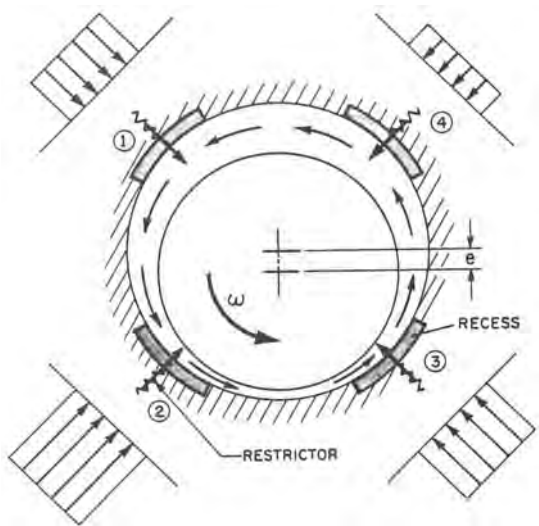


FIGURE 11 Cross-coupling influences in hybrid bearings

- Their clearances and tolerances are generally more liberal than in hydrodynamic bearings
- They require external fluid-supply systems
- They are applied when there is not a sufficient generating speed or when a high-load capacity and stiffness are required
- They are sometimes applied to prevent a whirl, but rotational speeds can unbalance recess pressures, introduce cross coupling, and promote a whirl

STEADY STATE PERFORMANCE

Computer-generated performances have been obtained for most of the bearing types previously discussed. Information has been plotted in a nondimensional format so that no restrictions exist on operating conditions, lubricant properties, and so on. Use of the charts will be subsequently demonstrated by numerical example.

Viscosity One of the key parameters in determining the performance of a bearing is the lubricant viscosity. Viscosity characteristics of commonly used Society of Automotive Engineers (SAE) grades of oil are shown in Figure 12. The units of viscosity are *microreyns*, where the reyn has the units of $\text{lb} \cdot \text{s}/\text{in}^2$ and comes from the ratio of shear stress to the velocity gradient across the film, as indicated by Eq. 21. Other units of viscosity are *centipoises*, *Saybolt seconds universal* (SSU), and *centistokes*. The conversion factors are as follows:

$$\mu(\text{reyns}) = Z \times 1.45 \times 10^{-7} \tag{25}$$

$$\nu(\text{centistokes}) = 0.22 (\text{SSU}) - \frac{180}{\text{SSU}} \tag{26}$$

$$Z(\text{centipoises}) = \nu(\text{centistokes}) \times \text{SG (sp. gr.)} \tag{27}$$

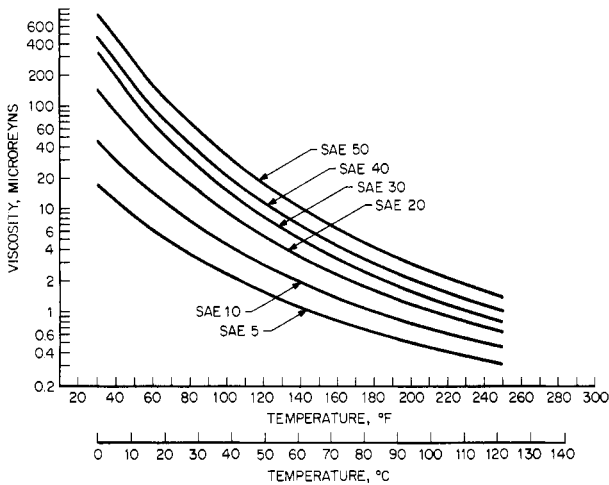


FIGURE 12 Viscosity characteristics for SAE oil grades

Performance Curves Performance plots have been generated for the following types of bearings:

- *Two-groove cylindrical bearings*
- *Symmetric three-lobe bearings:* Each lobe is offset such that in the concentric position the minimum film thickness in the center of each lobed region is half the machined clearance c (see definition following). The pads are each 110° in the angular extent.
- *Canted three-lobe bearing:* The lobing is canted such that in the concentric position the leading edge clearance was twice the trailing edge and the trailing edge film thickness (minimum) in the concentric position was $0.5c$ where c equals the machined clearance. The pads are each 110° in the angular extent.
- *Tilting-pad bearing:* The tilting-pad bearing that information is obtained for is a five-pad bearing with a 60° pad and a preload ratio of 30%.

Two length/diameter ratios are examined for each type of bearing: $L/D = 0.5$ and 1.0 . The definition of the nondimensional parameters is as follows:

$$W = \text{nondimensional load parameter} = wc^2/6\mu\omega RL^3 \quad (28)$$

$$P = \text{nondimensional viscous power loss parameter} = 1100cp/\mu(\omega RL)^2 \quad (29)$$

$$Q = \text{nondimensional flow parameter} = 2q/0.26\omega RLc \quad (30)$$

$$HM = \text{nondimensional minimum film thickness} = h_M/c \quad (31)$$

where w = bearing load capacity, lb (N) and

c = reference clearance (machined clearance = radius of bearing – radius of shaft), in (mm)

μ = absolute viscosity, reyns (lb · s/in²) (cP)

ω = shaft or journal rotational speed rad/s

R = shaft radius, in (mm)

L = bearing length, in (mm)

p = viscous power loss, hp (kW)

q = flow, gpm (m³/h)

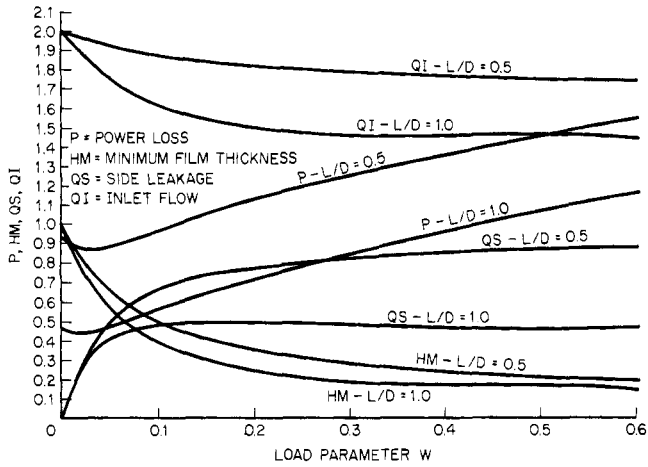


FIGURE 13 Performance characteristics for two-groove cylindrical bearings

- q_i = inlet flow to the leading edge of bearing (for multipad bearings, equals the sum of inlet flow to each pad), gpm (m^3/h)
- q_s = side leakage flow or flow out of the bearing ends (for multipad bearings, equals the sum of side leakage flow of each pad), gpm (m^3/h)
- h_M = minimum film thickness in bearing, in (mm)

Performance curves are shown in Figures 13 through 17.

At times, the nondimensional data can be confusing and lead to erroneous judgments. For example, the nondimensional power loss P is greater for an L/D equal to 0.5 than for an L/D equal to 1.0. However, when the dimensional value of the power loss is being computed, the nondimensional value is multiplied by L^2 . Therefore, the power loss for the L/D

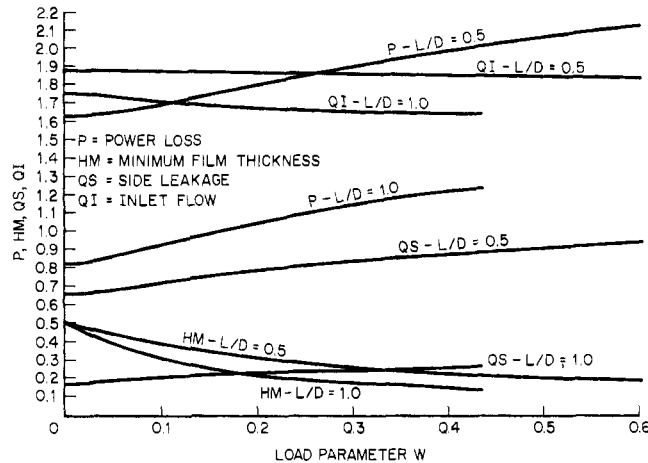


FIGURE 14 Performance characteristics for three-lobe bearings

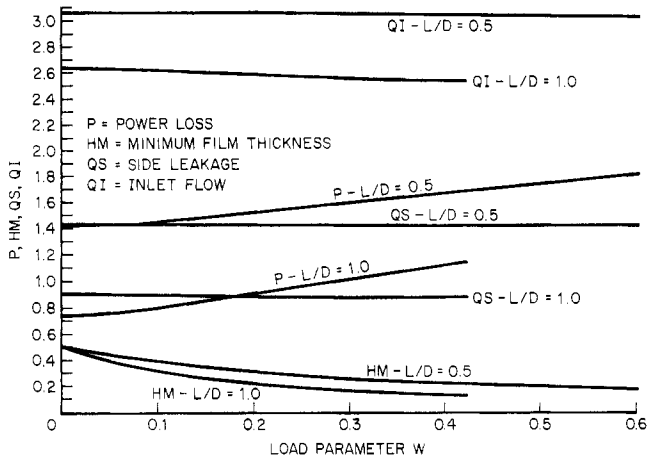


FIGURE 15 Performance characteristics for canted three-lobe bearings

equal to 1.0 will be, as expected, greater than for the L/D equal to 0.5. If the reader uses the data as presented, the dimensional information will prove consistent.

To make comparisons among the bearings, using the nondimensional data is not strictly proper because there may be slight inconsistencies in preloads, the bearings will not be operating at the same average viscosity, and so on. Subsequently, dimensional data derived from the performance curves will be compared, but comparisons of the nondimensional information will provide an indication of performance parameters among the bearings. Comparisons have been made at equal values of the load parameter W and the results are shown in Table 1.

Comparisons of the different bearing types should be made only at the same L/D ratio because of the anomalies (discussed above) of nondimensional parameters that occur at

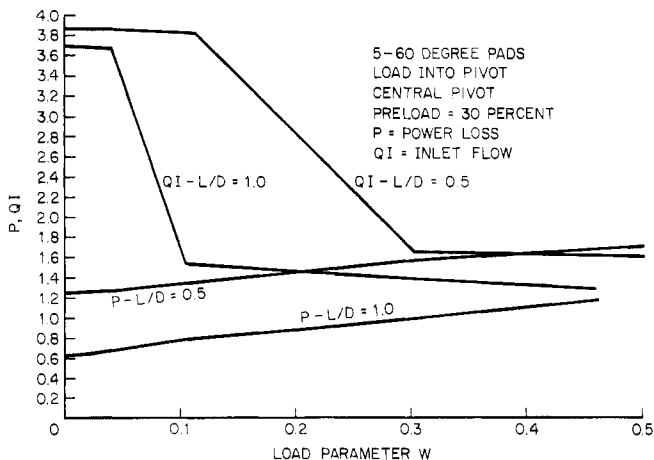


FIGURE 16 Performance characteristics for tilting-pad bearings

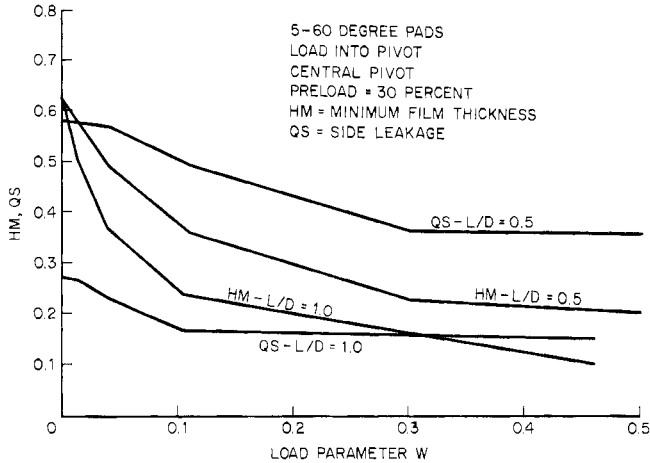


FIGURE 17 Performance characteristics for tilting-pad hearings

different L/D ratios. If we assume that all the reference variables that go into the nondimensional parameters are identical, we can establish the following conclusions:

- The two-groove cylindrical hearing has the highest film thickness and thus the highest load capacity.
- The symmetric three-lobe bearing has the highest power loss.
- The canted three-lobe bearing has the greatest flow requirements.

Note that these comparisons were made on the basis of steady-state performances only. The major reason for applying lobe and tilting-pad bearings is to avoid dynamic instabilities.

Heat Balance Performance is based upon the assumption of a uniform viscosity in the fluid film. Since the viscosity is a strong function of temperature and since the temperature rise of the lubricant due to viscous heat generation is not known a priori, an iterative procedure is required to determine the average viscosity in the film. To determine an average viscosity, there must be a simplified heat balance in the film. The assumption is made that all the viscous heat generated in the film is absorbed by the lubricant as it flows through the film and produces a temperature rise.

TABLE 1 Comparative Results of Bearing Types at $W = 0.2$

Bearing type	$L/D = 0.5$				$L/D = 1$			
	P	HM	QS	QI	P	HM	QS	QI
Two-groove cylindrical	1.13	0.36	0.78	1.81	0.72	0.25	0.50	1.50
Three-lobe	1.80	0.32	0.79	1.85	1.05	0.22	0.22	1.65
Canted three-lobe	1.52	0.31	1.42	3.30	0.91	0.22	0.90	2.58
Tilting-pad	1.50	0.30	0.44	2.82	0.90	0.20	0.16	1.50

QI = nondimensional inlet flow = $2q_i/0.26 wRLc$

QS = nondimensional side leakage flow = $2q_s/0.26 wRLc$

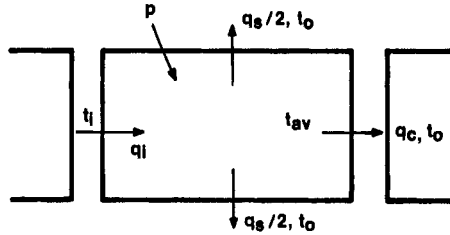


FIGURE 18 Heat balance in a fluid film

Figure 18 shows a developed view of the bearing surface and the parameters involved with conducting the heat balance. The lubricant enters a pad or bearing at the leading edge with an inlet flow q_i and an inlet temperature t_i . As the lubricant enters the bearing surface and flows through it, the lubricant is exposed to a viscous shear, which adds heat to the fluid. In Figure 18, the viscous shear is indicated as a power input p . Some of the fluid flows out of the sides of the bearing, represented by $q_s/2$, and some of the fluid, q_c , is carried either over to the grooving of the next pad or back to the inlet groove for a single pad bearing.

Although the fluid temperature, and thus the viscosity, changes along the length of the pad, it is assumed that the temperature in the film increases to some value t_o for both the side leakage and carry-over fluids. Then the heat balance is conducted as follows:

Heat added to a fluid by a viscous shear equals heat absorbed by the fluid.

$$\begin{aligned} \text{Heat added to fluid by viscous shear} &= \text{heat absorbed by fluid} \\ pJ &= q_s \rho C_p (t_o - t_i) = q_c \rho C_p (t_o - t_i) = (q_s + q_c) \rho C_p (t_o - t_i) \end{aligned} \quad (32)$$

Since by continuity of flow,

$$q_i = q_s + q_c \quad (33)$$

$$pJ = q_i \rho C_p \Delta t \quad (34)$$

$$\Delta t = \frac{pJ}{q_i \rho C_p} \quad (35)$$

where p = viscous heat generation, hp

J = heat equivalent constant = 0.7069 Btu/hp · s (J/kg · s)

ρ = specific weight of flow, lb/in³ (kg/m³)

C_p = specific heat of fluid, Btu/lb · °F (J/kg · °C)

q_i = bearing inlet flow, in³/s (mm³/s)

Δt = temperature rise, °F (°C)

If q_i is in gallons per minute [(1 in³/s = 0.26 gpm) = 16.39 cm³/s],

$$\Delta t = \frac{0.7069(0.26)p}{q_i \rho C_p} = \frac{0.1838p}{q_i \rho C_p} \quad (36)$$

Before proceeding to a sample problem, a word about the flows q_i and q_s , and the general heat balance philosophy. The flow required by the bearing to prevent starvation is q_i . The minimum make-up flow, or the flow that is lost from the ends of the bearing, is the side leakage q_s . Theoretically then, the only flow that need be supplied to the bearing is q_s . However, if this were true, then the heat balance formulation would require another balance between the carry-over flow q_c at some temperature t_o and the make-up flow q_s coming into the pad at temperature t_i . It would be found that, if we supplied only q_s to the bearing, excessive temperatures would result. In most instances, the amount of flow supplied to a bearing exceeds both the inlet flow and the side leakage flow by a significant

margin. Designers often determine the flow to be supplied to a bearing system by a bulk temperature rise of the total flow entering the inlet pipe and exiting the exhaust pipe. Thus, the entire bearing is treated as a black box, and the total flow q_T to the bearing system for a given bulk temperature rise Δt_t is

$$q_T = \frac{pJ}{\rho C_p \Delta t_t} \quad (37)$$

Normally, Δt_t is selected between 20 and 40°F. Since q_T will exceed q_i , not all the flow will enter the film. Some will surround the bearing or flow through the feed groove and act as a cooling medium for heat transfer through the bearing walls or for cooling the fluid that does enter the bearing surface q_i .

EXAMPLE This example demonstrates the use of the design curves and how to perform a simplified heat balance in a bearing analysis.
In this example, we know these values:

Bearing type: two-groove cylindrical

Bearing length $L = 3$ in (76 mm)

Bearing diameter $D = 6$ in (152 mm)

$L/D = 0.5$

$N = 1800$ rpm; $\omega = 1800 \times \pi/30 = 188.5$ rad/s

Bearing load $\omega = 6000$ lb (2721 kg)

Inlet temperature $t_i = 120^\circ\text{F}$ (49°C)

Lubricant = SAE 20

Bearing radial clearance $c = 0.003$ in (0.076 mm)

NOTE: A general rule of thumb is that $c/R = 0.001$.

Assume an average lubricant temperature of $130^\circ\text{F} - \mu$ (SAE 20 at 130°F) = 4.5×10^{-6} reyn ($\text{lb} \cdot \text{s}/\text{in}^2$) (31×10^{-3} Pa $\cdot$ s)

$$\rho = 0.0307 \text{ lb}/\text{in}^3 (849.7 \text{ kg}/\text{m}^3) \text{ (average value for most oils)}$$

$$C_p = 0.5 \text{ Btu}/\text{lb} \cdot ^\circ\text{F} (2093 \text{ J}/\text{kg} \cdot ^\circ\text{C}) \text{ (average value for most oils)}$$

Compute the following:

$$W = \frac{wc^2}{6\mu\omega RL^3} = \frac{(6000)(0.003^2)}{(6)(4.5 \times 10^{-6})(188.5)(3)(3^3)} = 0.1309$$

From Figure 13 and Eq. 29,

$$P = 1.0 = \frac{1100cp}{\mu(\omega RL)^2}$$

$$P = \frac{(1)(4.5 \times 10^{-6})(188.5 \times 3 \times 3)^2}{(1100)(0.003)} = 3.92 \text{ hp (2.95 kW)}$$

From Figure 13 and Eq. 30,

$$Q_I = 1.85 = \frac{2q_i}{0.26\omega RLC}$$

$$q_i = \frac{(1.85)(0.26)(188.5)(3)(3)(0.003)}{2} = 1.224 \text{ gpm (4.63 l/min)}$$

From Eq. 36,

$$\Delta t = \frac{0.1838p}{q_i \rho C_p} = \frac{(0.1838)(3.92)}{(1.224)(0.0307)(0.5)} = 38.35^\circ\text{F} (21.3^\circ\text{C})$$

$$t_{av} = \frac{t_i + (t_i + \Delta t)}{2} = \frac{2t_i + \Delta t}{2} = \frac{(2)(120) + 38.85}{2} = 139.2^\circ\text{F} (59.6^\circ\text{C})$$

The assumed t_{av} of 130°F was apparently not high enough and therefore we must repeat the calculations with a higher value of assumed t_{av} . Assume $t_{av} = 138^\circ\text{F}$, $\mu = 3.8 \times 10^{-6} \text{ lb} \cdot \text{s/in}^2 (26.2 \times 10^{-3} \text{ Pa} \cdot \text{s})$ and repeat the calculations:

$$W = \frac{5.8945 \times 10^{-7}}{\mu} = 0.1552$$

$$P = 1.08 = (1.1466 \times 10^{-6}) \left(\frac{P}{\mu} \right)$$

$$p = \frac{(1.08)(3.8 \times 10^{-6})}{1.1466 \times 10^{-6}} = 3.579 \text{ hp} (2.67 \text{ kW})$$

$$QI = 1.83 = \frac{q_i}{0.6616}$$

$$q_i = 1.211 \text{ gpm} (4.58 \text{ l/min})$$

$$\Delta t = 11.974 \frac{p}{q_i} = \frac{(11.974)(3.579)}{1.211} = 35.38^\circ\text{F} (19.65^\circ\text{C})$$

$$t_{av} = \frac{(2)(120) + 35.38}{2} = 137.7^\circ\text{F} (58.7^\circ\text{C})$$

For all practical purposes, these equal the assumed value of $t_{av} = 138^\circ\text{F} (58.9^\circ\text{C})$.

In conducting the iterative procedure for determining the average fluid temperature, it is sometimes helpful to plot points on a viscosity chart. Referring to Figure 19, suppose

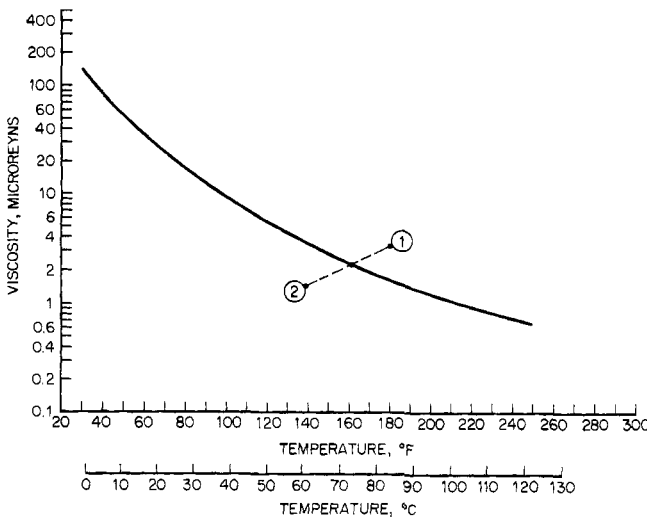


FIGURE 19 Graphical determination of average viscosity for SAE 20 oil

point 1 is the resultant average viscosity and temperature of the initial guess and point 2 represents the result of the second guess. Then, by drawing a straight line between points 1 and 2 and establishing where it intersects the lubricant viscosity curve, we can determine the convergence to the proper result of average viscosity and temperature.

Now that we have convergence, we can determine the remaining variables, which are the minimum film thickness H_M and side leakage Q_s from Figure 13:

$$\begin{aligned} HM &= 0.41 = \frac{h_M}{c} \\ h_M &= (0.41)(0.003) = 0.00123 \text{ in (0.031 mm)} \\ QS &= 0.74 = \frac{2q_s}{0.26\omega RLC} = \frac{q_s}{0.6616} \\ q_s &= (0.6616)(0.74) = 0.490 \text{ gpm (1.85 l/min)} \end{aligned}$$

A summary of the total bearing performance is as follows:

- Load $\omega = 6000 \text{ lb (2721 kg)}$
- Minimum film thickness $h_M = 0.00123 \text{ in (0.031 mm)}$
- Viscous power loss $p = 3.579 \text{ (2.669 kW)}$
- Inlet flow $q_i = 1.211 \text{ gpm (4.58 l/min)}$
- Side leakage $q_s = 0.490 \text{ gpm (1.85 l/min)}$
- Fluid temperature rise $\Delta t = 35.38^\circ\text{F (19.65}^\circ\text{C)}$

We can repeat the same calculation for the other types of bearings, using Figures 14 through 17. The results are shown in Table 2.

The two-groove cylindrical bearing operates with the highest film thickness. The symmetric three-lobe has the lowest film thickness and highest temperature rise. It appears that, on the basis of steady-state performance, manufacture, and cost, the two-groove cylindrical bearing is the best choice.

BEARING DYNAMICS

Fluid film bearings can significantly influence the dynamics of rotating shafts. They are a primary source of damping and thus can inhibit vibrations. Alternatively, they provide a

TABLE 2 Comparative Bearing Performance

Bearing type	$\Delta t, ^\circ\text{F}$ ($^\circ\text{C}$)	$t_{\text{avg}}, ^\circ\text{F}$ ($^\circ\text{C}$)	$p, \text{ hp}$	$q_i, \text{ gpm}$ (cm^3/s)	$q_s, \text{ gpm}$ (cm^3/s)	$h_M, \text{ mils}$ (mm)
Two-groove cylindrical	35 (19.44)	138 (58.9)	3.58	1.21 (76.7)	0.490 (31.05)	1.23 (0.0312)
Symmetric three-lobe	49 (27.22)	144 (62.2)	4.97	1.22 (77.3)	0.516 (32.7)	0.96 (0.0244)
Canted three-lobe	31 (17.22)	135 (57.2)	5.20	2.01 (127.4)	0.940 (59.6)	1.05 (0.0267)
Tilting-pad	26 (14.44)	134 (56.7)	4.88	2.26 (143.2)	0.311 (19.71)	1.02 (0.0259)

The following values were used: $\omega = 6000 \text{ lb (2721 kg)}$, SAE 20 oil, $L = 3 \text{ in (76 mm)}$, $D = 3 \text{ in (76 mm)}$, $N = 1800 \text{ rpm}$, $t_i = 120^\circ\text{F (48.9}^\circ\text{C)}$, $c = 0.008 \text{ in (0.076 mm)}$.

mechanism for self-excited rotor whirl. Whirl is manifest as an orbiting of the journal at a subsynchronous frequency, usually close to one-half the rotating speed. Whirl is usually destructive and must be avoided.

It is the purpose of this section to provide some insight into bearing dynamics, present some background on analytical methods and representations, and discuss some particular bearings and factors that can influence dynamic characteristics. Dynamic performance data and sample problems are presented for several bearing types.

The Concept of Cross Coupling As mentioned in the opening paragraphs of this section, a journal bearing derives load capacity from viscous pumping of the lubricant through a small clearance region. To generate pressure, the resistance to pumping must increase in the direction of the fluid flow. This is accomplished by a movement of the journal such that the clearance distribution takes on the form of a tapered wedge in the direction of rotation, as shown in Figure 1.

The attitude angle γ in Figure 1 is the angle between the load direction and the line of centers. Thus, the displacement of the journal is not along a line that is coincident with the load vector, and a load in one direction causes not only displacements in that direction, but orthogonal displacements as well.

Similarly, a displacement of the journal in the bearing will cause a load opposing the displacement and a load orthogonal to it. Thus, strong cross-coupling influences are introduced by the mechanism by which a bearing operates. The concept of cross-coupling is significant in dynamic characteristics.

It is the cross-coupling characteristics of a journal bearing that can promote self-excited instabilities in the form of bearing whirl. Motion in one direction produces orthogonal forces that in turn cause orthogonal motion. The process continues, and an orbital motion of the journal results. This orbital motion is generally in the same direction as shaft rotation and subsynchronous in frequency. Half-frequency whirl is a self-excited phenomenon and does not require external forces to promote it.

Cross-Coupled Spring and Damping Coefficients For dynamic considerations, a convenient representation of bearing characteristics is a cross-coupled spring and damping coefficients. These are obtained as follows (refer to Figure 1):

1. The equilibrium position to support the given load is established by computer solution of Reynolds' equation.
2. A small displacement to the journal is applied in the y direction. A new solution of Reynolds' equation is obtained, and the resulting forces in the x and y directions are produced. The spring coefficients are as follows:

$$K_{xy} = \frac{\Delta F_x}{\Delta y} \quad (38)$$

$$K_{yy} = \frac{\Delta F_y}{\Delta y} \quad (39)$$

where K_{xy} = the stiffness in the x direction due to y displacement
 ΔF_x = the difference in x forces between displaced and equilibrium positions
 Δy = the displacement from the equilibrium position in the y direction
 K_{yy} = the stiffness in the y direction due to y displacement
 ΔF_y = the difference in y forces between displaced and equilibrium positions

3. The journal is returned to its equilibrium position and an x displacement is applied. Similar reasoning produces K_{xx} and K_{yx} .

The cross-coupled damping coefficients are produced in a similar manner, except, instead of displacements in the x and y direction, velocities in these directions are consecutively applied with the journal in the equilibrium position. The mechanism for increasing the load capacity is squeeze film in which the last term on the right-hand side of

Equation 14 is actuated. Thus, for most fixed bearing configurations, eight coefficients exist: four spring and four damping. The total force on the journal is

$$F_i = K_{ij}x_j + D_{ij}\dot{x}_j \tag{40}$$

where F_i = force in the i th direction.

Repeated subscripts imply the following summation:

$$K_{ij}x_j = K_{ix}x + K_{iy}y + \dots$$

It should be realized that the cross-coupled spring and damping coefficients represent a linearization of bearing characteristics. When they are used, the equilibrium position should be accurately determined, as the coefficients are valid for only a small displacement region encompassing the equilibrium position of the journal. This is true because the spring and damping coefficients remain constant for only a small region of the equilibrium position.

Consider the two-groove cylindrical bearing shown in Figure 1, with the geometric and operating conditions indicated in Table 3. The computer solution (also the performance curves in Figure 13) produces the following results:

Bearing load $w = 20,780$ lb (9,424 kg)

Power loss = 15.51 hp (11.56 kW)

Minimum film thickness $h_M = 0.00125$ in (0.032 mm)

Side leakage $q_s = 0.941$ gpm (3.56 l/mm)

The spring and damping coefficients are

Spring coefficients, lb/in (kg/mm):

$$\begin{bmatrix} K_{xx} & K_{xy} \\ K_{yx} & K_{yy} \end{bmatrix} = \begin{bmatrix} -12.14 \times 10^6 & -4.64 \times 10^6 \\ 28.3 \times 10^6 & -20.41 \times 10^6 \end{bmatrix}$$

Damping coefficients, lb · s/in kg · s/in:

$$\begin{bmatrix} D_{xx} & D_{xy} \\ D_{yx} & D_{yy} \end{bmatrix} = \begin{bmatrix} -2.85 \times 10^4 & 2.66 \times 10^4 \\ -2.69 \times 10^4 & -1.11 \times 10^5 \end{bmatrix}$$

The negative signs imply a positive stiffness because the restoring load is opposite the applied load. Note that for this bearing configuration there is very strong cross-coupling, evidenced by the magnitude of the off-diagonal terms.

Critical Mass The cross-coupled spring and damping coefficients provide a convenient way of representing a bearing in a stability analysis. They reduce the fluid film bearing to a spring-mass system (see Figure 20), and consequently stability and dynamics problems are simplified considerably.

Consider a journal of mass M operating in a bearing. The journal can be considered to have two degrees of freedom, x and y . The governing equations are

TABLE 3 Two-Groove Cylindrical Bearing Geometry and Operating Conditions

Journal diameter D	=	5 in (127 mm)
Bearing length L	=	5 in (127 mm)
Active pad angle θ_p ,	=	160° (10° grooves on either side)
Radial clearance c	=	0.0025 in (0.064 mm)
Operating speed N	=	5000 rpm
Lubricant viscosity μ	=	2×10^{-6} lb · s/in ² (13.79×10^{-3} Pa · s)
Eccentricity ratio ε	=	0.5
Load direction is vertical downward		

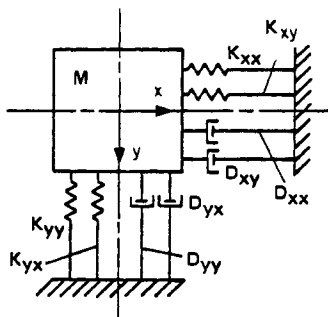


FIGURE 20 Point mass representation of a bearing supported on cross-coupled springs and dampers

$$M\ddot{x} + D_{xx}\dot{x} + D_{xy}\dot{y} + K_{xx}x + K_{xy}y = 0 \quad (41)$$

$$M\ddot{y} + D_{yy}\dot{y} + D_{yx}\dot{x} + K_{yy}y + K_{yx}x = 0 \quad (42)$$

Assume a sinusoidal response to the form

$$x = x_0 e^{\beta t} \quad (43)$$

$$y = y_0 e^{\beta t} \quad (44)$$

where β is a complex variable:

$$\beta = \alpha + i\omega \quad (45)$$

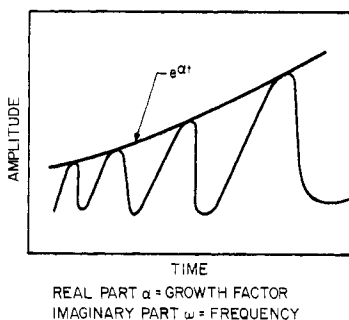
By the Euler expansion of $e^{\beta t}$, another way to write Eqs. 43 and 44 is

$$x = x_0 e^{\alpha t} (\cos \omega t + i \sin \omega t) \quad (46)$$

$$y = y_0 e^{\alpha t} (\cos \omega t + i \sin \omega t) \quad (47)$$

An interpretation of α and ω is shown in Figure 21. The real part of $\beta = \alpha$ is called the *growth or attenuation factor*. The imaginary part is the frequency of vibration. A positive real part means that the response to a disturbance grows in time. The growth factor is similar to the logarithmic mean decrement, which is common in vibration theory:

$$\alpha \tau = \ln \frac{x_{n+1}}{x_n} \quad (48)$$



Interpretation of $\chi = \chi_0 e^{\beta t}$

where

$$\beta = \alpha + i\omega$$

$$\chi = \chi_0 e^{\alpha t} e^{i\omega t} = \chi_0 e^{\alpha t} (\cos \omega t + i \sin \omega t)$$

FIGURE 21 Interpretation of the growth factor α and orbital frequency ω

where τ = period of vibration
 x_{n+1} = amplitude at time $n + 1$
 x_n = amplitude at time n

Thus, the growth factor α is a measure of the growth or decay of the journal to a small disturbance. A positive growth factor implies an instability.

The solutions to Eqs. 41 and 42 are obtained by substituting Eqs. 43 and 44, which produces the following:

$$\begin{bmatrix} (M\beta^2 + D_{xx}\beta + K_{xx}) & (D_{xy}\beta + K_{xy}) \\ (\beta D_{yx} + K_{yx}) & (M\beta^2 + \beta D_{yy} + K_{yy}) \end{bmatrix} \begin{Bmatrix} x_0 \\ y_0 \end{Bmatrix} = \{0\}$$

To obtain a solution, the determinant of the coefficient matrix must vanish. Expansion produces a polynomial in β that can be solved for the roots of β , which in turn provide the growth factors and frequencies of vibration.

It is possible to obtain a closed-form solution of Eqs. 41 and 42 for the critical mass and resulting orbital frequency. The critical mass M is defined as that mass above which an instability will occur. At the threshold of instability, the real part of $\beta = \alpha$ goes to zero and $\beta = i\omega$.

Substituting into Eq. 49, expanding the determinant and separating real and imaginary components produces the following equations:

$$M^2\omega^4 - \underbrace{M(K_{yy} + K_{xx})\omega^2}_A + \underbrace{(D_{yx}D_{xy} - D_{xx}D_{yy})\omega^2}_B + \underbrace{K_{xx}K_{yy} - K_{xy}K_{yx}}_C = 0 \quad (50)$$

$$\underbrace{M(D_{yy} + D_{xx})\omega^2}_D + \underbrace{(D_{xy}K_{yx} + D_{yx}K_{xy} - D_{xx}K_{yy} - D_{yy}K_{xx})}_E = 0 \quad (51)$$

The two equations can be solved for M and ω :

$$M = \frac{BED}{E^2 - AED + CD^2} \quad (52)$$

$$\omega = \sqrt{\frac{-(AED + E^2 + CD^2)}{BD^2}} \quad (53)$$

Thus, if the cross-coupled coefficients are known, it is possible to determine the critical mass and the orbital frequency. If the mass acting on the bearing exceeds or equals the critical value, then an instability will occur.

Dynamic Stability of Various Bearing Types Several parameters can be used to establish the stability characteristics of a particular type of bearing. The most significant is the critical mass, derived above. It is also possible to get some feel for stability from purely steady-state performances by examining the bearing attitude angle. The larger the attitude angle, the greater the cross-coupling influence and the worse the stability characteristics. Interpreting attitude angles, however, can prove misleading.

Figure 22 shows plots of the attitude angle γ versus the load parameter W for some of the different types of bearings previously described. The contradiction in these results is that the symmetric three-lobe bearing has higher attitude angles than the two-groove cylindrical bearing even though, as will be subsequently demonstrated, it has superior stability characteristics. The reason for the higher attitude angle is that the bearing is cavitating in the diverging region of the loaded pad, which results in a large shift in the journal. Whirl motion, however, is prevented by the accompanying lobes.

A more direct and accurate approach to establishing bearing stability is to determine the critical mass acting on the bearing. Figure 23 shows the results obtained for the fixed

CENTRIFUGAL PUMPS

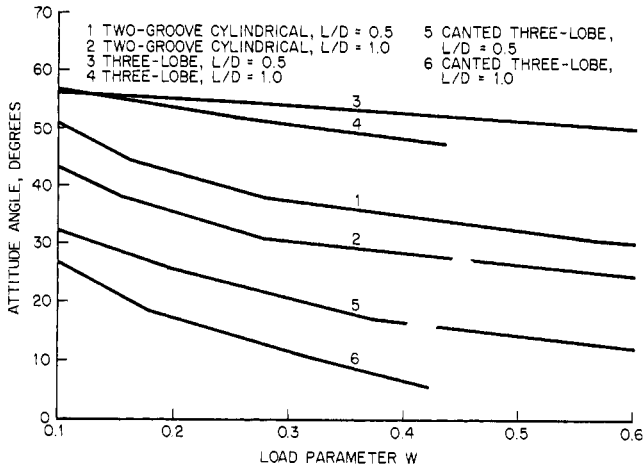


FIGURE 22 Attitude angle versus load parameter for various bearing types

bearing geometries previously discussed. For any particular bearing geometry, if the critical mass attributable to the bearing exceeds that of the data plotted, the bearing will be unstable. Thus, if the plotted point falls to the right of the bearing line, the bearing is stable; if it falls to the left, the bearing is unstable.

The critical mass is defined as follows:

$$M = m\omega R c^3 / 24\mu L^5 \quad (54)$$

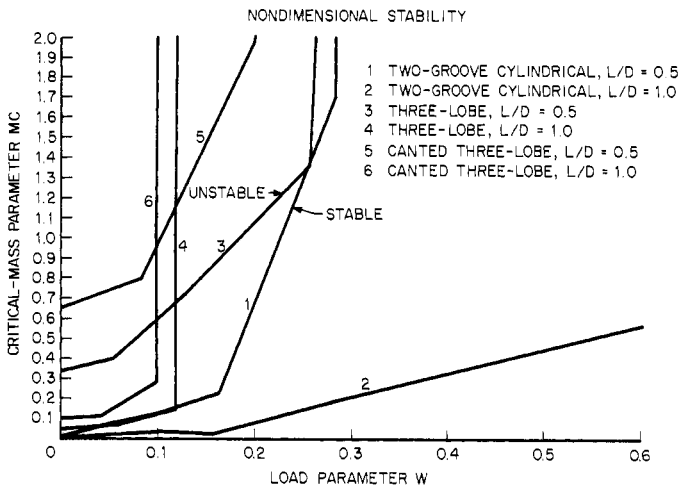


FIGURE 23 Critical mass versus load parameter for various bearing types

- where M = nondimensional critical mass
 m = dimensional critical mass, lb · s²/in (kg · s²/mm)
 ω = shaft speed, rad/s
 R = shaft radius, in (mm)
 c = bearing machined radial clearance, in (mm)
 μ = lubricant viscosity, lb sec/in² (Pa · s)
 L = bearing length, in (mm)

An examination of the curves in Figure 23 clearly indicates the superiority of the canted three-lobe bearing and the inferiority of the cylindrical configuration.

EXAMPLE Consider a high-speed bearing for which

- $N = 60,000$ rpm
 $\omega = 60,000\pi/30 = 6283$ rad/s
 $D = 1$ in (25.4 mm)
 $L = 0.5$ in (12.7 mm)
 $\mu = 1 \times 10^{-6}$ lb · s/in² (6.9×10^{-3} Pa · s)
 $C = 0.0005$ in (0.013 mm)
 $w = 100$ lb (45.35 kg)

$$W = \frac{wC^2}{6\mu\omega RL^3} = \frac{(100)(0.0005^2)}{(6)(1 \times 10^{-6})(6283)(0.5)(0.5^3)} = 0.0106$$

The value of M is obtained from Figure 22 and indicated in Table 4 for the various bearing types considered. Dimensional units are also given. Table 4 clearly indicates that the lobe bearings can permit significantly more attributable mass than the cylindrical bearing can. If a symmetric rotor is being supported by two bearings, the cylindrical bearings would be unstable if half the weight of the rotor exceeded 14.7 lb (6.67 kg). The half weights go to 258 lb (117 kg) and 501 lb (227.2 kg) for the symmetric three-lobe and canted three-lobe bearings, respectively. No mention has been made of the tilting-pad bearing because, for all practical purposes, these bearings are always stable.

OPERATING CONDITIONS THAT AFFECT BEARING STABILITY _____

Cavitation Figure 1 shows a pressure distribution in a journal bearing. Positive pressure is generated in the converging wedge because the journal is pumping fluid through a restriction. On the downstream side of the minimum film thickness, the journal is pumping fluid out of a diverging region. In this region, the pressure decreases. Either the pressure becomes negative, which is defined as pressure below the ambient pressure,

TABLE 4 Stability Comparison Chart

Bearing type	M	m , lb · s ² /in (kg · s ² /mm)	w' , lb (kg) ^a
Two-groove cylindrical	0.02	0.038 (6.79×10^{-4})	14.7 (6.7)
Symmetric three-lobe	0.85	0.668 (11.9×10^{-3})	257.9 (117.1)
Canted three-lobe	0.68	1.299 ($23.2 > 10^{-3}$)	501.4 (227.6)
$m = \frac{24M\mu h^5}{\omega Rc^3} = \frac{M(24)(10^{-6})(0.5^5)}{(6283)(0.5)(0.0005^3)} = 1.91M$			

^aThe w' represents the maximum mass in weight units that could act on the bearing.

or the film cavitates and decreases to atmospheric pressure as the lubricant releases entrapped air.

With respect to stability, cavitation is a more desirable condition than the development of negative pressure. From an examination of Figure 1, it can be seen that the negative pressure pulls the journal in an orthogonal direction and increases the cross-coupling. The more eccentric the bearing, the larger the negative pressure or cavitated region.

In lightly loaded bearings that are pressure-fed, negative pressures can occur because pressures in the divergent region have not approached atmospheric pressure. Cavitation does not occur, and the bearing is prone to instability. Thus, the feed pressure and load on a bearing are two additional parameters that affect stability.

Lobe bearings are often used because of their excellent antiwhirl characteristics. Figure 9 earlier showed two types of lobe bearings: symmetric and canted. The symmetric lobe bearing is designed so that, in the concentric position, the minimum film thickness occurs at the center of each lobe. Note that this permits a region of converging film followed by a region of diverging film. Thus, depending upon the ambient pressure, it is possible to have negative pressures in a symmetric lobe bearing. Under very high ambient conditions, a symmetric lobe bearing can go unstable.

The canted lobe bearing is designed to have a completely converging wedge and positive pressure throughout its arc length. Its stability characteristics are superior to those of the symmetric lobe bearing. Its steady-state characteristics are also superior.

Hybrid Bearings At times, externally pressurized bearings are resorted to for stability improvements. The philosophy is that externally pressurized bearings are not subject to high attitude angles, as is the case with hydrodynamic journal bearings. Although this is generally true, hybrid bearings can still be subject to considerable cross-coupling. Figure 11 showed a schematic arrangement of a hybrid bearing. Oil is fed through restrictors from an external source into pocket recesses. From there, it exits into the clearance region between recesses. Lubricant is also pumped into and out of recesses by the rotating shaft by a viscous drag in the same manner as with a purely hydrodynamic bearing.

Consider recess 2 in Figure 11. Oil is pumped from the shaft via a converging wedge, and it augments the pressure in the recess provided by the external system. The net result is a higher pressure in the bearing domain covered by recess 2 than would occur without rotation. Now consider recess 3. Here the journal is pumping fluid out of the recess into a diverging film, so that the hydrodynamic action tends to reduce the pressure in this recess domain. By similar reasoning, it can be shown that recess 4 operates at a lower pressure than recess 1. The net result of these variations in pressure due to rotation is that cross-coupling forces are introduced and the hybrid bearing may not prevent instability.

BEARING MATERIALS AND FAILURE MODES

Materials The most common material used for oil-lubricated fluid film bearings is babbitt. Tin- and lead-based babbitts are relatively soft materials and offer the best insurance against shaft damage. They also enable embedded dirt and contaminants without significant damage.

Two types of babbitts are in common use. One has a tin base (86 to 88%), with about 3 to 8% copper and 4 to 14% antimony. The other has a lead base with a maximum of 20% tin and about 10 to 15% antimony. The remainder is principally lead. The physical properties of babbitt are shown in Table 5. The primary limitations of babbitt are operating temperature (300°F [140°C] max) and fatigue strength. The chemical composition of various babbitt alloys are indicated in Table 6.

Tin-based babbitts have better characteristics than lead-based babbitts; they have better corrosion resistance, are less likely to wipe under poor conditions of lubrication, and can be bonded more easily than lead-based materials. Because of cost considerations, however, lead-based babbitts are widely used. The more widely used is the SAE 15 alloy containing 1% arsenic (refer to Table 6).

TABLE 5 Properties of Bearing Alloys

Bearing material	Brinell hardness at room temperature	Brinell hardness at 30°F (17°F)	Minimum Brinell hardness of shaft	Load carrying capacity, lb/in ² (kg/m ²)	Max operating temperature, °F (°C)
Tin-based babbitt	20–30	6–12	150 or less	800–1500 (5.62×10^5 – 10.55×10^5)	300 (149)
Lead-based babbitt	15–20	6–12	150 or less	800–1200 (5.62×10^5 – 8.44×10^5)	300 (149)

SOURCE: Ref. 8

Tin-based babbitts do not experience as many corrosion problems as lead-based babbitts. SAE 11 babbitts (containing 8% antimony and 8% copper) are used extensively for industrial applications.

Babbitts are bonded to a backing shell of another material, such as steel or bronze, because they are not a good structural material. The thinner the babbitt layer, the greater the fatigue resistance. In automotive applications where resistance to fatigue is important, babbitt thickness is from 0.001 to 0.005 in (0.02 to 0.12 mm). For pump applications, the thickness varies from $\frac{1}{32}$ to $\frac{1}{8}$ in (0.8 to 3 mm). The thicker layers provide good conformity and embedability.

Failure Modes The failure modes most commonly found are fatigue, wiping, overheating, corrosion, and wear.

Fatigue occurs because of cyclic loads normal to the bearing surface. Figure 24 shows babbitt fatigue in a 7-in (178-mm) diameter journal bearing from a steam turbine.

Wiping results from surface-to-surface contact and smears the babbitt, as shown in Figure 25. The usual causes of wiping are a bearing overload, insufficient rotational speed to form a film, and loss of lubricant.

TABLE 6 Composition* Percentages of Babbitts with SAE Classifications of 11 to 15

Element	SAE No. (Similar ASTM Spec.)				
	11 (None)	12 (B23, alloy 2)	13 (None)	14 (B23, alloy 7)	15 (B23, alloy 15)
Tin (mm)	86.0	88.2	5.0–7.0	9.2–10.8	0.9–1.2
Antimony	6.0–7.5	7.0–8.0	9.0–11.0	14.0–16.0	14.0–15.5
Lead	0.5	0.5	Remainder	Remainder	Remainder
Copper	5.0–6.5	3.0–4.0	0.5	0.5	0.5
Iron	0.08	0.08	—	—	—
Arsenic	0.1	0.1	0.25	0.6	0.8–1.2
Bismuth	0.08	0.08	—	—	—
Zinc	0.005	0.005	0.005	0.005	0.005
Aluminum	0.005	0.005	0.005	0.005	0.005
Cadmium	—	—	0.05	0.05	0.05
Others	0.2	0.2	0.2	0.2	0.2

*The percentage of minor constituents represents limiting values except as noted.

SOURCE: SAE Handbook, Society of Automotive Engineers. New York, 1960, p. 201.



FIGURE 24 Babbitt fatigue in a 7-in (178-mm) diameter turbine bearing (Westinghouse Electric Corp. *Photo originally reproduced in Ref. 8, p.18–19.*)



FIGURE 25 Bearing wipe on a 3-in (76-mm) diameter bearing due to temporary loss of lubricant (Westinghouse Electric Corp. *Photo originally reproduced in Ref. 8, p.18–25*)

Overheating is manifest by discoloration of the surface and cracking of the babbitt material. Corrosion is failure by a chemical action. It is more common with lead-based babbitts, which react with acids in the lubricant. Figure 26 shows corrosion damage for a 5-in (127-mm) diameter, lead-based babbitt bearing.

Wear results from contaminants in the film and is evidenced by scoring marks that may be localized or persist around a large circumferential region of the bearing.

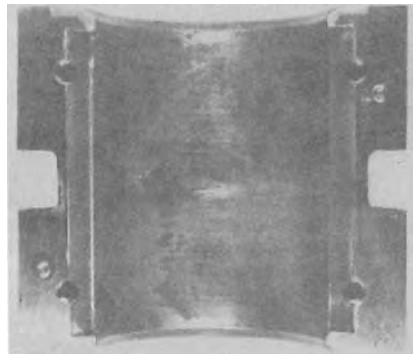


FIGURE 26 Corrosion on a 5-in (127-mm) diameter, lead-based babbitt bearing (Westinghouse Electric Corp. *Photo originally reproduced in Ref. 8, p. 18–21*)

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SECTION 6.3

CENTRIFUGAL PUMP MAGNETIC BEARINGS

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Magnetic bearings maintain the rotor of a pump in suspension through the forces of attraction of a magnetic circuit. Thus, although they bear up the weight and hydraulic loads of the impellers and the shaft, they are not really bearings in the traditional sense of the rotating and stationary surfaces bearing on one another. The supporting magnet circuit for each bearing includes stationary magnets in a stator surrounding the shaft, a laminated rotor that fits on the shaft, and the shaft itself. The stator consists of electromagnets in the traditional heteropolar design, and if a homopolar design is employed, permanent magnets can be added. Sensors monitor the position of the shaft and signal a controller to adjust the magnetic loads to keep the shaft to within about 0.001 in (25 μm) of the desired position.

Magnetic bearings are found in small, high-speed turbomachinery such as high-speed, multistage, axial-flow turbomolecular vacuum pumps¹. They were introduced into large turbomachinery in the early 1980s, mainly in gas compressors and turboexpanders. Their use and acceptance has grown slowly but steadily since then². Pump applications of a significant size have appeared and have confirmed the general position that magnetic bearings can provide a technically sound bearing with maintenance and operating advantages, including zero wear. However, due to the technical complexity of magnetic bearing systems, the economies of scale associated with production quantities are required to make these systems affordable.

Two representative magnetic-bearing-equipped pumps are summarized in Table 1. One is a multistage boiler feedwater pump³⁻⁶ and the other a single-stage double-suction hydrocarbon process pump⁷. The multistage pump was retrofitted with magnetic bearings (as shown in Figure 1) and is shown in Figure 2, together with another identical pump that still contains the oil-lubricated bearings—both installed in an electric generating station. The magnetic-bearing pump is not encumbered with the usual complexity of a bearing lubrication system.

TABLE 1 Example of magnetic-bearing-equipped pumps

Parameter	Multistage Pump	Single-Stage Pump
Power, hp (MW)	610 (0.46)	800 (0.6)
Rated speed, rpm	3,580	1780
Shaft weight, lb (kg)	520 (236)	732 (332)
Radial bearing design load, lb (kN)	800 (3.6)	Thrust end: 930 (4.1) Drive end: 1,415 (6.3)
Thrust bearing design load, lb (kN)	4,000 (17.8)	4,000 (17.8)
Number of stages	8	1

MAGNETIC BEARING PRINCIPLES

How Magnetic Bearings Work In an active magnetic bearing system, a stator composed of an array of stationary magnets, or electromagnetic coils, interacts with a ferrous rotor (or a ferrous sleeve on a nonferrous rotor) so as to suspend the shaft in a magnetic field (see Figure 3).

The position of the shaft is maintained dynamically through a continuous feedback system which comprises a position sensor, a controller, and an amplifier system (see Figure 4). Typically there are two radial bearings and one thrust bearing for a complete system. This system is tuned to the required characteristics of the pump, through a digital or analog controller (Figure 5), with the capability of adjusting the bearing stiffness and damping as a function of pump speed. Alarms and trips can be set at any required rotor offset or bearing load to provide the operator with warnings or to trip the drive unit as necessary. Figure 6 illustrates a typical bearing transfer function, showing a statically stiff bearing, with a dynamic stiffness over the operating speed range designed to meet the rotor dynamics requirements of the unit. The stiffness is then rolled off above the operating range to avoid excitation of higher modes in the rotor or stator.

Controller redundancy can be provided with the control loops switching to backup units upon sensing a failure. Also, backup power supplies should be provided, either through alternate sources or a battery system. Typically the power required is only 1 or 2 kW or less.

A catcher (also known as auxiliary, backup, or touchdown) bearing (indicated in Figure 1) is required to protect the rotor stator interface during maintenance and in the event of loss of power or a severe transient beyond the force capability of the bearing. Typically, the catcher bearing is designed for 5 to 20 lifetime drops from full speed, and will be a readily replaceable rolling element or sleeve bearing. The pump of Figures 1 and 2 has rolling

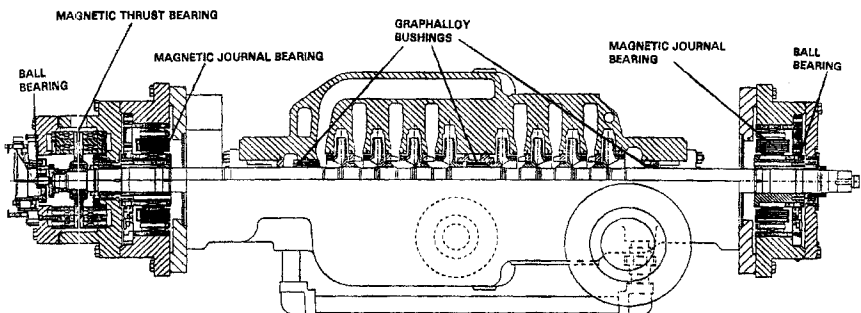


FIGURE 1 Magnetic bearing configuration in multistage pump⁶

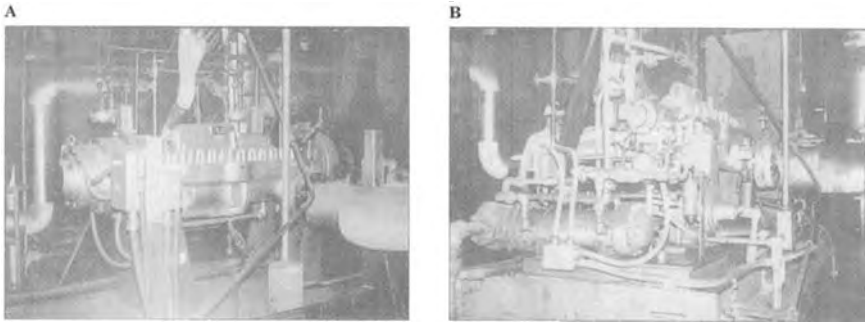


FIGURE 2 Multistage pumps installed in boiler feed service [610 hp (0.46 MW)]: a) pump with magnetic bearings; b) the same pump with oil-lubricated bearings. (Flowserve Corporation)

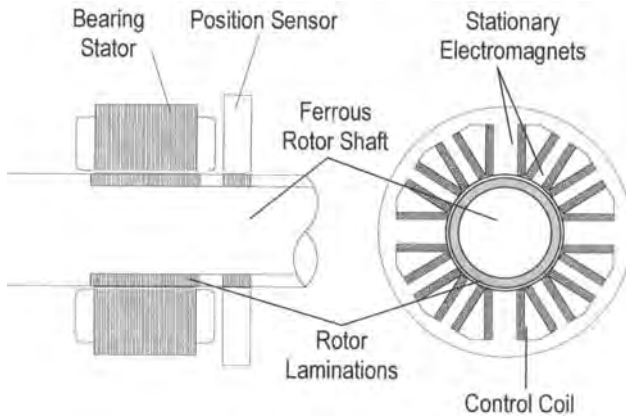


FIGURE 3 Typical radial electromagnetic bearing (Axial thrust bearings have the stator coils arranged in a disk configuration, a ferrous rotating disk being supported in the resulting magnetic circuit.)

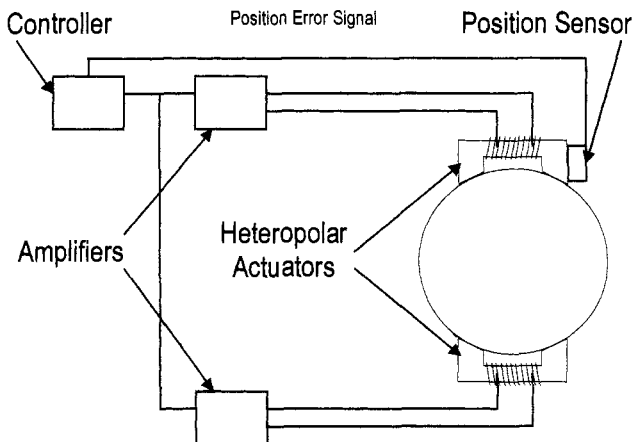


FIGURE 4 Typical system control loop

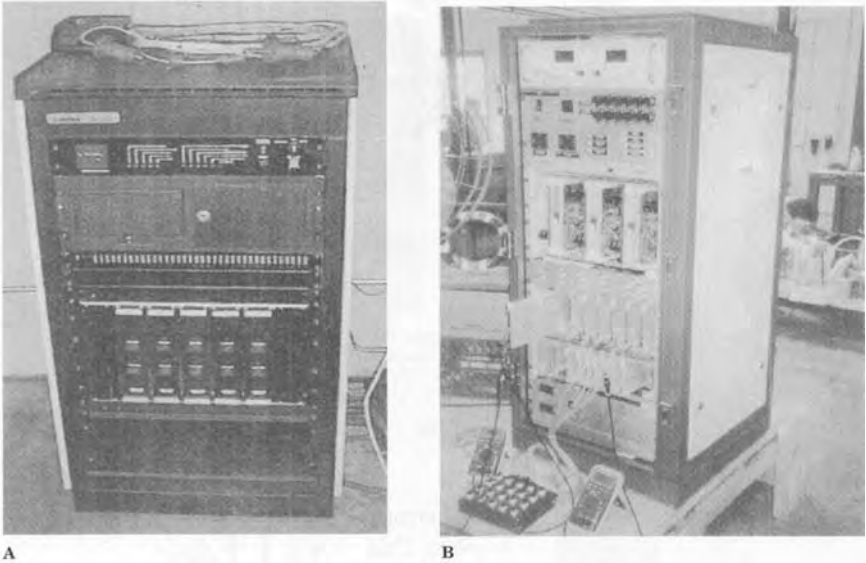


FIGURE 5 Controllers for magnetic bearings, containing rectifier and amplifiers: a) digital controller; b) analog controller

element catcher bearings. The radial clearances G1 (see Figure 7) between the magnetic bearing stator and rotor are of the order of 20 to 40 mil (0.5 to 1 mm), and those in the catcher bearings (G2) are about half of that value.

Reasons for Using Magnetic Bearings There are several reasons to use magnetic bearings in pumps. While any one of these reasons may not be sufficient justification on its own, together they can provide a strong justification.

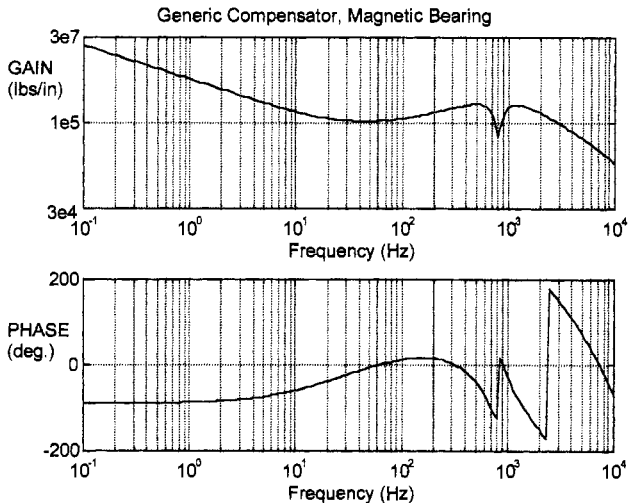


FIGURE 6 Bearing transfer function

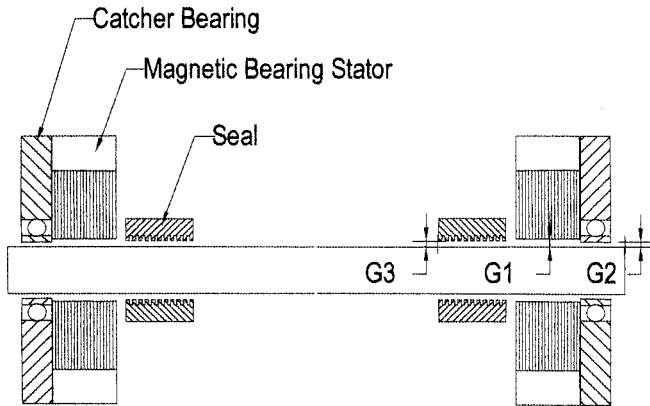


FIGURE 7 Clearance arrangement: Seal ring clearance G3 is greater than catcher-bearing clearance G2, but is less than magnetic-bearing clearance G1.

Reliability is a key incentive. The components of a magnetic bearing are essentially the same components as are found in an electric motor: laminations and coils. Because no wear is involved due to the lack of contact, these components will generally last the life of the equipment involved. Thus maintenance of a magnetic bearing system is transferred from mechanical components inside the pump to the external controller, which has plug-in card replacement maintenance. Pump reliability is therefore improved, whereas repair times and costs are reduced.

Reduced power consumption is a second advantage, with the elimination of all losses associated with fluid film bearings and oil pumping equipment. This is replaced by the smaller power requirements of the bearing controller. Further, if the lifting force is supplied by a permanent magnet, supplemented by an active control circuit, this power requirement can be even smaller.

The ability to submerge the bearing in the pump fluid is a major advantage that allows the outboard mechanical seal to be eliminated, thereby eliminating maintenance and replacement of this seal⁶. [This was not done for the pumps of Table 1 (and Figures 1 and 2), because in both cases magnetic bearings were retrofitted to existing machines.]

More indirect savings are also possible in two other areas. Rotor dynamics can be controlled through the ability to adjust stiffness and damping as a function of pump speed, allowing higher imbalance without the need for shutdown. The diagnostic output inherent in the information provided in the controller can be fed into the overall plant operating system and the short-term and long-term health of the pump and the system can be monitored. This is done by inferring seal wear, transient hydraulic loads, and so on.

The actual figures for the savings possible due to the previous advantages are very pump- and system-specific, and general numbers are not very useful. Reference 9 has developed methodology for considering the economic effect of the types of advantages given.

Main Types of Magnetic Bearings and Their Selection There are two main types of magnetic bearings: passive and active. Passive bearings rely only on permanent magnets in repulsion and provide low stiffness, low damping, and no ability to control either of these parameters. Passive bearings are not applicable to pumps for this reason.

Active bearings using the feedback system previously described are essential for pump applications. Within the active bearing systems are the options of heteropolar and homopolar and of electromagnetic and permanent magnet bias. The principles of the heteropolar and homopolar approaches are shown in the Figure 8.

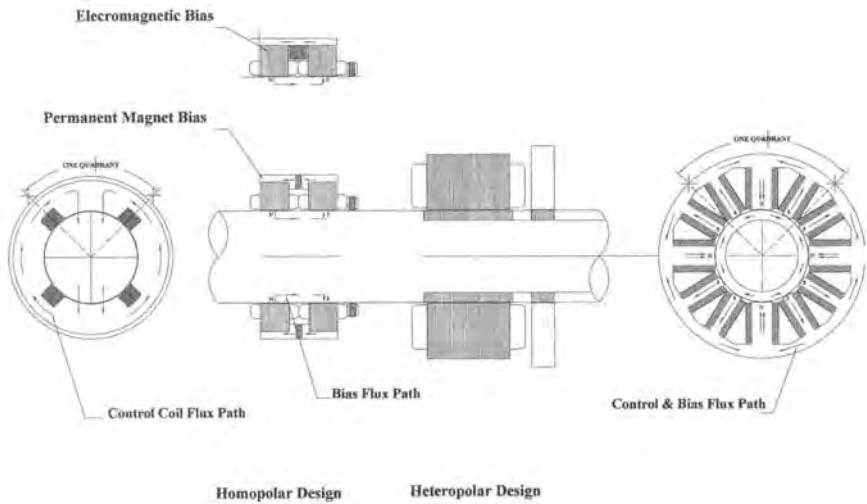


FIGURE 8 Heteropolar and homopolar bearings

The main difference between the two types is that in the heteropolar design, the bias and control flux flow in the same magnetic circuit radially through the rotor, whereas in the homopolar design the bias flux flows axially along the rotor and only the control flux flows radially through the rotor.

The homopolar design has two options for providing the bias flux for the bearing system¹⁰. One is to use an electromagnetic effect, and the other is to use a permanent magnet. The permanent magnet generation in the homopolar configuration results in a more linear relationship between force and distance. In a simple magnetic circuit, the attraction force of a magnet on a ferromagnetic target decreases as the square of distance (the target cuts $\frac{1}{4}$ as many flux lines at twice the distance). With the permanent magnet in a homopolar circuit, the effect of the air gap is therefore reduced.

DESIGN CONSIDERATIONS

Design Loads The specification of a magnetic bearing requires a different approach to that required for a conventional bearing system. This section identifies the key areas where these differences occur.

The rotor can move considerably within the magnetic bearing and catcher bearing clearances, typically up to 10 mil (0.25 mm), before any contact is involved. Thus the clearances G3 in the internal ring-seal system (see Figure 7) become the key controlling parameter in setting the bearing clearance design limits and in the degree of motion permitted during transients. Thus, very early in the design process, the magnetic bearing clearances, catcher bearing clearances, and sealing-ring clearances must be optimized with due consideration for manufacturing tolerances and assembly concerns.

Synchronous filters or open-loop control methods can be used to handle imbalance loads so that the degree of imbalance acceptable (based on allowable bearing loads and shaft motion within the catcher bearing clearances) can often be significantly greater than in conventional bearings.

There is an area where careful analysis is needed for each new pump configuration. A conventional bearing, if overloaded, will accept the load with a higher wear rate, but a magnetic bearing has a sharp cut-off at the point where the flux saturation level is

reached. At that point, any additional load will be transferred to the catcher bearing. Thus it is important to include all loads in the design specification, with the appropriate margins. All applications to date have generally shown a) that there are loads which were mistakenly considered to be insignificant, or b) that the values of the loads were underestimated due to lack of knowledge.

Magnetic bearings also must be carefully designed to handle transient loads, some of which may occur only once in a lifetime. Two approaches can be taken. One is to over-design the bearing to handle the load without contacting the catcher bearing; the other is to allow momentary contact with the catcher bearing. Examples of these types of loads are hydraulic loads, seal touchdown loads, system loads such as water hammer, valve operation, pump switchover, pump to driver alignment loads, and seismic loading.

Rotordynamics Considerations Magnetic bearings have the capability to control the rotor dynamics of a pump very effectively. If the pump is running below its first flexible mode, this is usually straightforward. If the pump has to traverse a flexible mode, the position of the bearings and the position sensor must be such that the modes can be recognized by the position sensor, and the bearings can exert a positive restoring force to control the mode. That is, the position sensors and bearings must not be positioned at or close to a node and certainly not positioned on opposite sides of a mode. Thus, rotordynamics considerations should be taken care of very early in the configuration of the system.

The two typical rigid body modes are shown in Figure 9. The location of the bearing and position sensor is not usually an issue for these modes, but the flexible modes require that the position sensor and bearing be positioned in such a way that the rotor deflection can be measured and the bearing can exert the necessary restoring force and damping to control the mode.

Magnetic bearing controllers are programmed with a transfer function designed specifically for the pump. This transfer function, or control algorithm, provides the necessary stiffness and damping at all operating speeds to control the rotor as previously described.

The key requirements of the transfer function are that it

- Provides correct damping and stiffness to handle the rotor rigid body modes
- Provides sufficient force with the appropriate control bandwidth to handle the rotor flexible modes below the maximum operating speed

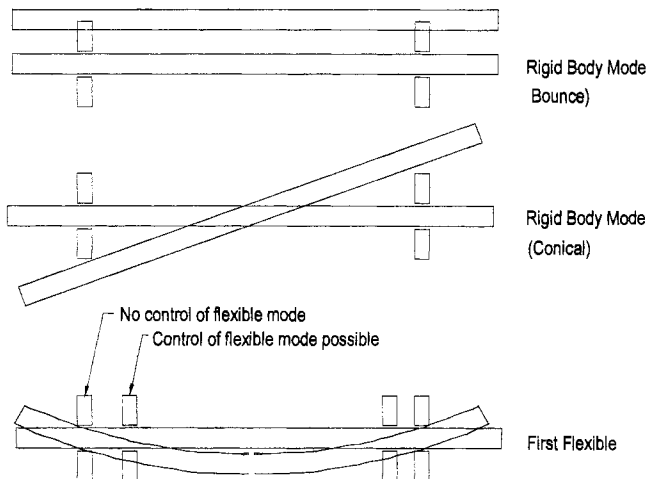


FIGURE 9 Flexible mode design considerations

- Does not excite any rotor modes above the maximum operating speed
- Does not excite any stator vibration modes at any frequency
- Takes into account the stiffness and damping contribution from wear rings and inter-stage annular seals

Auxiliary or Catcher Bearings However reliable magnetic bearings become, a landing surface for maintenance will be required. Further, designing a bearing that will take all transient loads without any possibility of overload will usually result in over-sized and costly bearings. Hence catcher, bearings are expected to be required. The design limits for catcher bearings are usually established by calculating the forces that will result from a drop at operating speed.

For applications running at significant speeds, a nonlinear analysis is required to determine the motion and loading on the bearing during such a drop. The key design considerations are impact loads, heat generation during the rundown, and the response to imbalance if, when running on the catcher bearings, any critical speeds have to be traversed.

Rolling element bearings have typically been used as catcher bearings; however, sleeve bearings and bushings have been used in several applications, and are better suited to a submerged application.

MAGNETIC BEARING SIZING

The fundamental magnetic bearing sizing problem is to define the pole area at the bearing air gap that is necessary to achieve the desired force capacity without saturating the selected pole materials. Given the pole area, the minimum stator outside diameter and maximum rotor inside diameter can be determined using simple algorithms to ensure that no other part of the magnetic circuit saturates. The stator geometry must also include sufficient volume for the control and bias coils.

Approximate size and load capability Table 2 contains rough approximations for the dimensions of magnetic bearings. (Refer to Figure 1.) These are based on the experience with the pumps of Table 1. Also included are the unit load capabilities, which are given a) for radial bearings in terms of radial load F_r , divided by the projected area DL of the active polar area of the bearing at the air gap, and b) for axial thrust bearings in terms of the axial load F_z , divided by the active pole area of the runner disk between the inner diameter D_i and the outer diameter D_o . These approximations provide the designer and user with an idea of the design configuration of a magnetic-bearing-equipped pump. The

TABLE 2 Approximate magnetic bearing sizing relationships

Sizing Parameter	Heteropolar	Homopolar
<i>Radial Bearings</i>		
a) Dimensions, in multiples of the shaft diameter		
Air gap diameter, D	2	1.5
Stator outer diameter	4	3
Overall axial length	2	1.5
Active axial length of poles at air gap, L	0.9	0.8
b) Unit load capability, F_r/DL , lb/in ² (MPa)	40 (0.28)*	60 (0.41)
<i>Axial Thrust Bearings</i>		
a) Diameter ratio D_i/D_o of active pole area	0.5	0.5
b) Unit load capability, $F_z / [\pi(D_o^2 - D_i^2)]$, lb/in ² (MPa)	50 (0.34)*	70 (0.48)

*Higher with special lamination material

low unit loads of magnetic bearings result in more space being needed for them in comparison to conventional bearings.

Theory of One-Dimensional Sizing To perform more accurate, in-depth sizing, the theory of both heteropolar and homopolar magnetic bearings is applied. Magnetic bearing sizing and geometry programs normally use simple one-dimensional magnetic circuit theory to obtain initial sizing and perform design iterations. This initial sizing is then followed up with design analysis using 2D or 3D magnetic FEA analysis to verify the design. The basis of the classic one-dimensional sizing for a magnetic bearing is discussed next, first for the heteropolar bearing and then for the homopolar bearing.

HETEROPOLAR BEARING

- a. *Magnetic circuit*—The basic magnetic circuit equation, derived from Ampere's Loop Law, is

$$MMF = \Phi R \quad (1)$$

where MMF = magnemotive force

Φ = magnetic flux

R = path reluctance

A sketch of one quadrant (one electromagnet) of a heteropolar bearing is shown in Figure 10.

Assuming the air gap area and path areas are equal, Eq. 1 becomes

$$2Ni = BA \left(\frac{g}{\mu_o A} + \frac{g}{\mu_o A} + \frac{l_{stat}}{\mu_o \mu_{r, stat} A} + \frac{l_{rot}}{\mu_o \mu_{r, rot} A} \right) \quad (2)$$

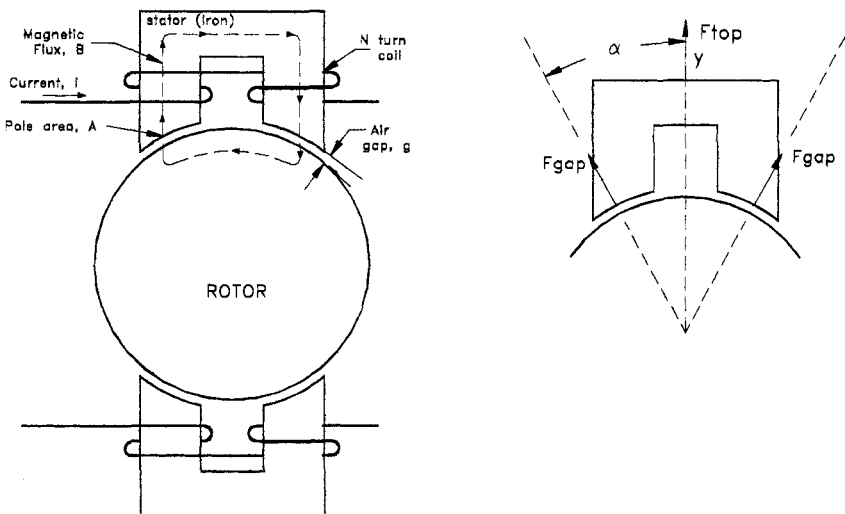


FIGURE 10 Simple electromagnetic circuit for one quadrant of a heteropolar magnetic bearing

Where N = number of turns per pole

i = coil current

B = magnetic flux density

A = pole area

g = air gap

μ_o = permeability of free space

μ_r = relative permeability

l = iron path length

If the iron permeability is high relative to the air gap ($\mu_{r, \text{rot}}, \mu_{r, \text{stat}} \gg \mu_o$), the iron reluctance terms are insignificant and the following equation can be obtained for the flux:

$$B = \frac{Ni\mu_o}{g} \quad (3)$$

b. *Force calculation*—The basic force equation for an air gap is

$$F_{\text{gap}} = \frac{B^2 A}{2\mu_o} \quad (4)$$

This equation assumes negligible leakage and fringing and that the flux density is uniform in the air gap. The combined vector force along the center of the bearing for the two air gaps of the top magnet is

$$F_{\text{top}} = 2F_{\text{gap}} \cos \alpha \quad (5)$$

Substituting from Eq. 4 gives

$$F_{\text{top}} = \cos \alpha \frac{B^2 A}{\mu_o} \quad (6)$$

If the saturation flux density, B_{sat} , of the iron material is used for B , Eq. 6 defines the load capacity as a function of pole area for a heteropolar magnetic bearing.

c. *Linearization of the force/current characteristic*—Substituting from Eq. 3 gives:

$$F_{\text{top}} = \cos \alpha \frac{A}{\mu_o} \left(\frac{Ni\mu_o}{g} \right)^2 = \cos \alpha AN^2 \mu_o \frac{i^2}{g^2} = k_1 \frac{i^2}{g^2} \quad (7)$$

$$k_1 = \cos \alpha AN^2 \mu_o$$

Thus, the force in a given magnet is proportional to the square of the current, a result that makes the bearing more difficult to control. Additionally, a single electromagnet can only apply a force in one direction (an attractive force). For these two reasons, opposed electromagnets are used together with a bias current (or flux) in each coil.

The current relationship is

$$i = i_{\text{bias}} + i_{\text{con}}$$

to increase the force,

$$i = i_{\text{bias}} - i_{\text{con}}$$

to decrease the force.

The rotor may also be off-center in the air gap, described by

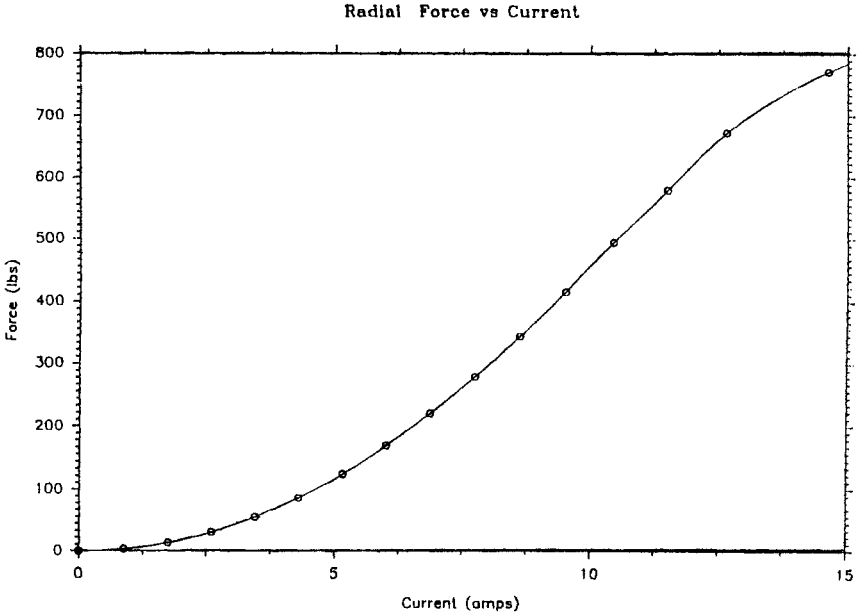


FIGURE 11 Force versus current for radial bearing of multistage pump ($\text{lb} \times 4.448 = \text{N}$)

$$g = g_0 - y \quad \text{top air gap}$$

$$g = g_0 + y \quad \text{bottom air gap}$$

Applying Eq. 7 to both top and bottom electromagnets yields

$$F_y = F_{\text{top}} - F_{\text{bot}} = k_1 \left[\left(\frac{i_{\text{bias}} + i_{\text{con}}}{g_0 - y} \right)_{\text{top}}^2 - \left(\frac{i_{\text{bias}} - i_{\text{con}}}{g_0 + y} \right)_{\text{bot}}^2 \right] \quad (8)$$

With the rotor centered ($y = 0$), this can be reduced to

$$F_y = 4 \frac{k_1}{g_0^2} i_{\text{bias}} i_{\text{con}} \quad (9)$$

Thus the net bearing force is proportional to the control current. Figures 11 and 12 are examples of measured force versus current for radial and axial thrust bearings respectively. These measurements were made on the magnetic bearing of the multistage pump in Table 1 (refer to Figures 1 and 2).

- d. *Force constant and negative stiffness*—Eq. 8 can be also be linearized for small motion about the center ($y \ll g_0$) by differentiating with respect to i_{con} and y , the two quantities in Eq. 8 that can change in normal operation of the bearing. The result is

$$F_y = -k_f i_{\text{con}} - k_n y \quad (10)$$

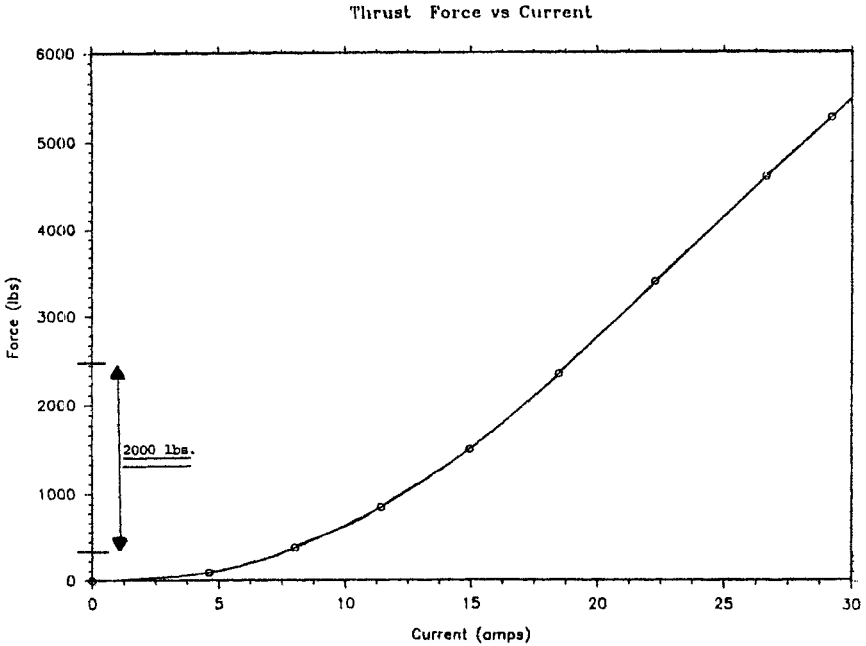


FIGURE 12 Force versus current for axial thrust bearing of multistage pump (lb $\times$ 4.448 = N)

where

$$k_f = \frac{-4k_1 i_{bias}}{g_0^2} = \text{force constant or current stiffness} \quad (11)$$

$$k_n = \frac{-4k_1 i_{bias}^2}{g_0^3} = \text{negative stiffness or position stiffness} \quad (12)$$

The position stiffness is the passive stiffness of the bearing with a bias field but with no control current. The position stiffness is always negative, indicating that if the rotor is displaced from center, it will be pulled further from its equilibrium position if no control current is applied. The force constant defines the relationship of the control force to current (lb/amp or N/amp) with the bearing centered. It is also negative, indicating that applying a control current pulls the rotor from its centered position. Many practitioners use positive values for the force constant and the position stiffness as a matter of convenience. In this case, the minus signs in Eq. 10 become plus signs.

The control current is determined by the measured displacement, y , and the characteristics of an adjustable sensor/compensator/amplifier transfer function:

$$G_{con}(s) = \frac{i_{con}}{y} \quad (13)$$

Substituting Eq. 11 into Eq. 10 gives

$$Fy = -k_F G_{con}(s)y - k_n y \quad (14)$$

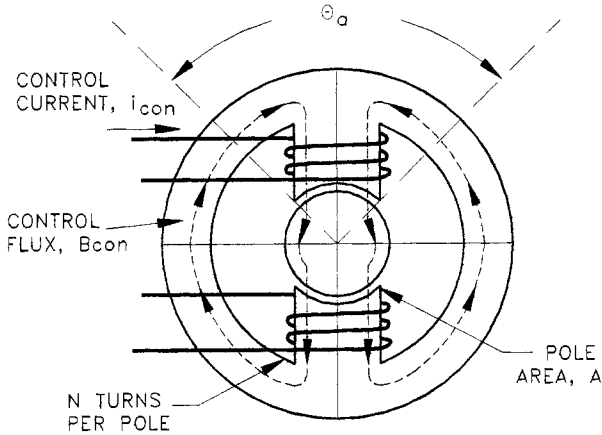


FIGURE 13 Control magnetic circuit for one axis of a homopolar magnetic bearing

The characteristics of the control transfer function $G_{con}(s)$ are determined in order to stabilize the rotor/bearing system.

HOMOPOLAR BEARING In the homopolar bearing, a bias flux is used for linearization just as in the heteropolar bearing; however, the bias flux and control flux follow different paths. Additionally, the bias flux can be generated by either a permanent magnet (most common) or an electromagnetic coil. The use of a permanent magnet for bias reduces power consumption and makes the bearing more linear at large position offsets.

- a. *Magnetic control circuit*—The control circuit for one half of the homopolar bearing, shown in Figure 13, is across the top air gap, through the rotor, out the bottom air gap, and around the back iron.

When Eq. 1 is applied to this circuit, the result is similar to Eq. 2:

$$2Ni_{con} = B_{con}A \left(\frac{g}{\mu_o A} + \frac{g}{\mu_o A} + \frac{l_{stat}}{\mu_o \mu_{r,stat} A} + \frac{l_{rot}}{\mu_o \mu_{r,rot} A} \right) \quad (15)$$

Again, if the iron permeability is high relative to the air gap, this can be reduced to the following:

$$B_{con} = \frac{Ni_{con}\mu_o}{g} \quad (16)$$

- b. *Magnetic bias circuit*—The magnetic circuit equation for the bias circuit, shown in Figure 14, is

$$\frac{B_r l_m}{\mu_o \mu_{r,mag}} = B_{bias} A \left(\frac{g}{\mu_o A} + \frac{g}{\mu_o A} + \frac{l_m}{\mu_o \mu_{r,mag} A_m} \right) \quad (17)$$

Where B_r = residual induction of the magnet

l_m = axial length of the magnet

$\mu_{r,mag}$ = relative permeability of the magnet (≈ 1.0 – 1.05)

A_m = magnet cross-sectional area per quadrant

With the magnet permeability assumed to be 1.0, this can be reduced to

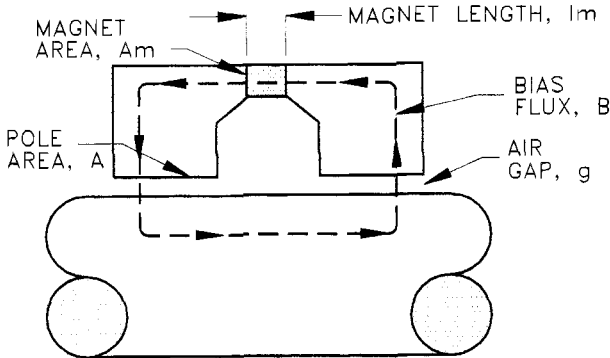


FIGURE 14 Bias magnetic circuit for one quadrant of a homopolar magnetic bearing

$$B_{bias} = \frac{B_r}{\frac{2g}{l_m} + \frac{A}{A_m}} \quad (18)$$

- c. *Force calculation*—The basic force equation for an air gap is given by Eq. 4. In the homopolar bearing, the pole face covers a broader arc of the rotor than in the heteropolar bearing; therefore, the air gap force must be integrated over the surface to obtain the desired vector force, F_{top} . Both the front and back control stacks must also be included. The result is

$$F_{top} = 2C_g F_{gap}$$

Where:

$$C_g = \frac{2\sin(\theta_a/2)}{\theta_a} \quad (19)$$

This result assumes negligible leakage and fringing and that the flux density is uniform in the air gap. Substituting from Eq. 4 into Eq. 19 gives

$$F_{top} = C_g \frac{B^2 A}{\mu_o} \quad (20)$$

If the saturation flux density, B_{sat} , of the iron material is used for B , Eq. 20 defines the load capacity as a function of pole area for a magnetic bearing.

- d. *Linearization of the force/current characteristic*—The air gap flux density in the homopolar bearing is the superposition of the bias and control flux. The control coils and permanent magnet polarity are arranged such that when the control flux adds to the bias in the top air gaps, the control subtracts from the bias in the bottom air gaps. Thus the net vertical force is

$$F_y = F_{top} - F_{bot} = \frac{C_g A}{\mu_o} [(B_{bias} + B_{con})_{top}^2 - (B_{bias} - B_{con})_{bot}^2] \quad (21)$$

This can be reduced in a similar manner as before to produce

$$F_y = \frac{4C_g A}{\mu_o} B_{bias} B_{con} \quad (22)$$

Substituting from Eq. 16

$$F_y = \frac{4C_g AN}{g} B_{bias} i_{con} \quad (23)$$

Thus the control force is proportional to the control current as desired.

- e. *Force constant and negative stiffness*—The expression given in Eq. 10 for the heteropolar bearing also applies to the homopolar bearing:

$$F_y = -k_f i_{con} - k_n y \quad (24)$$

The definition of the force constant is

$$k_f = \frac{-4C_g AN B_{bias}}{g_0} = \text{force constant or current stiffness} \quad (25)$$

The expression for negative stiffness is not easily reducible to analytical form due to the complexity of the bias circuit. However, the existence of the permanent magnet in the bias flux path as a fixed and large reluctance improves the linearity of this bearing for off-center operation. Eqs. 13 and 14 for the heteropolar bearing apply to the homopolar bearing as well.

INSTALLATION AND TUNING

Mechanical Installation Magnetic bearings have the advantage that they can be set, by adjusting offsets in the controller, to center the rotor on the magnetic bearing stator, the catcher bearing, the seals, or any other mechanical reference in the pump. The rotor can even be offset vertically to cancel the effect of gravity, thus reducing static power requirements to near zero. Careful consideration is needed during design to decide which is the best approach.

The position sensors in the bearings can also be used to measure the clearances between the rotor and any physical stops such as a sealing ring, without disassembly.

Tuning The rotor dynamics analysis performed during the early design stage will be used to determine the initial controller compensation, which will have a transfer function matched to the pump requirements. During initial testing, this transfer function will have to be adjusted to match the as-built dynamics of the rotor and support structure.

Normally the rotor will be accurately modeled and little change will be needed. If there are shrink fits or bolted joints, there may be some stiffness variation from the theoretical model. This may require on-site controller compensation adjustment. However, the stator is often a complex structure, and adjustments may be needed to avoid the excitation of stator modes.

DIAGNOSTICS AND USER INTERFACE

Diagnostic Capabilities The controller, in order to function, must analyze a continuous stream of information on the shaft location in each of the five control axes, two for each radial bearing and one for the thrust bearing. This information can be accessed for external diagnostic use, as can the corresponding information on bearing current, from which can be inferred the bearing load. This diagnostic information can be a very useful source of information on the health of the pump, its mechanical components, and on the system it is operating in.

Interface Requirements The magnetic bearing system controller can also be interfaced with the plant control system with the following type of logic:

- No drive unit start without levitation
- No delevitation at speed
- Rotor offset and bearing load alarms
- Rotor offset and bearing load driver trips, with possible time delay

RELIABILITY AND MAINTENANCE

Bearing Cartridges As explained earlier, the reliability of the bearing stator and rotor components should be such as to provide lifetime service.

Controller The main life-limiting component in the controller is likely to be the amplifier. In a redundant system, online replacement is possible without loss of levitation. In a nonredundant system, a preventative maintenance approach should be used for this component.

OPERATING EXPERIENCE

Multistage Boiler Feed Pump The multistage pump of Table 1 (refer to Figure 1) was installed as one of three otherwise identical pumps (two in parallel, one standby) in an electric utility generating plant (refer to Figure 2). The objective was to show that magnetic bearings would work in a typical field application of a pump of significant power level. This project took the first step of replacing the conventional bearings in this 610 hp (0.46 MW) eight-stage centrifugal pump, which were outboard of the pump itself, and replacing them with heteropolar active magnetic bearings without any major design changes⁶. This was seen as the first of two steps, the second being a project where the bearings would be submerged in the operating fluid, allowing one seal system to be replaced⁹.

As with many magnetic bearing projects, the main lesson learned was that the transient bearing loads could not always be predicted ahead of time, and that the magnetic bearings gave very precise and important feedback of this information. Table 3 contains the design and field data. In this case, the 2500 lb (11 kN) transient load occurred when the plant underwent a suction pressure transient in which the available NPSH became so low that the first stage of this horizontally-opposed staging configuration (refer to Figure 1) apparently lost pressure rise completely⁶. The axial thrust of this stage was accordingly lost, destroying the intended axial hydrodynamic thrust balance of the pump. (See Sections 2.1.1 and 2.2.1.) Nevertheless, the conservative design of the thrust bearing enabled it to accommodate this load.

TABLE 3 Operating experience with multistage pump

Bearing	Expected Load, lb (kN)		Design Load, lb (kN)	Actual Load, lb (kN)	
	Steady	Transient		Steady	Transient
Radial	280 (1.2)	560 (2.5)	800 (3.6)	280 (1.2)	580 (2.6)
Axial	1,000 (4.4)	2,000 (8.9)	4,000 (17.8)	1,100 (4.9)	2,500 (11.1)

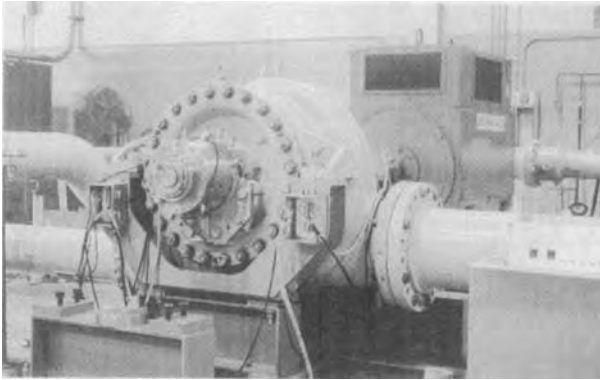


FIGURE 15 Single-stage double-suction 10 × 26 size process pump [800 hp (0.6 MW)] with magnetic bearing (Reference 7)

Single-Stage Process Pump The 800 hp (0.6 MW) single-stage double-suction process pump of size 8 × 26 (200 mm × 660 mm) of Table 1 was retrofitted with homopolar bearings⁷.

Closed-loop testing was conducted in the pump manufacturer's facility (see Figure 15). As indicated in Table 4, the results showed that a substantial operating margin exists for the axial thrust bearing. Greater design capacity would provide the same margin with respect to radial loads.

The margins evident in Table 3 for the multistage pump may appear excessive, but until more operating knowledge about such pumps is acquired, it would appear that this degree of conservatism in designing magnetic bearings for pumps is merited.

COSTS

Backed by the vision of complete magnetic suspension and the attendant benefits, magnetic bearing technology has been proven and demonstrated in pumping machinery. However, the major deterrent to further application of this technology is cost. Retrofitting the magnetic bearings to the multistage pump described above cost at least twice the price of the pump itself, and this included the analog controller, which accounted for about a third of the retrofit cost. Digital controllers, a later development, are one-third the size and cost of analog controllers; this has significantly reduced the overall cost. Much of this overall cost is in the engineering of the magnetic bearings, which includes matching this system to the rotordynamic characteristics of the pump. Use of the same system in quantity production would reduce the cost by up to 80%.

TABLE 4 Design and experimental loads for the single-stage pump

Bearing	Expected Load, lb, (kN)	Design Load, lb (kN)	Actual Load, lb (kN)
Radial	953 (4.2)	1,430 (6.4)	1,450 (6.4)
Axial	1,000 (4.4)	4,000 (17.8)	2,100 (9.3)

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JET PUMPS

SECTION 7.1

JET PUMP THEORY

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INTRODUCTION

The jet pump transfers energy from a liquid or gas primary fluid to a secondary fluid. The latter may be a liquid, a gas, a two-phase gas-in-liquid mixture, or solid particles transported in a gas or a liquid. Examples of all these combinations have been reported in the technical literature. Reference 1, the major bibliography in this field, contains over 400 abstracts. Although the terms “ejector” and “eductor” are also applied, the term “jet pump” will be used here. The jet pump offers significant advantages over mechanical pumps: no moving parts for improved reliability, adaptability to installation in remote or hazardous environments, simplicity, and low cost. The primary drawback is efficiency: both frictional losses and unavoidable mixing losses are incurred. Nevertheless, careful design can produce pumps with efficiencies on the order of 30–40%. The jet pump in Figure 1 is typical of liquid-jet pumps and low Mach-number gas-jet/gas pumps. Compressible-flow pumps, for example, steam-jet ejectors, employ converging-diverging nozzles for full expansion of the jet.

NOMENCLATURE

- A = area, ft^2 (m^2)
- A_w = throat wall area, ft^2 (m^2)
- C = velocity of sound, ft/sec (m/s)
- CR = cavitation resistance
- D = diameter, ft (m)

- E = energy rate, ft lb/sec (joule/s)
 K = friction loss coefficient
 $^{\circ}\text{K}$ = absolute temperature, Kelvin
 LJL = liquid-jet liquid pump
 LJG = liquid-jet gas compressor
 LJGL = liquid-jet gas and liquid pump
 M = liquid/liquid flow ratio, Q_2/Q_1
 MN = Mach number
 N = pressure ratio, LJL jet pump
 NPSH = net positive suction head
 P, P = pressure: static, total psia (kPa abs.)
 P_v = vapor pressure, psia (kPa abs.)
 Q = volumetric rate, ft^3/sec (m^3/s)
 R = gas constant, ft lbs/slug $^{\circ}\text{R}$ (joules/kg $^{\circ}\text{K}$)
 $^{\circ}\text{R}$ = absolute temperature, Rankine
 S = density ratio, ρ_2/ρ_1
 T = temperature $^{\circ}\text{R}$ ($^{\circ}\text{K}$)
 V = velocity, ft/sec (m/sec)
 W = work rate, ft lbs/sec (joules/s)
 Z = jet dynamic pressure, psi (kPa)
 a = diffuser area ratio, A_d/A_d
 b = jet pump area ratio, A_n/A_t
 $c = A_{2G0}/A_n = (A_t - A_n)/A_n = (1 - b)/b$
 m = mass flow rate, slugs/s (kg/s)
 s = seconds
 psi = pounds per square inch
 psia = pounds per square inch, absolute
 r_v = gas/liquid vol. flow rate ratio Q_G/Q_2
 r_{v0} = gas/liquid vol. flow rate ratio at 0: Q_{G0}/Q_2
 r_m = gas/liquid mass flow-rate ratio
 sp = nozzle-to-throat spacing, ft(m)
 sp/D_{th} = spacing, throat diameters
 η = efficiency
 γ = gas density ratio at s , ρ_{Gs}/ρ_1
 $\gamma\phi_s = m_{Gs}/m_1$
 ρ = density, slugs/ ft^3 (kg/ m^3)
 τ = shear stress, psi (kPa)
 ϕ = gas flow ratio Q_G/Q_1
 ϕ_s = gas flow ratio Q_{Gs}/Q_1 at s

Subscripts

- 1 = liquid primary flow
 2 = liquid secondary flow

G = gas secondary flow
 $2G$ = bubbly secondary flow
 $2G0$ = bubbly sec. flow at 0
 mep = maximum efficiency point
 op = operating point
 L = limit for cavitating flow
 3 = combined fluids 1, 2, and G
 i, s, n = locations (see Figure 1)
 $0, t, d$ = locations (see Figure 1)
 $c0$ = flow-ratio cut off
 f = friction loss
 n = nozzle
 en = throat entry
 th = mixing throat
 di = diffuser
 td = throat and diffuser

Subscript Examples

Q_2 = secondary liquid vol. flow rate
 Q_{G0} = secondary gas flow rate, at 0
 Q_{2G0} = flow rate of bubbly mixture of gas in the secondary liquid, at 0

NOTE: The convention for pressure and stress in this section is to use psi (and psia) for pounds per square inch (and pounds per square inch absolute) instead of the conventional lb/in² used throughout the rest of the handbook.

LIQUID-JET PUMP THEORY FOR THREE SECONDARY-FLOW TYPES

The liquid-jet pump model is based on conservation equations for energy, momentum, and mass. Real-fluid losses are accounted for by friction-loss coefficients (K). The primary or motive fluid is a liquid of density ρ_1 . In the following derivation, the secondary/pumped fluid

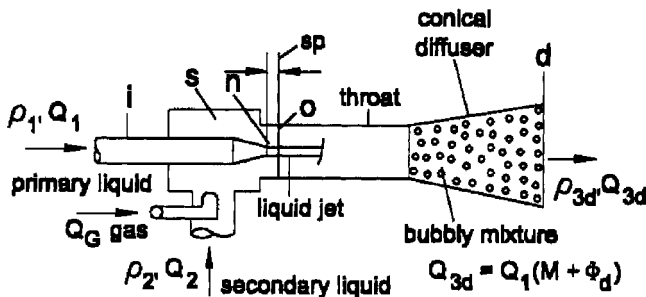


FIGURE 1 Liquid-jet gas and liquid pump and nomenclature

can be a second liquid of density ρ_1 or ρ_2 , or a gas-in-liquid bubbly mixture, or a gas. These three jet pump flow regimes are referred to as liquid-jet liquid (LJL), liquid-jet gas liquid (LJGL), and liquid-jet gas (LJG). Equations (1), (3), (5), and (7) below apply to all three.

Assumptions:

- The primary and secondary streams enter the mixing throat with uniform velocity distributions, and the mixed flows leave the throat and diffuser with a uniform velocity profile.
- The gas phase—if present—undergoes isothermal compression in the throat and diffuser.
- All two-phase flows at the throat entry and exit consist of homogeneous bubble mixtures of a gas in a continuous liquid.
- Heat transfer from the gas to the liquid is negligible—the liquid temperature remains constant.
- Change in solubility of the gas in the liquid from pressure P_s to P_a is negligible.
- Vapor evolution from and condensation to the liquid are negligibly small.

NOZZLE EQUATION With reference to Figure 1

$$P_i + \rho_1 \frac{V_i^2}{2} = P_o + \rho_1 \frac{V_n^2}{2} + K_n \rho_1 \frac{V_n^2}{2} \quad (1)$$

For

$$P_i = \bar{P}_i$$

the nozzle equation is

$$P_i - P_o = Z(1 + K_n)$$

THROAT-ENTRY EQUATION The two-phase secondary flow is described by

$$\frac{dP}{\rho} = VdV + d\left(K_{en} \frac{V^2}{2}\right) = 0 \quad (2)$$

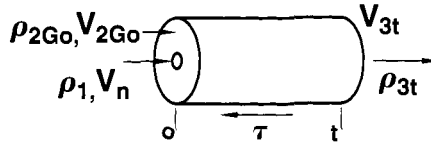
Density of the secondary fluid as a function of static pressure and flow ratios M and ϕ is

$$\rho_{2G} = \frac{m_2 + m_G}{Q_2 + Q_G} = \frac{m_1 \left[\frac{m_2}{m_1} + \frac{m_G}{m_1} \right]}{Q_1(M + \phi)} = \rho_1 \left[\frac{SM + \gamma\phi_s}{m + \phi} \right]$$

Integration of Eq. (2) using this density relation and continuity results in the throat-entry equation:

$$M(P_s - P_o) + P_s \phi_s \ln \frac{P_s}{P_o} = Z \frac{SM + \gamma\phi_s}{c^2} (1 + K_{en})(M + \phi_o)^2 \quad (3)$$

MIXING THROAT MOMENTUM EQUATION Equating control volume forces and fluid momentum changes:



$$(P_0 - P_t)A_{th} - \tau A_w = (m_1 + m_2 + m_G) V_{3t} m_1 V_n - (m_2 + m_G) V_{2Go} \quad (4)$$

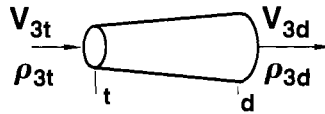
where

$$\frac{\tau A_w}{A_{th}} = \frac{K_{th} \rho_{3t} V_{3t}^2}{2}$$

Substituting the density and continuity relations in Eq. (4) and dividing A_{th} produces a quadratic equation in P_t . The throat momentum equation becomes

$$\begin{aligned} P_t^2 - Z \left[2b - b^2(2 + K_{th})(1 + SM + \gamma\phi_s)(1 + M) \right. \\ \left. + 2(SM + \gamma\phi_s)(M + \phi_0) \frac{b^2}{(1 - b)} + \frac{P_0}{Z} \right] P_t \\ \left. + Z[b^2(2 + K_{th})(1 + SM + \gamma\phi_s)P_s\phi_s] = 0 \right. \end{aligned} \quad (5)$$

DIFFUSER EQUATION The mixed primary and secondary fluids flow from t to d :



The t - d flow is described by Eq. (6):

$$\int_t^d \frac{dP}{\rho} + \int_t^d V dV + \int_t^d \frac{d\Delta P_f}{\rho_{3t}} = 0 \quad (6)$$

Integrating Eq. (6) and substituting the density and continuity relations, the diffuser equation becomes

$$\begin{aligned} (P_d - P_t) + \frac{P_s \phi_s}{1 + M} \ln \frac{P_d}{P_t} = Zb^2 \left[\frac{1 + SM + \gamma\phi_s}{1 + M} \right] \\ \times [(1 + M + \phi_t)^2 - a^2(1 + M + \phi_d)^2 - K_{di}(1 + M + \phi_t)(1 + M)] \end{aligned} \quad (7)$$

LIQUID-JET PUMP EQUATIONS The liquid jet pump is described in terms of the four flow processes by Equations (1), (3), (5), and (7). The term $\gamma\phi_s = \rho_{Gs} Q_{Gs}/\rho_1 Q_1$. For isothermal flow, $\gamma\phi_s = 144P_s\phi_s/RT_s\rho_1$. For air, with $R = 1716$ ft lb/slug $^{\circ}R$, and water, $\rho_1 = 1.94$ slugs/ft³ and $\gamma\phi_s = .0432P_s\phi_s/T_s$. In SI, $R = 286.92$ mN/kg $^{\circ}K$, $\rho_1 = 1000$ kg/m³, and $\gamma\phi_s = .00348P_s\phi_s/T_s$, with P_s in kN/m², and T_s in $^{\circ}K$. The R and/or ρ_1 values must of course be replaced for fluids other than water and air.

In the LJGL pump, the secondary flow is a bubbly mixture of a gas in a liquid. Compressible flow phenomena must be considered because the velocity of sound is quite low in bubbly fluids. For example, the velocity of sound in a 50/50 uniform mixture of air in water is about 70 ft/s (21.3 m/s), far below sonic velocities in air (1100 ft/s or 335 m/s) or water (5000 ft/s or 1524 m/s).

The Mach number of a bubbly secondary flow at the throat entrance is (see Reference 5):

$$\text{MN}_{2G0} = \frac{V_{2G0}}{C_{2G0}} = \frac{\phi_0}{c} \sqrt{\frac{2Z}{P_s \phi_s} (SM + \gamma \phi_s)} \quad (8)$$

When this Mach number is 1.0, the LJGL pump has reached limiting flow; that is, a reduction in back pressure P_d no longer causes an increase in the bubbly secondary-flow rate.

Pump Efficiencies The LJGL pump produces two useful work results:

- Static-pressure increase of the liquid component of the secondary flow stream.
- If a gas is entrained in this liquid stream, isothermal compression of the gas component.

With W as the work rate, ft-lb/s, (power) $W_L = Q_2 (P_d - P_s)$ is the work rate on the liquid component, and $W_G = \rho_{Gs} Q_{Gs} R T \ln(P_d - P_s)$ is the work rate on the gas component. The energy rate input is $E_{in} = Q_1 (P_i - P_o)$. The LJGL pump mechanical efficiency is the total work rate divided by the energy rate in

$$\eta = \frac{M(P_d - P_s) + P_s \phi_s \ln \frac{P_d}{P_s}}{P_i - P_d} = \eta_L + \eta_G \quad (9)$$

The Jet Loss Jet pumps in practical applications have nozzle-to-throat spacings sp/Dth of one or more mixing-throat diameters. The power jet traverses from a static pressure at or near P_s down to P_o , with no useful work recognized in the one-dimensional theory. Thus a “jet loss” occurs, which is in addition to the frictional and mixing losses (see Reference 9).

In the LJL and LJGL (but not the LJG) pumps, throat-inlet pressure drops—and hence jet losses—are significant (Reference 6).

Pump Efficiency, Incorporating Jet Loss In Eq. (9), $(P_i - P_d)$ is expanded: $(P_i - P_d) = (P_i - P_s) - (P_d - P_s)$, and $(P_i - P_s) = (P_s - P_o) = Z(1 + K_n) - j(P_s - P_o)$, where $j = 1$ for a fully inserted nozzle, *no jet loss*; and $j = 0$ for the usual case of retracted nozzle, which produces *full jet loss*. Eq. 9 now becomes

$$\eta = \frac{M(P_d - P_s) + P_s \phi_s \ln \frac{P_d}{P_s}}{Z(1 + K_n) - j(P_s - P_o) - (P_d - P_s)} = \eta_L + \eta_G \quad (10)$$

Eq. (10) is recommended for predicting liquid-jet pump efficiencies as follows: Use $j = 0$ for pumps with normally-retracted nozzles (full jet loss); use $j = 1$ for no-jet-loss pumps (thin-walled nozzle tip fully inserted so $sp = 0$). The pressure in Eq. (10) should be calculated from the one-dimensional theory using Eqs (1), (3), (5), and (7). (See below for the LJL jet pump.)

Computer Programs for LJGL and LJG Models Solutions for the compressible flow cases are generated using computer spreadsheet or Fortran programs. Values for Z , b , P_s , T_s , R , ρ_1 , S and the four K coefficients are fixed/assumed for each pump and operating conditions. Eqs. (3), (5), and (7) are then solved for each step increase in flow-ratio M , with ϕ_s held constant. Alternatively, M may be held constant and the equations solved for step increase in ϕ_s . Eqs. (3), (5), and (7) are interdependent: solution of Eq. (5) requires P_o values from Eq. (3) and solution of Eq. (7) requires P_i values from Eq. (5). The program outputs at each flow-ratio step are static pressure P_o , P_s , P_d , and the three pump efficiencies defined by Eq. 10.

LJGL FLOW CUT-OFF Compressible-flow choking of the secondary stream at the throat entrance will occur at $\text{MN}_o = 1$. The flow ratio at which this will occur can be predicted from critical-flow theory. For further details, see Reference 5.

Performance of the gas compressor (LJG) can be calculated from Eqs. (1), (3), (5), and (7) by setting $M = 0$. Although simplified, the equations are still coupled as in the LJGL case. The one-dimensional theory predicts actual performance quite well (see References 6 and 7) provided the mixing is completed within the length of the mixing throat. Theory-experiment agreement fails and the gas compressor efficiency declines—mixing is allowed to extend into the diffuser.

THE LIQUID-JET LIQUID (LJL) PUMP

Equations for the LJL pump are much simpler than the corresponding LJGL and LJG equations because of the elimination of all ϕ terms. In this case, Eqs. (1), (3), (5), and (7) reduce to the following set:

$$\text{Nozzle} \quad P_i - P_o = Z(1 + K_n) \quad (1)$$

$$\text{Throat Entry} \quad P_s - P_o = ZS(1 + K_{en})M^2/c^2 \quad (11)$$

$$\text{Throat} \quad P_t - P_o = Z[2b + 2SM^2b^2/(1 - b) - b^2(2 + K_{th})(1 + SM)(1 + M)] \quad (12)$$

$$\text{Diffuser} \quad P_d - P_t = Z_b^2(1 + SM)(1 + M)(1 - K_{di} - a^2) \quad (13)$$

Pump efficiency η is defined as the ratio of useful work rate on the secondary fluid Q_2 to the energy extracted from the primary liquid:

$$\eta = Q_2(P_d - P_s)/Q_1(P_i - P_d) = MN \quad (14)$$

Two other definitions of efficiency are found in the literature, as follows:

$$\eta' = Q_2(P_d - P_s)/Q_1(P_i - P_s) = \eta/N + 1$$

$$\eta'' = (M + 1)(P_d - P_s)/(P_i - P_s)$$

These two other definitions assume that the primary/power stream pressure falls to P_s , not P_d , as shown in Figure 2. Efficiency conversions are possible only if all three pressures and two flow rates involved are given. Comparisons of efficiencies reported in the literature should be made with caution.

Combining Eqs. 1, 11–14, the theoretical pressure characteristic N for the LJL pump is

$$N = \frac{2b + \frac{2SM^2b^2}{1-b} - b^2(1 + K_{td} + a^2)(1 + M)(1 + SM) - \left(\frac{SM^2}{c^2}\right)(1 + K_{en})}{1 + K_n - 2b - \frac{2SM^2b^2}{1-b} + b^2(1 + K_{td} + a^2)(1 + M)(1 + SM) + (1-j)\left(\frac{SM^2}{c^2}\right)(1 + K_{en})} \quad (15)$$

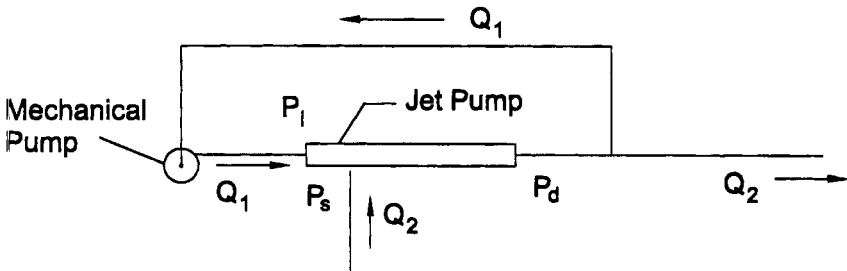


FIGURE 2 LJL jet pump and installation

where $K_{td} = K_{th} + K_{di}$. The term $j = 0$ for the normal (jet loss) case, and $J = 1$ for no jet loss. Jet pumps normally are designed with a finite nozzle-throat spacing, typically $sp/D_{th} = 1$; jet loss is experienced, and thus $j = 0$ for this (usual) case. Eq. (15) becomes

$$N = \frac{2b + \frac{2SM^2b^2}{1-b} - b^2(1 + K_{td} + a^2)(1 + M)(1 + SM) - \left(\frac{SM^2}{c^2}\right)(1 + K_{en})}{1 + K_n - \text{numerator}} \quad (16)$$

In terms of pressure,

$$N = (P_d - P_s)/(P_i - P_d)$$

Two simplifications of Eq. (16) are often appropriate: 1) The area ratio term $a^2 = 0$ for the usual 5°–8° included-angle diffuser ($a = A_i/A_d$ is small). 2) The density ratio $S = 1$ for similar primary and secondary liquids, for example, a water primary jet pumping water as the secondary fluid. With these simplifications, Eq. (16) for the normal (jet loss) case becomes

$$N = \frac{2b + \frac{2SM^2b^2}{1-b} - b^2(1 + K_{td})(1 + M)^2 - \left(\frac{M^2}{c^2}\right)(1 + K_{en})}{1 + K_n - \text{numerator}} \quad (17)$$

And pump efficiency is

$$\eta = \eta_L = MN \quad (18)$$

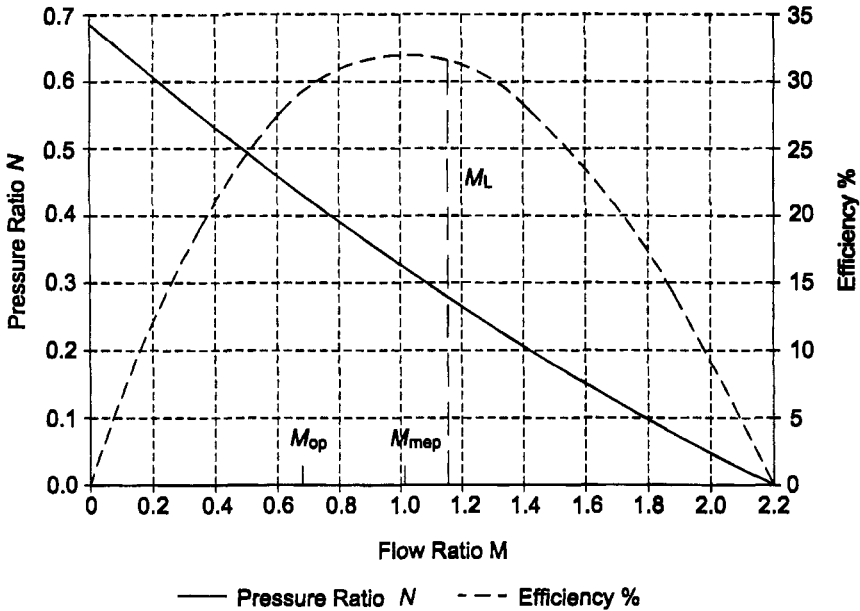
For the L_{JL} jet pump, note also that

$$(P_d - P_s)/(P_i - P_s) = N/(N + 1) \quad (19)$$

Computer Programs for L_{JL} Models It is convenient to use Eqs. (17) and (18) in spreadsheet form to generate tables of “ $N(b, M, K_n, K_{td}, K_{en})$,” that is, the L_{JL} jet pump pressure characteristic N as a function of the bracketed variables. Typically the K s and the area ratio b are held constant and a table is generated using step increases in M to show resultant N values. Table 1 shows $N(M)$ for $b = .25$; and $K_n = .05$, $K_{td} = .2$, and $K_{en} = 0$. The performance of this pump is shown in Figure 3 as $N(M)$ and $\eta(M)$.

TABLE 1 Performance versus M , for $b = .25$, $K_n = .05$, $K_{td} = .2$, $K_{en} = 0$

M	$(P_d - P_s)/Z$	$(P_i - P_d)/Z$	N	$\eta\%$
0.000	0.4250	0.6250	0.6800	0.00
0.200	0.3942	0.6558	0.6012	12.02
0.400	0.3619	0.6881	0.5259	21.04
0.600	0.3280	0.7220	0.4543	27.26
0.800	0.2926	0.7574	0.3862	30.90
1.000	0.2556	0.7944	0.3217	32.17
1.200	0.2170	0.8330	0.2605	31.26
1.400	0.1769	0.8731	0.2026	28.36
1.600	0.1352	0.9148	0.1478	23.65
1.800	0.0920	0.9580	0.0960	17.29
2.000	0.0472	1.0028	0.0471	9.42
2.200	0.0009	1.0491	0.0008	0.19
2.204	0.0000	1.0500	0.0000	0.00
0.676	0.3147	0.7353	0.4280	28.93
1.014	0.2529	0.7971	0.3173	32.17

FIGURE 3 $K_n = .05, K_{i1} = .20, K_{en} = 0$

It has been shown (see References 2, 3, and 4) that one-dimensional analyses successfully predict actual LjL pump performance. But theory-experiment agreement obtains only if the test-pump flow conforms with the model assumptions. Two conditions that will cause departure of measured N and η data from the theoretical curves are

- Cavitation, which occurs in the mixing throat
- Extension of the throat mixing process out into the diffuser

Operating Flow Ratio M_{op} In Figure 3, the flow ratio M_{op} is indicated on the left slope of the “parabolic” efficiency curve, which peaks at $M = M_{mep}$. The recommended location of the design operating point is to set $M_{op} = \frac{2}{3} M_{mep}$. Higher operating M ratios would provide slightly higher efficiencies, but at a greater risk of cavitation. (Cavitation as part of the design process is included in the following examples given.) Finding M_{mep} can be readily accomplished from Table 1 type data using spreadsheet successive approximations. Alternatively, Eqs. (17)–(18) can be differentiated and set equal to zero to find the peak efficiency M_{mep} value.

Cavitation LjL pumps may encounter cavitation, which occurs in the mixing throat. With reference to Figure 3, the LjL pump normally responds to a reduction in back pressure P_d (Q_1 and P_s constant) by producing a larger Q_2 secondary flow, and hence a larger M . Measured pressure ratios (N) and efficiencies (η) track along these theory-based characteristic curves as shown in Figure 3. But after the throat-inlet pressure (P_d) is reduced to the vapor pressure (P_v) of the secondary liquid, any further drop in the back pressure has no effect on the flow ratio, which stabilizes at $M = M_L$, the *cavitation-limited flow ratio*. Note the vertical dashed line in Figure 3: measured N and η values fall on this vertical line, under M_L operating conditions. In this manner, cavitating-pump performance departs radically from predicted/normal behavior.

Published studies (see References 1 and 8) have shown that NPSH-type correlations adequately explain and predict cavitation-limited flow phenomena. **Comparing the predicted M_L with the intended M_{op} is an essential step in designing a jet pump**

installation. If $M_{op} < M_L$, the L_JL pump can be expected to perform “on design,” that is, to follow the Eq. (17) $N(M)$ relation, with no cavitation. M_L can be predicted from the operating conditions as follows (see Reference 8):

$$M_L = c \sqrt{\frac{P_s - P_v}{\sigma Z}} \tag{20}$$

where σ is a cavitation coefficient.

Experiments, primarily with water and lubricating oils, have shown a σ range of 0.8–1.4 (see Reference 1). One investigator (see Reference 10) found that improving the nozzle exterior profile and related throat-inlet internal profile reduced the measured σ from 1.4 to 1.0, indicating a significant improvement in cavitation resistance. For design use, a conservative value of $\sigma = 1.35$ is recommended (see Reference 8).

Area Ratio b The nozzle-to-throat area ratio b is the only geometric parameter in these liquid-jet pump models, and including the L_JL case, Eq. (17). Area ratio b is all-important, affecting pump efficiency, flow capacity, cavitation, and pressure characteristic $N(M)$. Comparisons of b -value series of pumps (see References 1–4) show that peak efficiency is highest for $b = .2$ to $.3$. This optimum occurs because efficiency of the jet pump reflects the two unavoidable losses: friction and the mixing loss. The latter (a maximum at zero M) decreases with rising M , whereas frictional losses increase with M . The sum of these losses is a minimum for pumps in the $b = .2 - .3$ range, each operating at or near the respective M_{mep} (see Reference 9). As indicated in the numerical Examples 1 and 2 further on, it is recommended that jet pump designs start—in the absence of other restraints—with $b = .25$. This initial area ratio can then later be adjusted, for example, to handle a cavitation problem. In specific cases the jet pump placement, or a desired flow ratio, may determine N or M , and hence b . Example 3 illustrates the situation.

Friction-Loss Coefficients Solutions of Eqs. (11)–(18) require use of appropriate K loss coefficients. The following sources are suggested:

- 1. Adopt published results from jet pump studies that have included measurements of K values (see References 3, 10, and 11).
- 2. Use published loss coefficients for the (four) components of the L_JL pump; that is, converging nozzle, tube entry, short tube, and diffuser.
- 3. Measure K s directly by running flow test of jet pumps or component parts. Full evaluation of the four K s requires measurement of two flow rates and five static pressures, P_i , P_s , P_o , P_t , and P_d . Most studies, however, have omitted P_o and P_t pressure taps, and measured only three of these pressures: P_i , P_s , and P_d . This omission limits K evaluations based on pump tests to K_n and K_{td} . Fortunately, the theory works quite well under this limitation, largely because the throat and diffuser K s are additive ($K_{id} = K_{th} + K_{di}$) in the $N(M)$ equations.

In regard to Item 1 above, Table 2 shows the range and recommended values for the K s which appear in the theoretical model equations.

Table 2 recommendations assume that only P_i , P_s , and P_d are utilized; if this is the case, it is necessary to assume that $K_{en} = 0$, as indicated. A finite value of K_{en} in Eqs. (15)–(17)

TABLE 2 Recommended values for K friction-loss coefficients

	Range	Rec. Values	References
K_n	.04–1.0	.05	3, 10, 11
K_{en}	—	0	ditto
K_{td}	.17–.40	.20	ditto

TABLE 3 Values of $M_{op} = \frac{2}{3}M_{mep}$, and N_{op} , for Figure 4

b	M_{mep}	M_{op}	N_{op}
0.05	4.8520	3.2347	0.0739
0.10	2.7120	1.8080	0.1509
0.15	1.8294	1.2196	0.2344
0.20	1.3330	0.8887	0.3264
0.25	1.0142	0.6760	0.4280
0.30	0.7913	0.5275	0.5406
0.40	0.5020	0.3347	0.8039
0.50	0.3270	0.2180	1.1210
0.60	0.2117	0.1411	1.4913

increases the slope of the theoretical $N(M)$ curve at low M values. At operating M values, however, this effect is negligible.

Finding b as a Function of N_{op} or M_{op} Figure 4 presents $N_{op}(b)$ and $M_{op}(b)$ for the recommended K friction coefficients of .05, 0, and .20. The Figure 4 curves are cross-plots of M_{op} values ($=\frac{2}{3}M_{mep}$) and associated N_{op} values. (To find M_{mep} value at b , Eqs. (17)–(18) spreadsheets of $\eta(M)$ and $N(M)$ were prepared for b values covering the range .05–.6; these numerical values are provided in Table 3.) Graphical representation of the LjL pump characteristics as shown in Figure 4 is useful in estimating b values when the jet pump application fixes N or M , and hence b . Example 3 illustrates the utility of Figure 4.

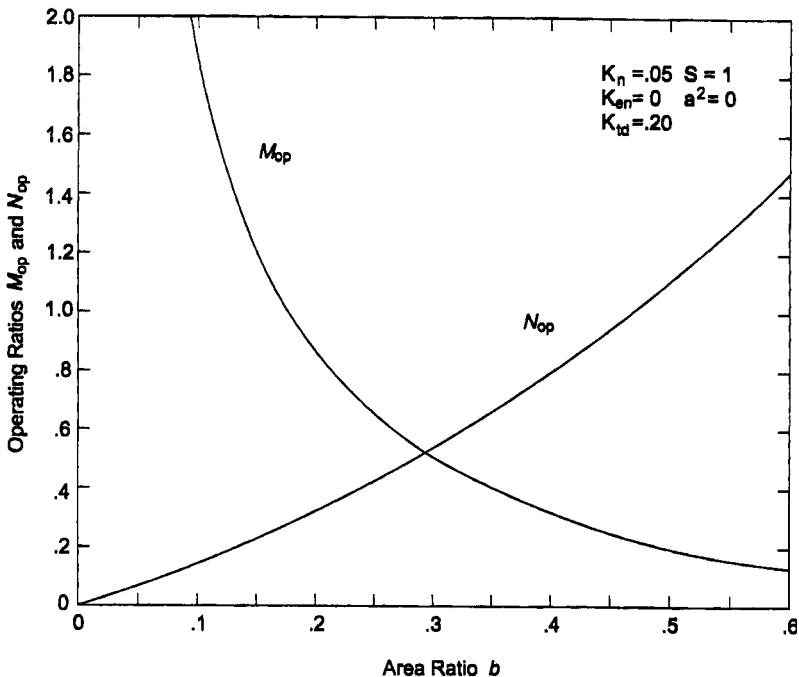


FIGURE 4 Response of LjL operating pressure ratio N_{op} and operating flow ratio M_{op} to jet pump design area ratio b

Straight-Line $N(M)$ Approximation Early investigators (see Reference 2) noted a nearly linear behavior of N versus M . A “straight-line” approximation for $N(M)$ based on the axis intercepts N_o and M_o would, of course, be useful. A true parabolic efficiency curve would result so $M_{\text{sep}} = \frac{1}{2} M_o$, and M_o is easily found. Unfortunately, the degree of this linearity varies with area ratio b . At about $b = .12$, the N curve is in fact a straight line. But as b is increased, the $N(M)$ curve is increasingly concave down so the $\eta = MN$ “parabolic” curve is increasingly skewed to the left. The converse is true at very small b values. Accordingly, the straight-line approximation is not recommended for use in designing LJL jet pumps. Instead, the Eqs. (17) and (18) theoretical model should be solved and charted for each b -value pump under consideration. Again, the recommended operating flow ratio $M_{\text{op}} = \frac{2}{3} M_{\text{sep}}$.

Longitudinal Dimensions of the LJL Jet Pump One-dimensional theory includes dimensions perpendicular to flow, but provides no guidance for longitudinal shapes/profiles. Two hardware dimensions of key importance must be taken from the literature or determined experimentally: nozzle-to-throat spacing “sp” and mixing throat length “ L .” Both are expressed in terms of throat diameters; sp/D_{th} and L/D_{th} . Recommended values are given as follows.

NOZZLE-THROAT SPACING Many experimental searches for optimum sp have been reported. Sanger and Vogel both found (see References 10 and 12) that maximum efficiency (slightly above 40% for Reference 10) is obtained with $\text{sp}/D_{\text{th}} = 0$, using nozzles with an external concave tapering, leading to a thin lip at the outlet. This configuration matches the theoretical model: The jet discharges to pressure P_o at the throat inlet, and thus the normal jet loss was eliminated. But a zero spacing promotes cavitation, even with a thin-lipped nozzle tip, and certainly with the rounded-nose exterior profile used in Reference 4. Sanger found that retracting nozzles a distance of about one diameter provided good cavitation resistance and at only a small loss in efficiency. Other investigators have found that performance is insensitive in the range of $\text{sp}/D_{\text{th}} = 0.5$ –2 diameters, and that small spacings do promote cavitation. (see Reference 1). *It is recommended that L/JL jet pumps be designed with $\text{sp}/D_{\text{th}} = 1$.*

MIXING-THROAT LENGTH The parallel-walled throat should be long enough to allow complete mixing, but throat lengths should be as short as possible to minimize frictional losses. L/D_{th} values as well as length (a venturi shape) ranging 1.0 to 10 have been reported for L/JL pumps. Several factors affect optimum throat length:

- When sp/D_{th} is finite—true for most jet pumps—mixing takes place in the distance (sp plus part or all of L , the mixing-throat length); thus, sp and L are interrelated. Sanger (see Reference 10) found that optimum sp/D_{th} increased from 0 to 2.3, for pumps with $L/D_{\text{th}} = 7.5$ and 3.5 respectively.
- The primary-flow nozzle affects required L : Long tapered nozzles promote boundary-layer build-up producing jets that delay mixing, increasing required L . Multi-hole nozzles and swirl-inducing nozzles promote mixing and reduce required length L , but pump efficiency suffers because of increased nozzle-flow losses. (See “Primary-Flow Nozzle Design.”)
- Pump area-ratio b can affect optimum throat length. Small- b pumps operate with high flow ratios and throat lengths of $L/D_{\text{th}} = 8$ were required. For pumps with larger b values, throat lengths of four diameters sufficed (see Reference 13).
- The gas compressor jet pump (LJG) requires longer throats, as high as 10–30 throat diameters (see Reference 7).

$L/D_{\text{th}} = 6$ is recommended for general L/JL design use. Efficiency of the proposed pump may subsequently be improved by optimizing L (experimentally) for the given pump and duty.

PRIMARY-FLOW NOZZLE DESIGN A short-entry internally-convex profile similar to the ASME metering flow nozzle, is recommended for the L/JL pumps. Avoid long conical nozzles.

Liquid-jet flow from a sharp-edged orifice mixes readily and is recommended for the LJG gas compressor (see Reference 7). The annular-nozzle liquid-jet pump has been investigated. In this configuration the secondary flow is axial, surrounded by the primary flow at the throat entrance. This arrangement is advantageous in pumping sticky secondary fluids because it prevents wall contact of the sticky fluid at the throat entry (see Reference 1). An obvious disadvantage is the increased nozzle frictional loss caused by flow of the primary fluid over the comparatively large surface area of the annular nozzle.

THROAT-INLET CONTOUR Many jet pumps reported in the early literature had long (small-convergence-angle) conical sections connecting the suction chamber to the throat, and usually including a sudden wall angle change at the throat entry. Later developments led to the short (large-convergence-angle) entry, well rounded at the throat. The long (small-angle) conical entry is wrong because its proximity to the nozzle exterior throttles the secondary flow and because it promotes cavitation. Secondly, a long approach section increases wall friction (reflected in a high K_{en} if it is measured). A short entry to the throat and a well-rounded profile connecting the suction chamber and throat is recommended.

Laboratory Flow Tests Performance testing of LJL pumps requires a facility with appropriate instrumentation, pumps, flow meters, and control valves. The flow rates Q_1 and Q_2 , and at least three static pressures, P_i , P_s , and P_d , must be recorded at each test point. In addition, it is recommended that the throat section(s) contain static pressure taps for measurement of P_o and P_t . Jet pump test data will then permit measurement of K_{en} , K_{th} , and K_{di} . For on-design operation, these K s vary little, if at all, with change in flow ratio M . Departures of $K(M)$ from nominal levels serve to reveal otherwise hidden problems. One example is that K_{th} will change if an increase in flow ratio causes mixing to extend/persist from the throat section into the diffuser (inadequate throat length). Another example is that a sudden rise in K_{di} may indicate diffuser-wall separation.

LIQUID-JET LIQUID (LJL) PUMP DESIGN EXAMPLES

Equations (1), (3), (5), and (7) will model jet pumps for LJL, LJG, and LJGL configurations. Equations (1), (11–19) apply specifically to the LJL jet pump. The three numerical examples provided here are for this widely used pump only. LJG and LJGL liquid-jet pumps are less common: please see References 5, 6, and 7.

Example 1 Design a jet pump to handle 50 gpm (11.36 m³/h) of water at 80°F (26.7°C) from a suction at 14.7 psia atmospheric pressure (101.325 kPa) to discharge at 40 psi (275.8 kPa). Determine the required primary flow rate, jet nozzle pressure, and dimensions of the jet pump.

Solution: For best efficiency, select $b = .25$. From Table 2, adopt $K_n = 0.5$, $K_{td} = .20$, and $K_{in} = 0$. Eq. (17) then produces $N(M)$. A computer spreadsheet table for Eqs. (17) and (18) showing output values based on increments of M is recommended, for example, see Table 1. As shown by the bottom line in Table 1, the $M_{nep} = 1.104$ (found by successive approximations using the spreadsheet program). The operating flow ratio is $M_{op} = \frac{2}{3}M_{nep} = .676$. The spreadsheet program at this M_{op} produces the N_{op} (.428) and η (28.934%) values shown in Table 1, second line from the bottom. From Eq. 19 ($P_d - P_s$)/($P_i - P_s$) = $N/(N + 1)$, so and $P_i - P_s = (1.428/.4828)40 = 1.3346$ psi (920.212 kPa). Thus $P_i = 1.3346 + 14.7 = 148.16$ psia (1021.56 kPa). Liquid jet pumps—with very few exceptions—operate with the nozzle tip withdrawn from the throat entry by one throat-diameter or more. The nozzle tip experiences a discharge pressure close to P_s , not P_o , and “jet loss” thus occurs. For this jet loss condition, Eq. (1) changes to ($P_i - P_s$) = $Z(1 + K_n)$. The jet velocity head $Z = 1.3346/1.05 = 1.2710$ psi (876.35 kPa).

CAVITATION-LIMITED FLOW RATIO M_L The flow ratio M_L is now evaluated using Eq. (20) to answer the question: will $M_{op} = .676$ avoid pump cavitation? The area ratio $c = (1 - .25)/$

.25 = 3. For the secondary fluid (water at 80°F, 26.7°C), $P_v = .506$ psia (3.49 kPa). The conservative value of $\sigma = 1.35$ is used in Eq. (20) as follows:

$$M_L = 3\sqrt{\frac{14.7 - .506}{1.35 \times 1.2710}} = .863$$

Because $M_{op} (.676) < M_L (.863)$, this pump will not cavitate at the specified operating condition.

JET PUMP DIMENSIONS $Q_1 = Q_2/M_{op} = 50/.676 = 73.96$ gpm (16.8 m³/h) = $73.96/(7.481 \times 60) = .165$ ft³/s. Longitudinal dimensions: $sp/D_{th} = 1.0$, thus $sp = .937$ in (23.81 mm). For $L/D_{th} = 6$, $L = 6 \times .937 = 5.62$ in (142.75 mm). A diffuser with an included angle of 5° (conservative) and area ratio $a = .224$ would have length of approximately 12 in (304.8 mm). A shorter diffuser may be desirable, but kinetic-energy leaving losses will be higher.

EXAMPLE 1 USING OTHER AREA RATIOS An infinite number of different jet pumps can be designed to handle the Example 1 duty of pumping 50 gpm at 40 psi discharge pressure. The previous numerical example ($b = .25$) was repeated using $b = .1, .4$, and $.6$. Table 4 compares the results of these four b -ratio pumps. In each case the design is based on $M_{op} = \frac{2}{3}M_{mep}$; the assumed K values and calculation procedure are similar for the four pumps.

The expression CR% in Table 4 indicates the $(M_L - M_{op})/M_{op}$ % separation of operating flow ratio M_{op} and the limiting flow ratio M_L ; the larger this number, the better. For this constant-duty example ($Q_2 = 50$ gpm and $P_d = 40$ psi), Table 4 shows that cavitation resistance can be improved by using a larger b ratio, i.e., the $(M_L - M_{op})/M_{op}$ % figure increases with b . Note that at the smallest b value (.1), the cavitation value (sixth column) is CR = -5.3%; this indicates that $M_{op} > M_L$, and this pump *would* encounter cavitation: $b = .1$ should not be used at these M_{op} and σ values. A redesign of the pump suction chamber (nozzle-external profile and throat-inlet profile) leading to an improvement in cavitation coefficient σ could render the $b = .1$ pump usable (the value $\sigma = 1.35$ used previously is conservative). For example, an improvement to $\sigma = 1.0$ would raise M_L to 2.0. With this change, a $b = .1$ pump could be used, cavitation-free.

Table 4 shows two other important facts: (1) The $b = .25$ pump provides the highest efficiency, even though all four are designed to operate at $\frac{2}{3}M_{mep}$, i.e., each at a similar position on the efficiency curve specific to that pump. (2) Small b -value pumps operate at large nozzle pressure-drops ($(P_i - P_s)$) and with small Q_1 primary-values. Conversely, high b -value pumps operate with small nozzle pressure-drops, but use a large flow rate Q_1 . The pumps have in common similar expenditures of energy ($Q_1 \times$ pressure change, and allowing for efficiency differences) to handle the duty, which is the same in all four cases. In some design problems the nozzle-pressure—or possibly the primary-fluid flow volume—might outweigh the importance of mechanical efficiency. Table 4 shows how an adjustment in b might be used to achieve improved cavitation, albeit with efficiency sacrifices.

REDUCING THE FLOW RATIO M_{op} TO COPE WITH CAVITATION For a given b -value LJL jet pump, a reduction in M_{op} offers a way to improve cavitation resistance. The $b = .1$ pump used

TABLE 4 Four b -value jet pumps for duty of 50 gpm (11.35 m³/h) at $P_d = 40$ lb/in² (275 kPa)

b	M_{op}	N_{op}	$\eta\%$	M_L	CR%	$(P_i - P_s),$ lb/in ²	Q_1 , gpm	D_n , in
0.10	1.8080	0.1509	27.28	1.7122	-5.30	305.08	27.65	0.233
0.25	0.6760	0.4280	28.93	0.8628	27.65	133.46	73.96	0.469
0.40	0.3347	0.8039	26.91	0.5261	57.19	89.76	149.39	0.736
0.60	0.1411	1.4913	21.04	0.2711	92.13	66.82	354.36	1.219

[Note: for SI conversion: in $\times 25.4 =$ mm]

TABLE 5 Reducing M_{op} improves CR% cavitation resistance ($b = .1$)

M_{op}	N_{op}	$\eta\%$	M_L	CR%	$(P_i - P_s)$, lb/in ²	Q_1 , gpm	D_n , in
1.808	0.1509	27.28	1.7122	-5.30	305.08	27.65	0.233
1.500	0.1631	24.46	1.7704	18.01	285.08	33.33	0.260
1.000	0.1823	18.23	1.8566	85.67	259.45	50.00	0.326
0.500	0.2006	10.03	1.9326	286.56	239.36	100.00	0.471

[For SI conversions: lb/in² $\times$ 6.89 = kPa; gpm $\times$ 0.277 = m³/h; in $\times$ 25.4 = mm]

in this example is examined for the effects of M_{op} reduction, in Table 5. The conditions for all four pumps in Table 5 are $b = .1$, and again, $K_n = .05$, $K_{td} = .2$, $K_{en} = 0$. The Table 5 top line is $M_{op} = 1.808$, followed by three .5 decrements of M_{op} . The corresponding N_{op} and η_{op} values are found from successive approximations using the spreadsheet based on Eqs. (17) and (18). The M_L values are calculated from Eq. (20). The CR% cavitation merit figure (5th column) is the same as used in Table 4. Table 5 shows that while a $b = .1$ pump at $M_{op} = 1.808$ flow ratio will cavitate, the three lower M_{op} cases all exhibit increased CR% cavitation resistance. In Table 5, M_L (4th column) changes little with M_{op} ; the cavitation improvements largely result from the decreases in M_{op} . The primary flow rate Q_1 increases markedly, and the nozzle pressure decreases somewhat as M_{op} is reduced. Compared with adjustment in b , cavitation avoidance by reducing M_{op} has obvious disadvantages: Efficiency is severely reduced, and the pump size (see D_n , 8th column) must be increased to handle the larger flow Q_1 .

In summary, every jet pump design should be evaluated for cavitation resistance. The CR% index used here ($(M_L - M_{op})/M_{op}$ %) is suggested if several options are to be compared. Otherwise, the simpler test $M_{op} < M_L$ is adequate. Cavitation resistance for a given pump duty can be improved by increasing the area ratio b , or by reducing M_{op} of a particular pump, or by reducing the cavitation coefficient σ by improved internal design.

Example 2 Design a jet pump to lift 80°F (27.67°C) water to 20 ft (6.1m) to discharge at an atmospheric pressure of 14.7 psia or 34 ft of water (10.3 m). A mechanical pump provides the primary flow stream; fluid power available at the jet pump is 10 kW. Find the secondary flow Q_2 , and pump dimensions.

Solution: Assume the same K values used in Example 1. Again select $b = .25$, for maximum efficiency. Eq. 17 (via spreadsheet program) produces $M_{mep} = 1.014$ and $M_{op} = \frac{2}{3}(1.014) = 6.76$, where $N_{op} = .428$ and $\eta = 28.93\%$. $P_s = (34 - 20) = 14$ ft water abs. (4.261 m abs.) at the suction chamber. And $P_s = 14 \times 1.934 \times 32.174/144 = 6.05$ psia (41.72 kPa). $(P_d - P_s) = (14.7 - 6.05) = 8.65$ psi (59.64 kPa). From Eq. 19, $(P_i - P_s) = (P_d - P_s)(N + 1)/N$, so that $(P_i - P_s) = (14.7 - 6.05)(1.428/.428) = 28.86$ psi (199 kPa). $P_i = 28.86 + 6.05 = 34.91$ psia (40.7 kPa). $Z = (P_i - P_s)/(1 + K_n) = 28.86/1.05 = 27.486$ psi (189.52 kPa).

Cavitation-Limited Flow Ratio M_L Limiting flow ratio (Eq. 20)

$$M_L = 3\sqrt{\frac{6.05 - .506}{1.35 \times 27.486}} = 1.16$$

Because $M_{op} (.676) < M_L (1.16)$, the proposed L/JL jet pump will not cavitate.

JET PUMP DIMENSIONS Power in = $Q_1(P_i - P_d)$; $Q_1 =$ power in/ $(P_i - P_d)$, where $(P_i - P_d) = (34.91 - 14.7) = 20.21$ psi. With 10 kW treated here as the net “fluid” power available at the nozzle. (In Figure 2 the jet and mechanical pumps are at the same elevation)

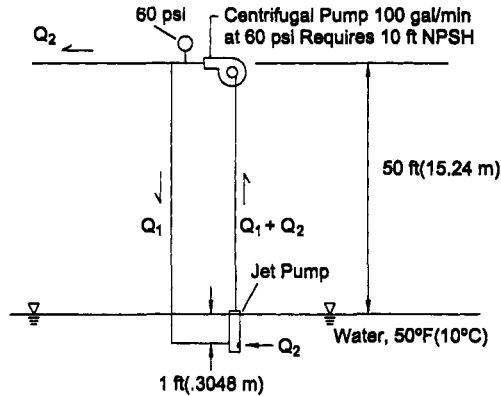


FIGURE 5 Jet and centrifugal pump system, for Example 3

$$Q_1 = \frac{10 \times 7.481 \times 60}{1.356 \times 10^{-3} \times 144 \times 20.21} = 1137.42 \text{ gal/min (258.6 m}^3\text{/h)}$$

(conversion factors 7.481 gal/ft³ and 1.356 watts/ft-lb/s were used previously)

Secondary flow $Q_2 = 1137.42 \times .676 = 768.9 \text{ gpm (174.6 m}^3\text{/h)}$.

Nozzle area:

$$A_n = \frac{Q_1}{V_n}, \text{ where } V_n = \sqrt{2 \times 27.476 \times \frac{144}{1.934}} = 64 \text{ ft/s (19.51 m/s)}$$

$$A_n = (768.9 \times 144) / (64 \times 7.481 \times 60) = 3.85 \text{ in}^2 (24.84 \text{ cm}^2)$$

Nozzle diameter:

$$D_n = \sqrt{3.85 \times \frac{4}{3.1416}} = 2.22 \text{ in (56.39 mm)}$$

Example 3 As shown in Figure 5, a surface-mounted centrifugal pump requires a NPSH of 10 ft (3 m) of water, and the centrifugal-pump discharge pressure is 60 psi (414 kPa) at 100 gpm (22.7 m³/h). Water at 50°F (10°C) is to be lifted 50 ft (15.2 m). Design the jet pump including dimensions, and find the primary and secondary flow rates.

Solution: The jet pump's placement and centrifugal-pump discharge and inlet pressures determine all three jet-pump operating pressures, and hence the pressure characteristic N .

$$P_i = 60 \text{ psi} + (50 + 1) \times 1.94 \times 32.17 / 144 = 82.1 \text{ psi (566.1 kPa)}$$

$$P_s = 1 \times 1.94 \times 32.17 / 144 = .433 \text{ psi (299 kPa)}. P_d = (50 + 10) \times$$

$$1.94 \times 32.17 / 144 = 26 \text{ psi (179.27 kPa)}. N_{op} = (P_d - P_s) / (P_i - P_d) = .456.$$

To estimate the b level which will operate at this N , enter Figure 4 with $N_{op} = .46$; the corresponding $b = .27$, and $M_{op} = .62$. Estimate b values should be confirmed—and adjusted if necessary—using a spreadsheet solution based on Eqs (17) and (18). In this case the

spreadsheet results are $N_{op} = .461$, $b = .265$, $M_{op} = .626$, and $\eta = 28.8\%$. The cavitation check and calculation of dimensions follow next. *Note:* This solution is a first estimate because piping frictional losses have not been included in the calculations. Second and third—if necessary—calculations must include the pressure losses in the piping connecting the two pumps.

CAVITATION LIMITED FLOW RATIO M_L Jet velocity head $Z = (P_i - P_s)/(1 + K_n) = (82.1 - .433)/(1 + .05) = 77.78$ psi (536.29 kPa).

For water at 50°F (10°C), $P_v = .178$ psia (1.23 kPa)

$$M_L = c \sqrt{\frac{P_s - P_v}{\sigma}} \times Z = 2.774 \sqrt{\frac{15.133 - .178}{1.35}} \times 77.78 = 1.047$$

Because $M_{op} (.626) < M_L (1.047)$, the jet pump will not cavitate at the operating point.

JET PUMP DIMENSIONS $Q_1 + Q_2 = 100$ gpm (19.71 m³/h), and $Q_2/Q_1 = .626$, so $Q_1 = 61.5$ and $Q_2 = 38.5$ gpm (12.12 and 7.59 m³/h). $A_n = Q_1/V_n$.

Jet velocity:

$$V_n = \sqrt{\frac{2Z}{\rho_1}} = \sqrt{\frac{2 \times 77.8}{1.94}} = 107.46 \text{ ft/s (32.75 m/s).}$$

$$Q_1 = 61.5/7.481 \times 60 = .137 \text{ ft}^3/\text{s (13.97 m}^3/\text{h)}$$

$$\text{Nozzle area } A_n = (.137/107.46)144 = 1.87 \text{ in}^2 (1.187 \text{ cm}^2).$$

Nozzle diameter:

$$D_n = \sqrt{\frac{4 \times A_n}{\pi}} = \sqrt{\frac{4(.184)}{3.1416}} = .483 \text{ in (12.27 mm).}$$

Throat diameter:

$$D_{th} = \frac{D_n}{\sqrt{b}} = \frac{.483}{.515} = .938 \text{ in (23.83 mm)}$$

Nozzle-throat spacing recommended: $sp/D_{th} = 1$, hence $sp = .938$ in (23.83 mm)

Mixing-throat length recommended: $L/D_{th} = 6$, hence $L = 6(.938) = 5.63$ in (143 mm)

Centrifugal-Jet-Pump Interactions Oil- and water-well pumping are major applications of L_{JL} pumps. Compact jet pumps, lowered down the bore hole near to or below the liquid surface, are powered by primary flow supplied from a mechanical pump located above, at the earth's surface. The jet pump pressurizes the surface pump's suction preventing cavitation. The design of a combination pump system normally requires designing for operation at the intersection of the two pumps' operating curves (References 17, 18, and 19) and of course inclusion of frictional losses (neglected in Example 3) in the associated piping. One operating point (100 gpm at 60 psi) in Example 3 represented the centrifugal pump. The 60 psi discharge pressure of the centrifugal pump and the vertical placement fixed the jet pump's N_{op} operating point. This in turn determined the jet pump's b value. Consider now a change in the centrifugal pump to one providing 100 gpm at 100 psi (22.7 m³/h at 6.89 kPa): the jet pump's N_{op} (and b) would be reduced, and M_{op} raised. Details are omitted here, but the resultant area ratio would be about $b = .167$, which would provide $M_{op} = 1.09$, a considerable increase over $M_{op} = .626$, as found in Example 3.

SIGNIFICANCE OF ON-DESIGN OPERATING CONDITIONS

With reference to Figure 3, assume that the pump is operating on-design at $M_{op} = \frac{2}{3}M_{mep}$; that is, on the left branch of the efficiency "parabola." Primary flow rate Q_1 and suction pressure P_s are constant. If the back pressure P_d is then *reduced*, the secondary flow rate Q_2 (and hence M_{op}) will rise and pressure ratio N_{op} will fall. The new operating points will move to the right, along the characteristic curves for N and η . This increased-flow response to lowered P_d will end when cavitation-limited flow ratio M_L is reached, assuming the NPSH is low enough to allow cavitation. Any further reduction of P_d will cause the operating points to leave the N and η theoretical curves on paths vertically downward as shown by the dashed line in Figure 3. Consider next the same initial condition (on design: $M_{op} = \frac{2}{3}M_{mep}$) followed by an *increase* in jet-pump discharge pressure P_d . In response, the secondary flow Q_2 (and hence M) will decrease, the N_{op} and η_{op} operating points moving left along the pump's N and η characteristic curves. Ultimately, increasing the back-pressure will result in reaching the $M = 0$ point, where $N = N_o$. (See Table 1, showing $N = N_o = .680$ at $M = 0$.)

Mechanical efficiency is important in most jet pump installations. Two low-efficiency situations that should obviously be avoided are

- Throttling the pump discharge pressure (to raise the P_d to the design point) wastes energy. Design P_d should match the pressure of the system into which the pump is to discharge.
- Operating the LJL pump with a discharge pressure P_d less than the design P_d (same Q_1 and P_s) will yield more Q_2 flow ($M > M_{op}$) and slightly higher efficiency. But such an increase in M may reach the cavitation point, that is, where $M = M_L$, causing a marked departure of actual/measured operating points downward from the N and η characteristic curves.

OTHER JET PUMP APPLICATIONS

Virtually any fluid can be pumped by the liquid-jet pump. Suspensions of solids in a liquid can be handled with the LJL analysis given by incorporating the "S" density-ratio term. In extreme cases, slurry pumping may also be affected by slip velocity and solids concentration (see References 1 and 15).

Steam jet pumps are found in the earliest literature (see Reference 1). Analyses have usually combined one-dimensional treatment with shock-wave compressibility phenomena. Theory has not contributed to the longitudinal dimensions problem, as is the case with liquid jet pumps. Advances in applying two- and three-dimensional flow theory have contributed to understanding low Mach-number mixing length requirements. Air/air jet pump theory and performance have been compared (see Reference 16). Analysis of high-velocity compressible-flow gas-jet pumps has been less successful.

Design of a two-pump combination system (illustrated by Examples 2 and 3) required system operation at an intersection of the mechanical- and jet-pump characteristic curves, selected so both pumps operate at or near their maximum efficiency points (see References 17, 18, and 19). And the system design must of course include consideration of the pressure losses that will occur in the piping connecting the two pumps.

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SECTION 7.2

JET PUMP APPLICATIONS

ALEX M. JUMPETER

This section contains extensive design and application experience for a variety of jet pump configurations. Although this section concentrates on eductors (termed L/JL jet pumps in Section 7.1), experience with other motive (primary) and secondary fluids is also included. The theoretical developments of Section 7.1 are the basis for what is presented here, the dimensional design ratios being generally within the ranges mentioned therein. Therefore, the only theory in this section is the empiricism that is utilized in the examples and applications presented. Refer to Section 7.1 for further explanation.

DEFINITION OF TERMS

A definition of standard ejector terminology is as follows:

<i>Ejector</i>	General name used to describe all types of jet pumps that discharge at a pressure intermediate between motive and suction pressures.
<i>Eductor</i>	A liquid jet pump using a liquid as motive fluid.
<i>Injector</i>	A particular type of jet pump that uses a condensable gas to entrain a liquid and discharge against a pressure higher than either motive or suction pressure; principally, a boiler injector.
<i>Jet Compressor</i>	A gas jet pump used to boost pressure of gases.
<i>Siphon</i>	A liquid jet pump utilizing a condensable vapor, normally steam, as the motive fluid.

EDUCTORS

Design The elements of an eductor design are shown in Figure 1. Quantities involved are defined as follows:

P_1 = static pressure upstream, lb/ft² (N/m²)

P_s = static pressure at suction (nozzle tip), lb/ft² (N/m²)

V = velocity, ft/s (m/s)

γ_1 = specific weight (force) of motive fluid, lb/ft³ (N/m³)

$$\frac{P_1 - P_s}{\gamma_1} = \text{operating head, ft (m)}$$

P_s = static pressure at suction, lb/ft² (N/m²)

P_2 = static pressure at discharge, lb/ft² (N/m²)

γ_2 = specific weight (force) of mixed fluids, lb/ft³ (N/m³)

$$\frac{P_2 - P_s}{\gamma_2} = \text{discharge head, ft (m)}$$

The head ratio R_H is defined as the ratio of the operating head to the discharge head:

$$R_H = \frac{(P_1 - P_s)/\gamma_1}{(P_2 - P_s)/\gamma_2} = \frac{(P_1 - P_s)\gamma_2}{(P_2 - P_s)\gamma_1} \quad (1)$$

Because ratios are involved, it is convenient to replace specific weight with specific gravity:

$$R_H = \frac{(P_1 - P_s)(\text{sp. gr.}_2)}{(P_2 - P_s)(\text{sp. gr.}_1)} \quad (2)$$

When the suction and motive fluids are the same, no gravity correction is required and Eq. 2 becomes:

$$R_H = \frac{H_1 - H_s}{H_2 - H_s} \quad (3)$$

where $H_1 - H_s$ = operating head, ft (m)

$H_2 - H_s$ = discharge head, ft (m)

Entrainment relates the mass (flow rate) of motive fluid and suction fluid:

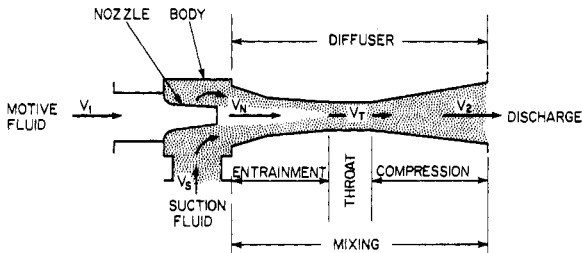


FIGURE 1 Eductor elements and terminology

M_1 = mass of motive (primary) fluid, slugs (kg)

M_s = mass of suction (secondary) fluid, slugs (kg)

$$R_w = \frac{M_s}{M_1} = \text{the weight operating ratio} \quad (4)$$

The volume ratio R_v is then simply

$$\frac{Q_s}{Q_1} = R_w \frac{\text{sp. gr.}_1}{\text{sp. gr.}_2} \quad (5)$$

where Q_s = suction (secondary) flow in volumetric units

Q_1 = motive (primary) flow in volumetric units

The performance of eductors is expressed here in terms of R_H and R_w , utilizing an empirical relationship that involves an efficiency factor ϵ as follows:

$$R_w = \epsilon \sqrt{R_H} - 1 \quad (6)$$

This equation is used to calculate the motive quantity or pressure from the operating parameters. This nozzle and diffuser diameters are calculated from the equation $Q = wAV$, using suitable nozzle and diffuser entrance coefficients. The principal problems in design concern the size and proportions of the mixing chamber, the distance between nozzle and diffuser, and the length of the diffuser. Eductor designs are based on theory and empirical constants for length and shape. The most efficient units are developed from calculated designs that are then further modified by prototype testing.

Figure 2 shows this factor plotted against NPSH (net positive suction head) for a single-nozzle and annular-nozzle eductor. In an annular-nozzle eductor, the motive fluid is introduced around the periphery of the suction fluid, either by a ring of nozzles (Figure 15) or by an annulus created between the inner wall of the diffuser and the outer wall of the suction nozzle (Figure 14). The NPSH is the head available at the centerline of the eductor to move and accelerate suction fluid entering the eductor mixing chamber. NPSH is the total head in feet (meters) of fluid flowing and is defined as atmospheric pressure minus suction pressure minus vapor pressure of suction or motive fluid, whichever is higher.

Increased viscosity of motive or suction fluid increases the frictional and momentum losses and therefore reduces the efficiency factor of Figure 2. Below 20 cP, the effect is minimal (approximately 5% lowering of ϵ). Above this value, the loss of performance is more noticeable and empirical data or pilot testing is used to determine sizing parameters.

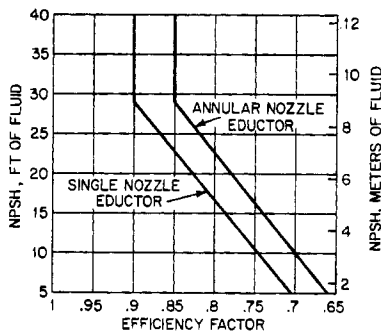


FIGURE 2 NPSH versus efficiency factor (Schutte and Koerting)

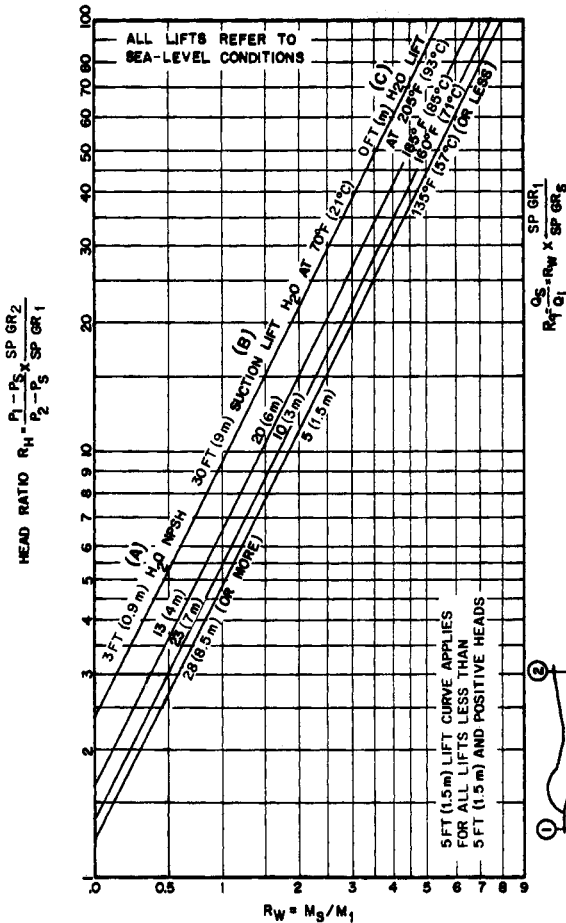


FIGURE 3 Estimating operating ratios for liquid jet ejectors. An ejector can be designed for only one head point (Schutte and Koerting)

Figure 3 shows the operating ratio R_w versus the head ratio R_H for various lift conditions. The efficiency factor has been incorporated into this curve.

The final size of the ejector is determined by the discharge line and is based on normal pipeline velocities, which are usually 3 to 10 ft/s (0.9 to 3 m/s). Figure 4a and 4b are used for estimating ejector size. To illustrate the use of Figures 3 and 4, consider the following example.

EXAMPLE 1 It is desired to remove 100 gpm (22.7 m³/h) of water at 100°F (38°C) from a pit 20 ft (6.1 m) deep. Discharge pressure is 10 lb/in² (0.69 bar*) gage. Motive water is available at 60 lb/in² (4.1 bar) gage and 80°F (26.6°C). The ejector is to be located above the pit. Find the ejector size and motive water quantity required.

*1 bar = 10⁵ Pa. For a discussion of bar, see *SI Units—A Commentary* in the front matter.

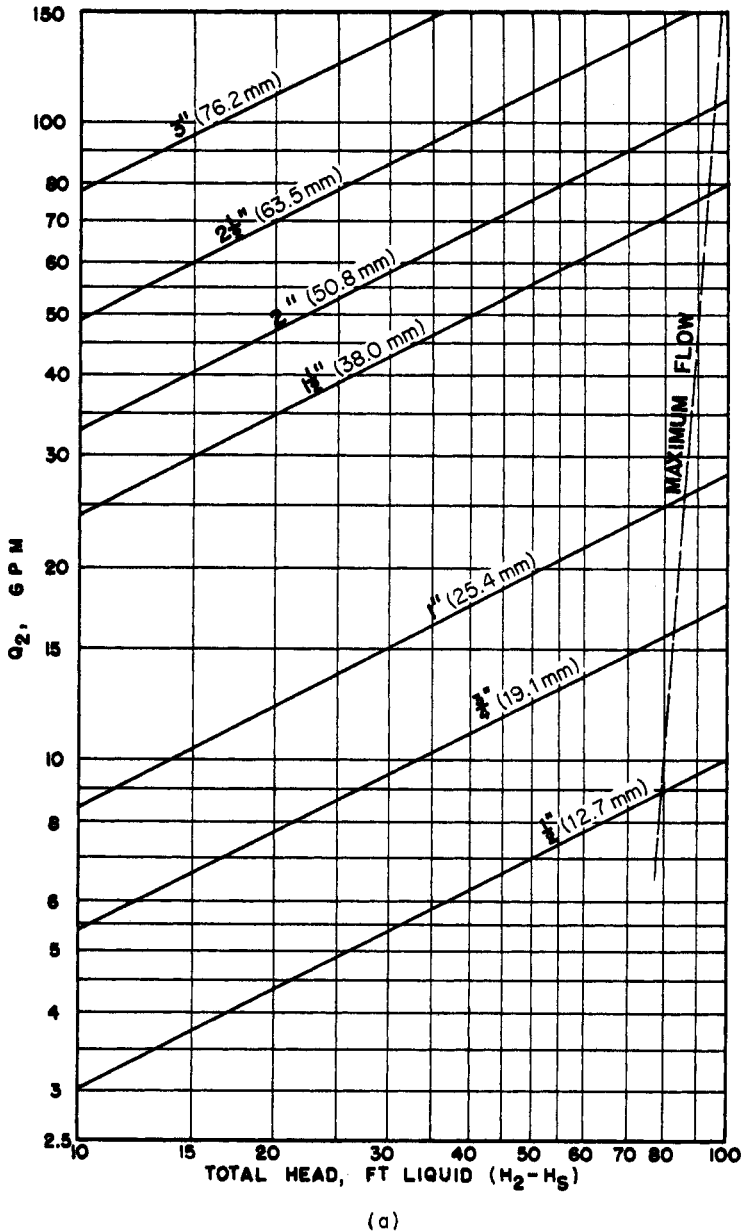


FIGURE 4A Sizing curve ($\text{gpm} \times 0.227 = \text{m}^3/\text{h}$; $\text{ft} \times 0.3048 = \text{m}$)

Solution To use Figure 3, it is necessary to determine the NPSH and the head ratio R_H . The centerline of the eductor is chosen as the datum plane, and NPSH is taken to be atmospheric pressure minus suction lift minus vapor pressure at 100°F (38°C):

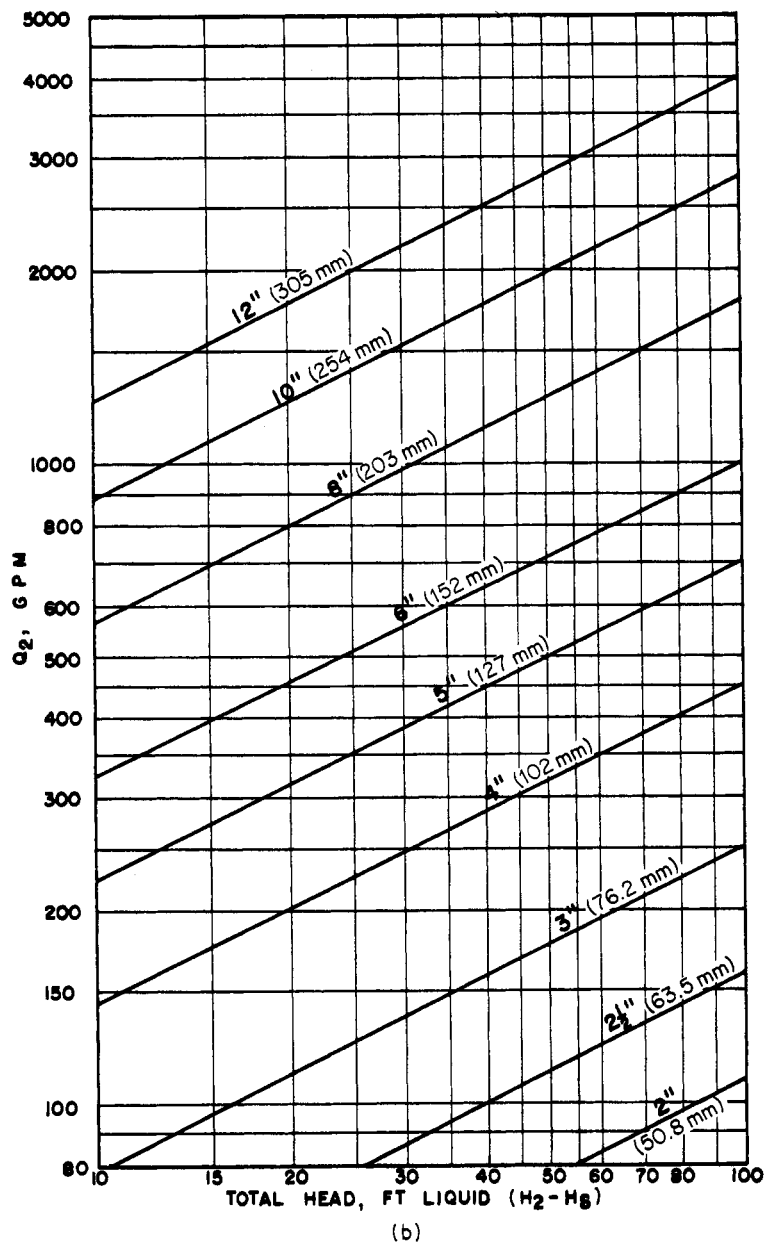


FIGURE 4B Sizing curve (continued)

$$\text{in USCS units} \quad \text{NPSH} = 34 \text{ ft} - 20 \text{ ft} - 1.933 \text{ inHg} \left(\frac{13.6}{12} \right) = 11.81 \text{ ft}$$

$$\text{in SI units} \quad \text{NPSH} = 10.36 \text{ m} - 6.1 \text{ m} - 49 \text{ mmHg} \left(\frac{13.6}{1000} \right) = 3.59 \text{ m}$$

Because motive and suction are the same fluid, it is convenient to work in feet (meters) rather than pounds per square inch (bar), and

$$P_1 = 60 \text{ lb/in}^2 \text{ gage} = 138.6 \text{ ft H}_2\text{O} \text{ (4.1 bar} = 42 \text{ m)}$$

$$P_2 = 10 \text{ lb/in}^2 \text{ gage} = 23.1 \text{ ft H}_2\text{O} \text{ (0.69 bar} = 7 \text{ m)}$$

$$P_s = -20 \text{ ft (-6.1 m)}$$

Then

$$\text{in USCS units} \quad R_H = \frac{138.6 - (-20)}{23.1 - (-20)} = \frac{158.6}{43.1} = 3.68$$

$$\text{in SI units} \quad R_H = \frac{42 - (-6.1)}{7 - (-6.1)} = 3.68$$

Enter Figure 3 at $R_H = 3.68$ and $\text{NPSH} = 11.81$ (3.59 m); read $R_w = 0.48$. Because there is no gravity correction,

$$\text{in USCS units} \quad R_w = R_q = \frac{0.48 \text{ gal suction}}{\text{gal motive}}$$

$$\text{in SI units} \quad R_w = R_q = \frac{0.48 \text{ m}^3 \text{ suction}}{\text{m}^3 \text{ motive}}$$

The same result can be obtained by using the efficiency factor from Figure 2. Then R_w is $0.77\sqrt{R_H - 1} = 0.48$ and the required motive fluid is

$$\text{in USCS units} \quad \frac{100 \text{ gpm suction}}{0.48} = 208 \text{ gpm at } 60 \text{ lb/in}^2 \text{ gage}$$

$$\text{in SI units} \quad \frac{22.7 \text{ m}^3/\text{h}}{0.48} = 47.3 \text{ m}^3/\text{h at } 4.1 \text{ bar gage}$$

Discharge flow is

$$\text{in USCS units} \quad 208 + 100 = 308 \text{ gpm}$$

$$\text{in SI units} \quad 47.3 + 22.7 = 70.0 \text{ m}^3/\text{h}$$

The size is obtained from Figure 4. Enter Figure 4b at $Q_2 = 308 \text{ gpm}$ (70 m³/h) and discharge head ($H_2 - H_3$) = $23.1 - (-20) = 43.1 \text{ ft}$ [$7 - (-6.1) = 13.1 \text{ m}$]; read eductor size of 4 in (102 mm) based on the discharge connection.

NOTE: If there were any appreciable length of run on the discharge line, it would be necessary to calculate the pressure drop in this line and recalculate the eductor size after adding the line loss to the discharge head required. Frictional losses on the suction side must also be included. In the example chosen, however, 100 gpm (22.7 m³/h) in a 4-in (102-mm) suction line 20 ft (6.1 m) long will have negligible frictional loss, less than 0.25 ft (0.08 m) H₂O.

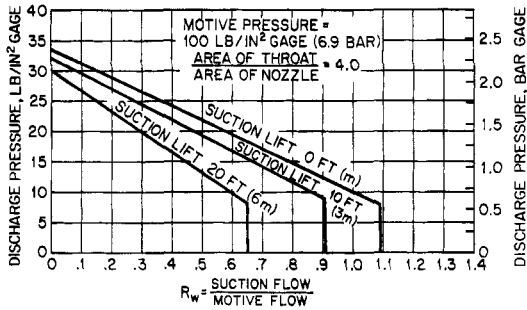


FIGURE 5 Characteristic performance of an eductor

Performance Characteristics Figure 5 illustrates the performance characteristics of eductors. Note the sharp break in flow rate below the design point. For this reason, all eductors are not designed for a peak efficiency. It is often advantageous to have a wide span of performance with lower efficiencies rather than a peak performance with very limited range. (See Section 7.1; where, also, the validity of assuming a straight-line head-vs.-suction flow characteristic is discussed.)

Applications Beside the obvious advantages of being self-priming, having no moving parts, and requiring no lubrication, eductors can be made from any machinable material in addition to special materials, such as stoneware, Teflon,* heat-resistant glass, and fiberglass. The applications throughout industry are too numerous to mention, but some of the more common will be discussed here. The type of eductor is determined by the service intended.

GENERAL PURPOSE EDUCTORS Table 1 is a capacity (flow rate) table for a general purpose eductor used for pumping and blending. This type of eductor, illustrated in Figure 6, has a broad performance span rather than a high peak efficiency point. Standard construction materials for this type of eductor are cast iron, bronze, stainless steel, and PVC. Typical uses include cesspool pumping, deep-well pumping, bilge pumping aboard ship, and condensate removal.

The following problem illustrates the use of Table 1.

EXAMPLE 2 Pump 30 gpm (6.81 m³/h) of water from a sump 5 ft (0.61 m) below ground. Discharge to drain at atmospheric pressure. Motive water available is 40 lb/in² (2.8 bar) gage.

Solution Enter left side of Table 1 at 5 ft (1.5 m) suction lift and 0 lb/in² (bar) gage discharge pressure. Read horizontally across to 40 lb/in² (2.8 bar) gage operating water pressure. Read 9.6 gpm (2.18 m³/h) suction and 7.3 gpm (1.66 m³/h) operating fluid. These values are obtained in a 1-in (25.4-mm) eductor with a capacity ratio of 1.0.

To determine the capacity ratio of the required unit, divide the required suction by the quantity handled in 1-in (25.4-mm) eductor:

$$\text{in USCS units} \quad \text{Capacity ratio} = \frac{30}{9.6} = 3.13$$

$$\text{in SI units} \quad \text{Capacity ratio} = \frac{6.81}{2.18} = 3.13$$

*Teflon is a registered trademark of E. I. DuPont de Nemours and Co., Inc.

TABLE 1 Capacity (flow rate) table of standard 1-in (25.4-mm) water-jet eductors, gpm^a

Suction lift, ft(m)	Discharge pressure, lb/in ² (bar) gage	Function	Operating water pressure, lb/in ² (bar) gage							
			10 (0.69)	20 (1.4)	30 (2.1)	40 (2.8)	50 (3.4)	60 (4.1)	80 (5.5)	100 (6.9)
0 (0)	0 (0)	Suction	5.85	8.1	9.5	10.0	12.0	12.0	12.0	12.0
		Operating	3.55	5.0	6.1	7.1	7.9	8.7	10.0	11.0
	5 (0.34)	Suction	...	1.4	4.1	6.0	8.0	10.0	11.0	12.0
		Operating	...	4.9	6.1	7.0	7.9	8.6	10.0	11.0
	10 (0.69)	Suction	...	...	0.28	2.3	4.8	6.4	8.8	11.0
		Operating	...	...	5.9	6.8	7.8	8.5	9.8	11.0
	15 (1.0)	Suction	...	...	...	...	1.2	3.4	5.9	8.6
		Operating	...	...	...	...	7.7	8.4	9.8	11.0
	20 (1.4)	Suction	...	...	...	...	...	0.3	3.5	5.9
		Operating	...	...	...	...	...	8.2	9.7	11.0
	25 (1.7)	Suction	...	...	...	...	...	...	0.83	3.9
		Operating	...	...	...	...	...	...	9.6	11.0
	30 (2.1)	Suction	...	...	...	...	...	...	...	1.7
		Operating	...	...	...	...	...	...	...	11.0
5 (1.5)	0 (0)	Suction	4.4	6.8	8.6	9.6	11.0	11.0	12.0	12.0
		Operating	3.9	5.3	6.4	7.3	8.1	8.8	10.0	11.0
	5 (0.34)	Suction	...	1.5	3.2	5.0	7.0	9.0	11.0	11.0
		Operating	...	5.2	6.3	7.2	8.0	8.7	10.0	11.0
	10 (0.69)	Suction	...	...	...	1.9	3.6	5.6	8.6	10.0
		Operating	...	...	...	7.1	7.9	8.6	10.0	11.0
	15 (1.0)	Suction	...	...	...	...	1.1	2.6	5.8	8.3
		Operating	...	...	...	...	7.8	8.6	9.9	11.0
	20 (1.4)	Suction	...	...	...	...	...	...	3.3	5.6
		Operating	...	...	...	...	...	...	9.8	11.0
	25 (1.7)	Suction	...	...	...	...	...	...	0.47	3.6
		Operating	...	...	...	...	...	...	9.8	11.0
	30 (2.1)	Suction	...	...	...	...	...	...	...	1.5
		Operating	...	...	...	...	...	...	...	11.0
10 (3.0)	0 (0)	Suction	2.0	4.6	6.7	8.3	9.0	10.0	10.0	10.0
		Operating	4.2	5.5	6.6	7.4	8.2	9.0	10.0	11.0
	5 (0.34)	Suction	...	...	2.0	4.3	5.9	7.7	9.9	10.0
		Operating	...	...	6.5	7.4	8.2	8.9	10.0	11.0
	10 (0.69)	Suction	...	...	...	1.1	3.0	4.5	8.1	9.6
		Operating	...	...	...	7.3	8.1	8.8	10.0	11.0
	15 (1.0)	Suction	...	...	...	...	1.1	2.1	5.6	7.3
		Operating	...	...	...	...	8.0	8.7	10.0	11.0
	20 (1.4)	Suction	...	...	...	...	...	...	2.8	5.3
		Operating	...	...	...	...	...	...	9.9	11.0
	25 (1.7)	Suction	...	...	...	...	...	...	...	2.8
		Operating	...	...	...	...	...	...	...	11.0
	30 (2.1)	Suction	...	...	...	...	...	...	...	1.1
		Operating	...	...	...	...	...	...	...	11.0
15 (4.6)	0 (0)	Suction	...	3.3	5.3	7.9	8.4	8.9	8.9	9.1
		Operating	...	5.7	6.8	7.6	8.4	9.1	10.0	12.0
	5 (0.34)	Suction	...	...	...	4.0	4.9	7.3	8.6	9.1
		Operating	...	...	...	7.6	8.3	9.0	10.0	11.0
	10 (0.69)	Suction	...	...	...	...	2.4	4.0	6.4	8.6
		Operating	...	...	...	...	8.2	9.0	10.0	11.0
	15 (1.0)	Suction	...	...	...	...	...	...	4.2	6.8
		Operating	...	...	...	...	...	...	...	...

(Continued)

TABLE 1 Capacity (flow rate) table of standard 1-in (25.4-mm) water-jet eductors, gpm^a (Continued)

Suction lift, ft(m)	Discharge pressure, lb/in ² (bar) gage	Function	Operating water pressure, lb/in ² (bar) gage							
			10 (0.69)	20 (1.4)	30 (2.1)	40 (2.8)	50 (3.4)	60 (4.1)	80 (5.5)	100 (6.9)
20 (6.1)	20 (1.4)	Operating	...	...	...	...	...	...	10.0	11.0
		Suction	...	...	...	...	...	...	2.1	4.5
	25 (1.7)	Operating	...	...	...	...	...	...	10.0	11.0
		Suction	...	...	...	...	...	...	...	1.9
		Operating	...	...	...	...	...	...	...	11.0
	0 (0)	Suction	...	2.0	4.0	6.4	7.8	7.8	7.8	7.8
		Operating	...	6.0	7.0	7.8	8.6	9.3	11.0	12.0
	5 (0.34)	Suction	...	...	...	2.8	3.9	6.3	7.8	7.8
		Operating	...	...	...	7.7	8.5	9.2	10.0	12.0
	10 (0.69)	Suction	...	...	...	...	1.2	3.1	5.7	7.1
		Operating	...	...	...	...	8.3	9.1	10.0	12.0
	15 (1.0)	Suction	...	...	...	...	...	...	3.6	5.4
		Operating	...	...	...	...	...	...	10.0	11.0
	20 (1.4)	Suction	...	...	...	...	...	...	1.4	3.8
		Operating	...	...	...	...	...	...	10.0	11.0
	25 (1.7)	Suction	...	...	...	...	...	...	...	1.5
		Operating	...	...	...	...	...	...	...	11.0
Relative capacities of standard sizes										
Size eductor, in (mm)	$\frac{1}{2}$ (12.7)	$\frac{3}{4}$ (19.1)	1 (25.4)	$1\frac{1}{2}$ (38.1)	2 (50.8)	$2\frac{1}{2}$ (63.5)	3 (76.2)	4 (102)	6 (152)	
Capacity ratio	0.36	0.64	1.00	2.89	4.00	6.25	9.00	16.00	36.00	

^agpm $\times$ 0.227 = m³/h.

Source: Schutte and Koerting.

Referring to the bottom of Table 1, a 2-in (50.8-mm) eductor with a capacity ratio of 4.0 is obtained. The required motive flow is then

in USCS units $4(7.3) = 29.2$ gpm

in SI units $4(1.66) = 6.64$ m³/h

and the suction capacity is

in USCS units $4(9.6) = 38.4$ gpm

in SI units $4(2.18) = 8.72$ m³/h

A $1\frac{1}{2}$ -in (38-mm) unit can handle 2.89 times the values in Table 1, or 27.7 gpm (6.3 m³/h) suction when using 21 gpm (4.8 m³/h) motive water at 40 lb/in² (2.8 bar) gage. If suction flow rate is not critical, some capacity can be sacrificed in order to use a smaller and therefore lower-cost eductor. If optimum performance is desired, it is necessary to size a special eductor using Figures 3 and 4.

Figure 7 illustrates more streamlined versions for higher suction lifts or applications involving the handling of slurries. This type of eductor is often used to remove condensate from vessels under vacuum. The advantage is that eductors require only 2 ft (0.61 m) NPSH and, being smaller than mechanical pumps, save considerable space. Further, a partial vapor load is much less likely to vapor-lock a jet pump because the venturi tube

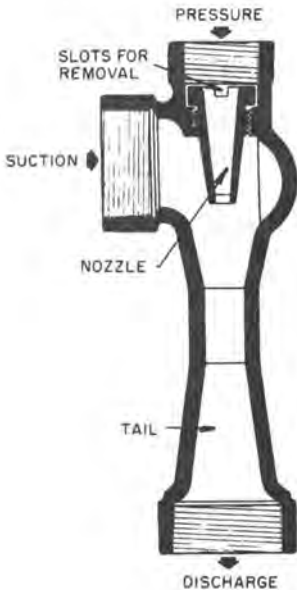


FIGURE 6 General purpose eductor (Schutte and Koerting)

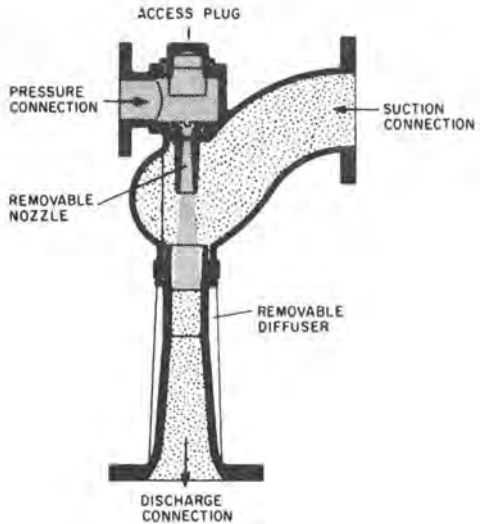


FIGURE 7 Streamlined eductor (Schutte and Koerting)

minimizes the expansion effect of flashing vapor. Sizing is done in the manner illustrated in Example 1, using Figures 3 and 4.

MIXING EDUCTORS Although any eductor is inherently a mixing device, some are specifically designed as mixers. They are used to replace mechanical agitators and are located inside the tank containing the fluid to be agitated. Figure 8 illustrates the simplest type of eductor, the *Sparger nozzle*. These units entrain a volume of suction fluid that is approximately three times the volume of motive fluid. A 20-lb/in² (1.4-bar) drop across the nozzle is recommended for proper mixing. Figure 9 shows the motive flow rates for this type

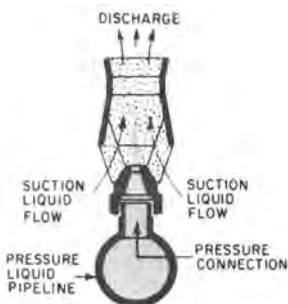


FIGURE 8 Sparger nozzle (Schutte and Koerting)

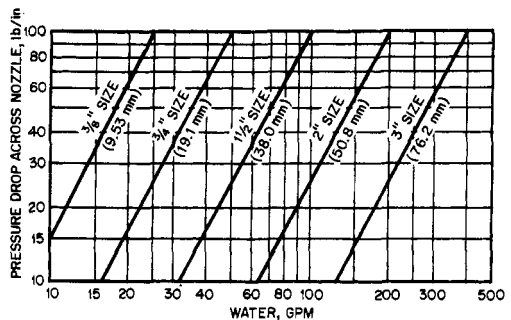


FIGURE 9 Motive flow rate of Sparger nozzles (gpm $\times$ 0.227 = m³/h; lb/in² $\times$ 0.0689 = bar) (Schutte and Koerting)

TABLE 2 Motive flow rates of tank mixing eductors, gpm^a

Pressure difference, inlet to tank, lb/in ² (bar) gage								
Size, in (mm)	10 (0.69)	20 (1.4)	30 (2.1)	40 (2.8)	50 (3.4)	60 (4.1)	80 (5.5)	100 (6.9)
$\frac{1}{4}$ (12.7)	3.5	5.0	6.0	7.0	8.0	8.5	10.0	11.0
$\frac{3}{8}$ (19.1)	10.0	14.5	17.5	20.0	23.0	24.5	29.0	32.0
1 (25.4)	14.2	20.0	25.0	28.0	30.0	34.5	40.0	44.5
$1\frac{1}{4}$ (31.8)	22.0	31.0	37.5	44.0	50.0	53.0	62.5	69.0
$1\frac{1}{2}$ (38.1)	31.5	45.0	54.0	63.0	72.0	76.5	90.0	99.0
2 (50.8)	56.0	80.0	96.0	112.0	128.0	136.0	160.0	176.0
3 (76.2)	126.0	180.0	216.0	252.0	288.0	306.0	360.0	396.0
4 (102)	224.0	320.0	384.0	448.0	512.0	544.0	640.0	704.0
5 (127)	350.0	500.0	600.0	700.0	800.0	850.0	1000.0	1100.0
6 (152)	494.0	720.0	864.0	1008.0	1152.0	1224.0	1440.0	1584.0

^agpm $\times$ 0.227 = m³/h

Source: Schutte and Koerting.

of eductor. Sparger nozzles are normally used for shallow tanks, whereas the following tank mixer described is preferred for deeper vessels.

Figure 10 illustrates a type of eductor called a *tank mixer*. It is installed under the tank containing the fluid to be agitated. Motive capacities are shown in Table 2. The units are usually custom-designed for a specific entrainment ratio, the required capacity being determined by the quantity of tank fluid, the ratio of mixture desired, and the depth of the tank being agitated.

EXAMPLE 3 It is desired to blend recycled tank fluid into a tank 20 ft (6.1 m) deep in a volume ratio of 1 motive to 1.5 suction. The tank contains 7500 gal (28.4 m³), and it is desired to turn over the tank in 30 min. The motive pump will deliver 60 lb/in² (4.14 bar) gage at the eductor nozzle. What size mixing eductor is needed?

Solution The 500 gal (28.4 m³) turned over in 30 minutes is equivalent to 250 gpm (56.8 m³/h). Because the motive fluid in this case is recycled from the tank, both motive and suction fluid contribute to the tank turnover. In the ratio of 1.5 suction to 1 motive fluid, the motive quantity required to attain a circulation rate of 250 gpm (56.8 m³/h) is 100 gpm (22.7 m³/h). To select the size, it is necessary to obtain the differential pressure across the nozzle orifice of the eductor. Because the eductor is below the tank, the net driving head is 60 lb/in² gage - 20/2.31 = 51.35 lb/in² (4.14 - 6.1/10.2 = 3.54 bar) gage across the nozzle. Enter Table 2 and interpolate between 50 and 60 lb/in² (3.4 and 4.1 bar) gage. A 1½-in (38-mm) eductor will pass only 73 gpm (16.6 m³/h), whereas a 2-in (51-mm) eductor will pass 129 gpm (29.3 m³/h). The selection would then be a 2-in (51-mm) mixing eductor.

SPINDLE PROPORTIONING EDUCTORS Another type of mixing eductor is illustrated in Figure 11. Typical applications of this type include mixing hydrocarbons with caustic, oxygen, or copper chloride slurries; producing emulsions; and proportioning liquids in chemical process industries. In critical applications, the regulating spindle is sometimes fitted with a diaphragm operator to achieve close control. Table 3 shows operating pressures and flow rates on several typical applications for units of this type.

SAND AND MUD EDUCTORS Figure 12 illustrates a sand and mud eductor used for pumping out wells, pits, tanks, sumps, and similar containers where there is an accumulation of

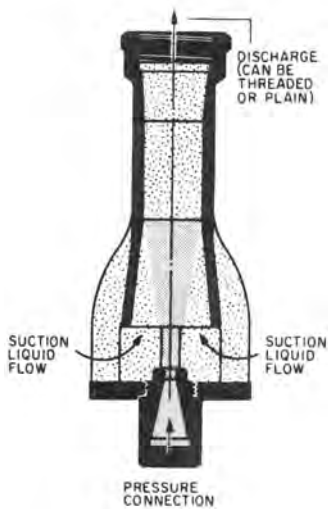


FIGURE 10 Tank mixing eductor (Schutte and Koerting)

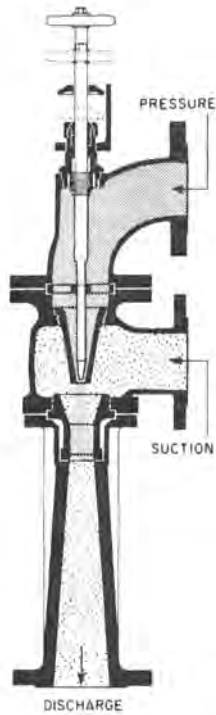


FIGURE 11 Proportioning eductor (Schutte and Koerting)

TABLE 3 Operating pressures and flow rates—proportioning eductor

Motive liquid/ suction fluid	Naphtha/ copper chloride slurry	Hydrocarbon/ hydrocarbon	Gasoline/ slurry	Gasoline/ water	Sour kerosene/ kerosene slurry
Pressure, lb/in ² (bar)					
gauge					
Motive	165 (11.4)	295 (20.3)	170 (11.7)	75 (5.2)	146 (10.1)
Suction	40 (2.8)	5 (0.3)	75 (5.2)	50 (3.4)	60 (4.1)
Discharge	75 (5.2)	10 (0.7)	100 (6.9)	50 (3.4)	70 (4.8)
Flow, gpm (m ³ /h)					
Motive	30 (6.8)	10 (2.3)	90 (20.4)	170 (38.6)	482 (109.5)
Suction	20 (4.5)	58 (13.2)	74 (16.8)	42 (9.5)	700 (159.0)
Discharge	50 (11.4)	68 (15.4)	164 (37.2)	212 (48.1)	1182 (268.5)
Eductor size, in (mm)	1½ (38.1)	3 (76.2)	4 (102)	4 (102)	6 (152)

Source: Schutte & Koerting.

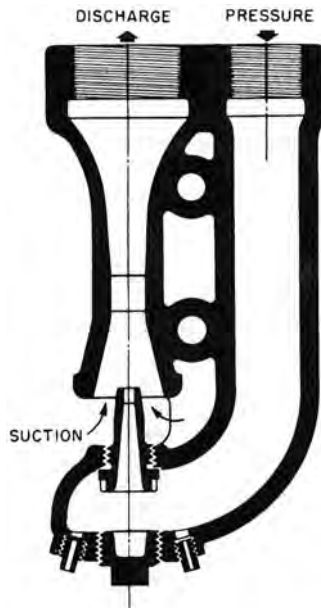


FIGURE 12 Sand and mud eductor (Schutte and Koerting)

sand, mud, slime, or other material not easily handled by other eductors. With this type of eductor, the bottom of the pressure chamber is fitted with a ring of agitating nozzles that stirs the material in which the jet is submerged to allow maximum entrainment. Relative capacities for this type of eductor are shown in Table 4, which is used in the same manner as Table 1. The required suction flow is divided by the suction capacity selected from Table 4 under the appropriate motive pressure. This value is the capacity ratio. From the table, select the eductor by choosing the next highest capacity ratio. Actual flow rates are then determined by multiplying the values in the table by the capacity ratio of the eductor selected. Maximum discharge head is read from the table.

SOLIDS-HANDLING EDUCTORS Figure 13 illustrates a specific type of eductor called a *hopper eductor*, made for handling slurries or dry solids in granular form and used for

TABLE 4 Relative capacities (flow rates) of sand and mud eductors

Capacity of standard 3-in (76.2-mm) eductor				
Operating water pressure, lb/in ² (bar) gage	40.0 (2.8)	50.0 (3.4)	60.0 (4.1)	
Total motive fluid, gpm (m ³ /h)	69.5 (15.8)	77.5 (17.6)	85.0 (19.3)	
Net suction fluid, gpm (m ³ /h)	30.0 (6.8)	34.5 (7.8)	38.5 (8.7)	
Maximum discharge head, ft (m)	22.0 (6.7)	26.0 (7.9)	32.0 (9.8)	

Relative capacities of standard sizes					
Size eductor, in (mm)	1½ (38.1)	2½ (63.5)	3 (76.2)	4 (102)	5 (127)
Capacity ratio	0.29	0.62	1.00	1.85	2.80

Source: Schutte and Koerting.

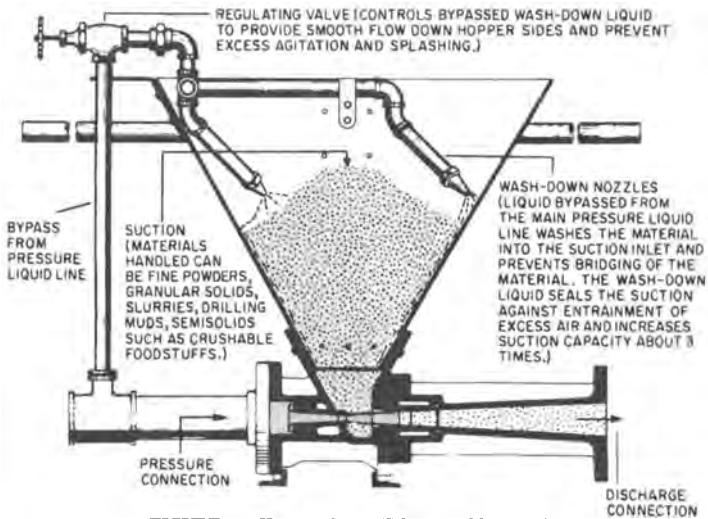


FIGURE 13 Hopper eductor (Schutte and Koerting)

ejecting sludges from tank bottoms, pumping sand from filter beds, and washing or conveying granular materials. Typical construction is cast iron with hardened steel nozzle and throat bushings. In operation, the washdown nozzles are adjusted to provide smooth flow down the hopper sides, thus preventing bridging of the material being handled and also sealing the eductor suction against excess quantities of air. Without this seal, the capacities shown in Table 5 should be divided by approximately 3. Table 6 shows typical materials handled by this eductor and their bulk density. Use of the capacity table for hopper eductors is similar to use of Tables 1 and 4, except the suction quantities required are expressed in cubic feet (cubic meters). Capacity ratio is determined by dividing the value in the table into the required suction flow, and the next largest size eductor is selected.

Another type of solids-handling eductor is illustrated in Figure 14. This *annular-orifice eductor* is used where the material being handled tends to agglomerate and gum up when wetted and has been used successfully for handling and mixing hard-to-wet solids. In this

TABLE 5 Relative capacities (flow rates) of hopper eductors

Capacity of standard 1½-in (38-mm) eductor					
Operating water pressure, lb/in ² (bar) gage	30 (2.1)	40 (2.8)	50 (3.4)	60 (4.1)	
Suction capacity, ft ³ /h (m ³ /h)	13 (0.37)	36 (1.0)	72 (2.0)	90 (2.5)	
Maximum discharge pressure, lb/in ² (bar) gage	14 (1.0)	17 (1.1)	18 (1.2)	20 (1.4)	
Motive water consumption, gpm (m ³ /h) ^a	35 (7.9)	40 (9.1)	45 (10.2)	50 (11.4)	
Relative capacities of standard sizes					
Size, in (mm)	1½ (38.1)	2 (50.8)	3 (76.2)	4 (102)	6 (152)
Capacity ratio	1.00	1.60	3.50	6.00	18.00

^aBased on using approximately 10% motive water through washdown nozzles.

Source: Schutte and Koerting.

TABLE 6 Typical materials handled by hopper eductors

Material	Approx. bulk density, lb/ft ³ (kg/m ³)	
Borax	50-55	(800-880)
Charcoal	18-28	(290-450)
Diatomaceous earth	10-20	(160-320)
Lime, pebble	56	(900)
Lime, powdered	32-40	(510-640)
Fly ash	35-40	(560-640)
Mash	60-65	(960-1040)
Rosin	67	(1070)
Salt, granulated	45-51	(720-820)
Salt, rock	70-80	(1120-1280)
Sand, damp	75-85	(1200-1360)
Sand, dry	90-100	(1440-1600)
Sawdust, dry	13	(210)
Soda ash, light	20-35	(320-560)
Sodium nitrate, dry	80	(1280)
Sulfur, powdered	50-60	(800-960)
Wheat	48	(770)
Zinc oxide, powdered, dry	10-35	(160-560)

Source: Schutte and Koerting.

unit, intimate mixing occurs in the throat, and the device is virtually clogproof. Normally this unit is installed directly over the tank into which the mixture is discharged. Table 7 shows capacities for this type of unit.

Capacity Table 7 is similar to Table 5, and the selection method is the same as discussed previously.

MULTINOZZLE EDUCTORS Figure 15 illustrates an annular multinozzle eductor designed for special applications where the suction fluid contains solids or semisolids. It is used primarily for large flows at low discharge heads. Because these units have relatively large air-handling capacities, they are well suited for priming large pumps, such as dredging pumps, where air pockets can cause these pumps to lose their prime. These eductors are designed by using the basic equations for head ratio. The appropriate efficiency factor is selected from Figure 2, and the volumetric flow ratio is calculated. Figure 4 is used to size the eductor after discharge flow has been determined.

DEEP-WELL EDUCTORS The eductor illustrated in Figure 16 is typical of those used in conjunction with a mechanical pump for commercial and residential water supply from a deep well. The eductor is used to lift water from a level below barometric height up to a level where the suction of the motive pump at the surface can lift the water the remaining distance.

In operation, the eductor is fitted with hoses connected to the suction and discharge of the motive pump and dropped into the well casing. An initial prime is required, which is maintained by the foot valve at the suction of the eductor. When the surface pump is activated, pressure water through the eductor entrains water from the well, lifting it high enough to enable the mechanical pump to carry it to the surface. A bypass valve at the surface diverts the suction quantity to a receiving tank.

Capacities of these units depend on the depth of the well and the centrifugal pump. The standard commercial unit has 1-in (25-mm) pressure and 1½-in (32-mm) discharge connections and is available with a variety of nozzle and diffuser combinations for use with standard centrifugal pumps at varying depths. The following example illustrates how to calculate this type of unit.

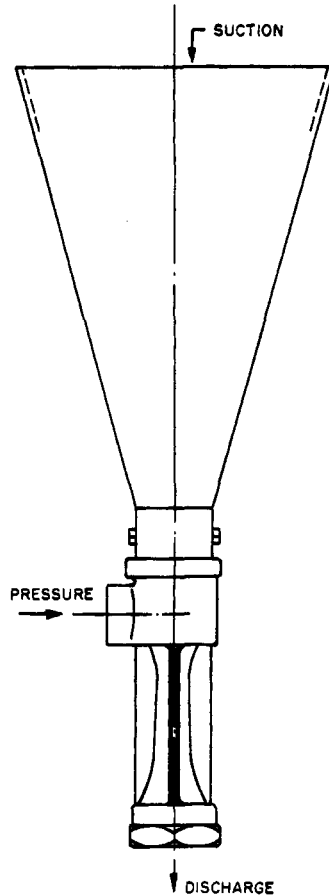


FIGURE 14 Annual eductor (Schutte and Koerting)

EXAMPLE 4 A centrifugal pump with a capacity of 100 gpm (22.7 m³/h) at a total discharge head of 150 ft (45.7 m) and requiring 10 ft (3.05 m) NPSH is available to operate an eductor to pump water at 50 ft (15.2 m) below grade. Find the quantity of water that can be delivered at 60 lb/in² (4.1 bar) gage (see Figure 17).

Solution The available operating head is 138.6 ft + 50 ft - frictional loss (42.2 + 15.2 - frictional loss). As a first assumption, the frictional loss is ignored and the head ratio is $188.6/40.83 = 4.62$ ($57.4/12.38 = 4.62$). From Figure 2 at NPSH 33.6 ft (10.24 m), $\varepsilon = 0.9$ and

$$R_w = R_q = 0.9\sqrt{4.62 - 1} = 0.934$$

(from Eqs. 5 and 6)

With Q_R fixed at 100 gpm (22.7 m³/h),

TABLE 7 Relative capacities (flow rates) of annual eductors

Capacity of standard 1½-in (38.1-mm) mixing eductor, 5 lb/in² (0.34 bar) gage discharge pressure						
Motive pressure, lb/in²(bar) gage	30 (2.1)	40 (2.8)	60 (4.1)	80 (5.5)	100 (6.9)	
Entrainment, ft³/h (m³/h)	2.6 (0.07)	7.1 (0.20)	17.9 (0.51)	22.0 (0.62)	23.8 (0.67)	
Motive flow, gpm (m³/h)	12.7 (2.88)	14.6 (3.31)	17.9 (4.06)	20.7 (4.70)	23.1 (5.24)	
Relative capacities of standard sizes						
Size, in (mm)	1¼ (31.8)	1½ (38.1)	2 (50.8)	2½ (63.5)	3 (76.2)	4 (102)
Capacity ratio	0.62	1.00	1.43	2.86	4.76	8.80

Source: Schutte and Koerting.

in USCS units $Q_1 = \frac{100}{1 + R_q} = \frac{100}{1.934} = 51.7 \text{ gpm}$

in SI units $Q_1 = \frac{22.7}{1.934} = 11.7 \text{ m}^3/\text{h}$

The motive line size is now chosen by selecting a reasonable velocity and frictional loss. Choosing a 2-in (51-mm) pipe size, velocity is 5.57 ft/s (1.70 m/s) and frictional loss is 6.1 ftH₂O (1.86 mH₂O). The revised operating head becomes 188.6 – 6.1 = 182.5 ft (57.4 – 1.86 = 55.6 m) and

in USCS units $R_q = 0.9\sqrt{182.5/40.83} - 1 = 0.9$

in SI units $R_q = 0.9\sqrt{55.6/12.38} - 1 = 0.9$

then $Q_1 = 100/1.9 = 52.6 \text{ gpm}$ ($22.7/1.9 = 11.9 \text{ m}^3/\text{h}$). (This value is close enough so that a third trial is not necessary.) The suction flow that can be delivered is then

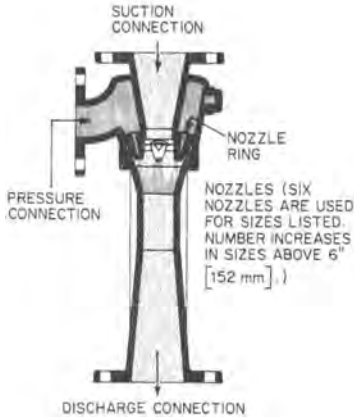


FIGURE 15 Multinozzle eductor (Schutte and Koerting)

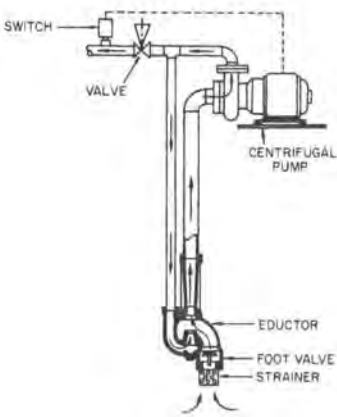


FIGURE 16 Centrifugal-jet pump combination

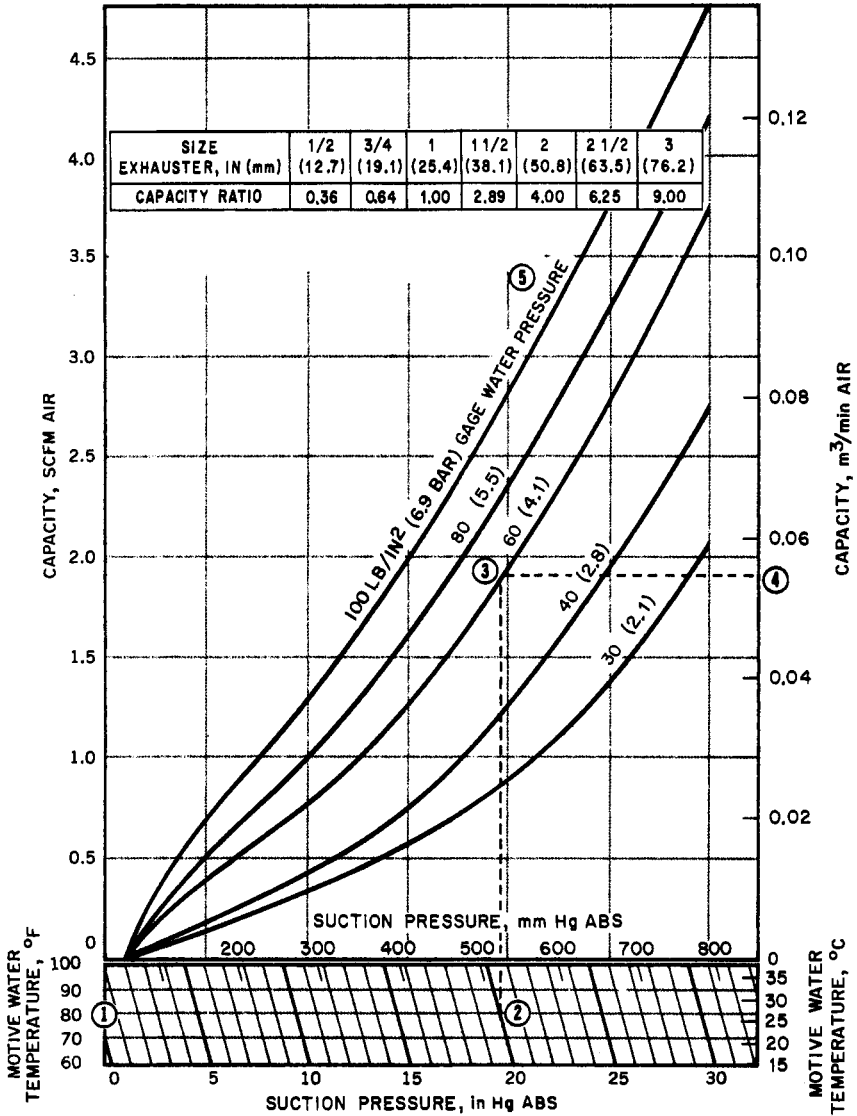


FIGURE 18 Capacity curve of water-jet exhausters (Schutte and Koerting)

in SI units $508 \text{ mmHg abs} \left(\frac{1.01 \text{ bar}}{762 \text{ mmHg}} \right) = 0.67 \text{ bar abs}$

Operating differential:

in USCS units $74.7 - 9.8 = 64.9 \text{ lb/in}^2$

in SI units $5.15 - 0.67 = 4.48 \text{ bar}$

TABLE 8 Approximate water consumption of water-jet exhausters, gpm^{a, b}

Water pressure, lb/in ² (bar) gage					
Size, in (mm)	30 (2.1)	40 (2.8)	60 (4.1)	80 (5.5)	100 (6.9)
$\frac{1}{2}$ (12.7)	2.6	2.9	3.4	3.8	4.2
$\frac{3}{4}$ (19.1)	4.6	5.3	6.4	7.4	8.3
1 (25.4)	6.2	6.8	8.1	9	10
$1\frac{1}{2}$ (38.1)	20	23	27	30	32
2 (50.8)	28	31	36	41	45
$2\frac{1}{2}$ (63.5)	46	51	60	67	73
3 (76.2)	66	73	86	96	106

^agpm $\times$ 0.227 = m³/h^bAll flows at 15 in (381 mm) Hg abs.

Source: Schutte and Koerting.

Table differential:

$$\text{in USCS units} \quad 74.7 - 15 \left(\frac{14.7}{30} \right) = 67.35 \text{ lb/in}^2$$

$$\text{in SI units} \quad 5.15 - 381 \left(\frac{1.01}{762} \right) = 4.64 \text{ bar}$$

Actual flow:

$$\text{in USCS units} \quad 86 \left(\frac{64.9}{67.35} \right)^{1/2} = 84.4 \text{ gpm}$$

$$\text{in SI units} \quad 19.5 \left(\frac{4.48}{4.64} \right)^{1/2} = 19.2 \text{ m}^3/\text{h}$$

SIPHONS

Operation As previously defined, the term *siphon* refers to a jet pump utilizing a condensable vapor to entrain a liquid and discharge to a pressure intermediate between motive and suction pressure. The principal motive fluid is steam.

In an eductor, the high-pressure motive fluid enters through a nozzle and creates a vacuum by jet action, which causes suction fluid to enter the mixing chamber. The siphon of Figure 19 is identical to the eductor of Figure 6 except that, unlike the eductor, the siphon motive nozzle is a converging-diverging nozzle to achieve maximum velocity at the nozzle tip. The velocity is supersonic at this point. The motive fluid is condensed into the suction fluid on contact and imparts its energy to the liquid, thus impelling it through the diffuser. The diffuser section is the same as an eductor diffuser, and it converts the velocity energy to pressure at the discharge. To achieve maximum performance, the siphon nozzle must be expanded to the desired suction pressure in order to achieve the highest possible velocity. Because negligible radiation losses are encountered, the siphon is 100% thermally efficient in that the heat in the incoming water plus the heat in the operating steam must equal the heat of the mixture plus its mechanical energy. Furthermore, the momentum of the incoming water plus the momentum of the expanded steam is equal to the momentum of the discharge mixture less impact and frictional losses.

It is important that the motive steam be condensed in the suction liquid prior to the throat for proper operation. If condensation does not occur, full available energy is not transferred. Furthermore, energy must be expended to recompress the uncondensed

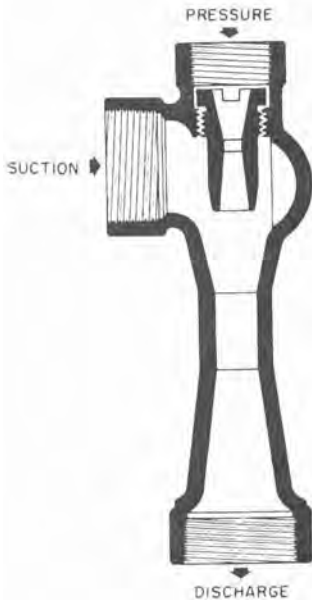


FIGURE 19 Standard siphon (Schutte and Koerting)

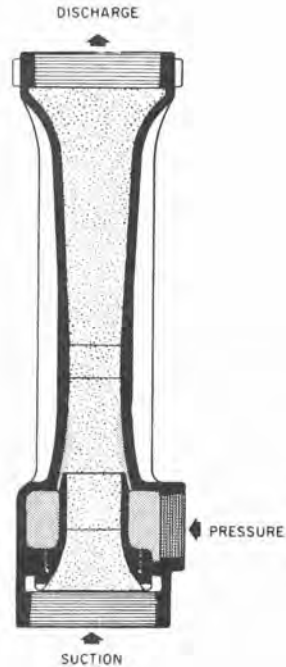


FIGURE 20 Annular siphon (Schutte and Koerting)

steam. For this reason, discharge temperature cannot exceed the boiling point at the discharge pressure. The fact that energy is required to recompress any uncondensed steam explains why air is a very poor motive fluid for liquid pumping.

STANDARD SIPHONS Tables 9 and 10 illustrate the capacity and operating characteristics of standard siphons. These tables are similar to the eductor capacity tables. To size a unit, read the suction capacity of a 1½-in (38-mm) unit from Table 9 at the appropriate motive steam pressure, suction lift, and discharge head. Divide the desired suction flow by this capacity to find the capacity ratio. From Table 10, find the unit with the next largest capacity ratio. Read the motive steam required under the proper motive pressure.

Standard materials of construction are cast iron, bronze, stainless, steel, and Pyrex®. If desired, special capacity ratios can be achieved by using a custom-designed unit. Sizes over 6 in (152 mm) can be fabricated of any suitable material.

ANNULAR SIPHONS Figure 20 illustrates an annular siphon. This unit is identical to the eductor of Figure 14 except that steam is the motive fluid. Capacity Table 11 is used in the same manner as previous examples. This type of siphon is used when inline flow is desired or when the suction liquid contains some solids. Units are available in cast iron through an 8-in (203-mm) size. Special materials or fabricated designs are also available.

OTHER JET PUMP DEVICES

Air Siphons As previously mentioned, air is a very poor motive fluid for entraining a liquid because energy must be expended in compressing the air back to the discharge pressure. There are, however, applications where it is necessary to sample a liquid with no dilution.

TABLE 9 Relative capacities of steam-jet siphons, gpm^a

Suction lift, ft (m)	Suction temp., °F ^b	Operating steam pressure, lb/in ² (bar) gage								Operating steam pressure, lb/in ² (bar) gage							
		40 (2.8)	50 (3.4)	60 (4.1)	80 (5.5)	100 (6.9)	120 (8.3)	160 (11)	240 (16)	40 (2.8)	50 (3.4)	60 (4.1)	80 (5.5)	100 (6.9)	120 (8.3)	160 (11)	240 (16)
1 (0.3)		0-ft (m) discharge head								20-ft (6.1-m) discharge head							
	70	52	51	51	49	46	43	37	30	36	47	48	48	45	41	35	29
	90	45	44	43	42	40	37	33	26	37	44	43	43	41	38	33	26
	110	40	38	36	36	35	33	28	22	38	39	37	37	36	34	30	23
	130	35	32	30	30	29	29	25	...	35	33	31	31	30	29	25	...
	150	26	25	24	24	24	24	21	...	26	25	24	24	23	22	20	...
165	17	17	17	18	18	17	17	...	17	17	17	17	17	17	16	...	
10 (3.0)	70	38	38	37	35	30	28	25	20	27	36	37	35	31	29	25	19
	90	34	34	33	30	27	25	21	17	26	32	32	29	26	23	20	17
	110	28	27	26	25	23	21	18	...	26	27	27	25	23	21	18	...
	130	21	21	21	20	18	16	14	...	21	22	22	21	18	16	14	...
	145	16	16	16	16	14	12	...	...	16	16	16	16	14	12	...	...
15 (4.6)	70	34	32	30	26	23	21	18	14	24	33	32	27	24	23	19	15
	90	29	28	26	23	20	18	16	12	23	28	27	23	20	19	16	12
	110	24	23	22	19	17	15	13	...	23	23	22	19	17	15	13	...
	130	17	17	17	15	13	11	...	...	17	17	17	14	13	...	...	...
	145	10	12	11	9	...	...	...	...	11	11	10	10	...	...	...	...
20 (6.1)	70	26	23	21	18	16	15	13	...	24	24	22	19	17	15	12	...
	90	22	19	17	15	14	12	11	...	19	20	18	15	14	12	11	...
	110	18	16	14	12	11	10	...	...	17	16	14	12	11	...	...	...
	125	13	12	11	...	...	...	...	...	12	11	10	...	...	...	...	...

(Continued)

TABLE 9 Relative capacities of steam-jet siphons, gpm^a (*Continued*)

Suction lift, ft (m)	Suction temp., °F ^b	Operating steam pressure, lb/in ² (bar) gage								Operating steam pressure, lb/in ² (bar) gage							
		40 (2.8)	50 (3.4)	60 (4.1)	80 (5.5)	100 (6.9)	120 (8.3)	160 (11)	240 (16)	40 (2.8)	50 (3.4)	60 (4.1)	80 (5.5)	100 (6.9)	120 (8.3)	160 (11)	240 (16)
1 (0.3)		40-ft (12.2-m) discharge head								50-ft (15.2-m) discharge head							
	70	...	20	33	47	44	41	36	29	...	...	18	44	44	41	36	28
	90	...	18	36	43	42	39	34	27	...	...	21	42	42	39	34	27
	110	...	20	36	37	37	35	30	24	...	...	24	37	36	34	30	24
	130	...	23	31	31	30	29	25	...	...	...	26	30	30	29	25	...
	150	...	24	24	24	24	24	20	...	...	...	24	24	24	24	20	...
165	...	17	18	18	18	18	16	...	...	...	18	18	18	18	16	...	
10 (3.0)	70	...	...	24	34	30	27	24	18	...	...	...	35	30	27	23	18
	90	...	...	26	29	26	24	20	17	...	...	...	29	27	24	21	17
	110	...	...	27	26	23	20	18	...	...	...	...	25	23	20	18	...
	130	...	...	22	21	19	16	14	...	...	...	...	21	19	17	14	...
	145	...	...	16	16	14	12	...	...	...	...	...	16	14	13	...	...
15 (4.6)	70	...	...	23	28	24	22	19	15	...	...	...	27	24	21	18	14
	90	...	...	20	24	20	18	16	...	...	...	...	24	21	18	16	...
	110	...	...	21	19	17	15	...	...	...	...	...	19	17	15	...	...
	130	...	...	17	14	12	...	...	...	...	...	...	14	...	...	...	...
	145	...	...	11	...	...	...	...	...	...	...	...	...	...	...	...	...
20 (6.1)	70	...	...	21	19	16	15	12	...	...	...	...	...	16	15	...	...
	90	...	...	18	16	14	13	11	...	...	...	...	...	15 ^c	13 ^c	...	...
	110	...	...	...	11	10	...	...	...	...	...	...	...	...	...	...	...

^agpm = 0.227 m³/h^b°C = (°F - 32)/1.8^cSuction temperature 85°F (29.4°C)

Source: Schutte and Koerting.

TABLE 10 Steam consumption of steam-jet siphons, lb/h^a

Siphon size, in (mm)	Capacity ratio	Operating steam pressure, lb/in ² (bar) gage							
		40 (2.8)	50 (3.4)	60 (4.1)	80 (5.5)	100 (6.9)	120 (8.3)	160 (11.0)	240 (16.5)
$\frac{1}{2}$ (12.7)	0.125	40	47	54	69	83	97	126	84
$\frac{3}{4}$ (19.1)	0.222	70	83	96	122	147	173	222	322
1 (25.4)	0.346	110	130	150	190	230	270	350	510
$1\frac{1}{2}$ (38.1)	1.000	318	376	434	550	665	780	1,012	1,475
2 (50.8)	1.38	440	520	600	761	920	1,080	1,400	2,040
$2\frac{1}{2}$ (63.5)	2.0	635	750	865	1,100	1,329	1,558	2,020	2,940
3 (76.2)	3.11	990	1,170	1,350	1,710	2,065	2,425	3,145	4,590
4 (102)	5.54	1,760	2,085	2,400	3,045	3,685	4,320	5,500	8,170
6 (152)	12.45	3,960	4,680	5,400	6,850	8,280	9,710	12,600	18,360

^aLb/h $\times$ 0.454 = kg/h

Source: Schutte and Koerting.

Small units (less than 1 gpm, 0.23 m³/h) can be supplied for limited discharge pressures, as indicated by Figure 21. With air as the motive fluid, the suction liquid can be very close to its boiling point and only a very slight NPSH is required.

When air is used as a motive fluid, the smaller sizes operate more efficiently because the air is more intimately mixed with a suction fluid. In larger sizes, the tendency is for the fluid to be discharged in slugs because intimate mixing does not readily occur. This has a detrimental effect on the performance and especially on available discharge head.

Air-Lift Eductors Air-lift pumps are frequently used for difficult pumping operations. Compressed air is forced into the bottom of a pipe submerged in the liquid to be pumped. The expanding air, as it rises up the pipe, entrains the suction fluid.

If compressed air is not available, it is possible to lift water higher than 34 ft (10 m) with the use of an eductor-air-lift combination. Figure 22 illustrates the suction capacity

TABLE 11 Relative capacities of annular siphons

Capacity of standard 3-in (76.2-mm) siphon, water temperature 100°F (37.8°C), 0 suction lift								
Steam pressure, lb/in ² (bar) gage	50 (3.45)	75 (5.17)	100 (6.90)	125 (8.62)				
Steam consumption, lb/h (kg/h)	1180 (535)	1620 (735)	2060 (934)	2490 (1130)				
Max back pressure, lb/in ² (bar) gage at zero flow	12 (0.83)	18 (1.24)	22 (1.52)	35 (2.41)				
Suction capacity, gpm (m ³ /h)	140 (31.8)	130 (29.5)	120 (27.2)	110 (25)				
Discharge pressure, lb/in ² (bar) gage	5 (0.34)	8 (0.55)	12 (0.83)	30 (2.07)				
Relative capacities of standard sizes								
Size, in (mm)	1¼ (31.8)	1½ (38.1)	2 (50.8)	2½ (63.5)	3 (76.2)	4 (102)	6 (152)	8 (203)
Capacity ratio	0.13	0.21	0.30	0.60	1.00	1.85	4.0	7.1

Source: Schutte and Koerting.

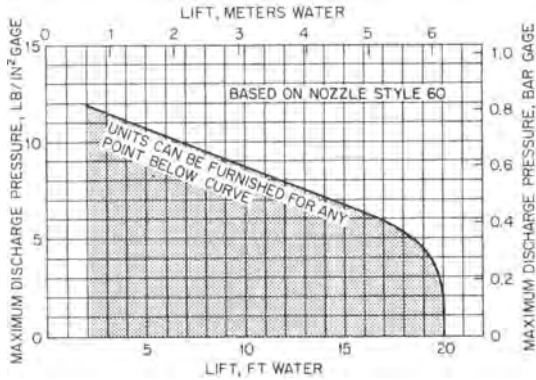


FIGURE 21 Air pumping liquid (Schutte and Koerting)

of a 2-in (51-mm) eductor drawing water from a 50-ft (15-m) depth and discharging it to the atmosphere. In operation, an air line from the atmosphere enters the suction pipe near the water level. As the eductor creates a vacuum in this line, atmospheric pressure forces air into the suction pipe. After it is in the line, the rising air carries the suction fluid to the surface and both fluids are discharged to the atmosphere through the eductor.

No sizing data are presented because this type of pump is best specified according to specific conditions.

Boiler Injectors The boiler injector is a jet pump utilizing steam as a motive fluid to entrain water, and it is used as a boiler feedwater heater and pump. It differs from a siphon in that the discharge pressure is higher than either motive or suction pressure. This is achieved by the double-tube design shown in Figure 23. In operation, the lower nozzle is activated by pulling the handle partway back. The lower jet creates a vacuum in the chamber, causing water to be induced into the unit. When water is spilling out the overflow, the handle is drawn back all the way. This closes the overflow and simultaneously admits motive steam to the upper jet. This second jet, which is of the straight or forcing type, picks up the discharge from the first jet and imparts a velocity to the water

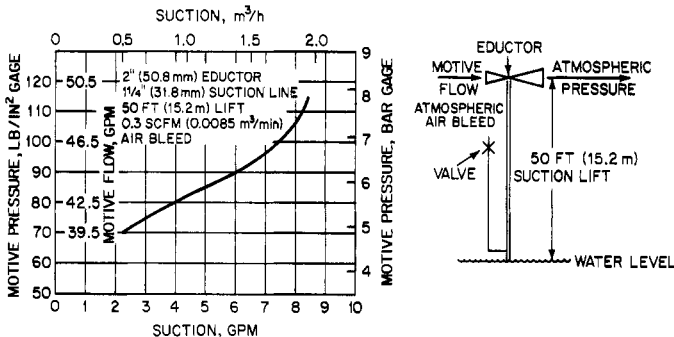


FIGURE 22 Suction capacity of eductor-air-lift combination ($\text{gpm} \times 0.227 = \text{m}^3/\text{h}$)

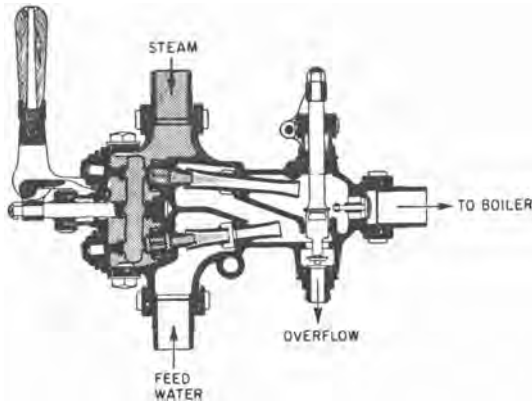


FIGURE 23 Boiler injector, starting position (Schutte and Koerting)

through the discharge tube. The energy contained is sufficient to open the check valve and discharge against the boiler pressure.

The now obsolete steam locomotives were the largest users of this type of injector. Principal use at present is as a backup to a regular boiler-feed pump. Capacity Table 12 illustrates the range of capacities available for the double-tube injector.

TABLE 12 Capacities of boiler injectors, gph^a

Size no.	Size iron pipe conn., in (mm)	Size copper pipe OD, in (mm)	Size overflow (drip funnel) pipe, in (mm)	Steam pressure, lb/in ² (bar)				
				50 (3.45)	100 (6.90)	150 (10.34)	200 (13.79)	250 (17.24)
0	$\frac{1}{4}$ (12.7)	$\frac{3}{8}$ (9.53)	$\frac{1}{4}$ (12.7)	80	100	110	100	90
1	$\frac{3}{8}$ (9.53)	$\frac{1}{2}$ (12.7)	$\frac{1}{4}$ (12.7)	110	140	180	140	130
2	$\frac{1}{2}$ (12.7)	$\frac{3}{4}$ (15.9)	$\frac{3}{8}$ (9.33)	170	210	230	200	190
3	$\frac{3}{4}$ (19.1)	1 (22.2)	$\frac{1}{2}$ (12.7)	280	340	400	340	320
$3\frac{1}{2}$	1 (25.4)	$1\frac{1}{4}$ (22.2)	$\frac{3}{4}$ (12.7)	400	470	550	470	440
4	$1\frac{1}{4}$ (31.8)	$1\frac{1}{2}$ (28.6)	$\frac{3}{4}$ (19.1)	530	620	720	620	590
5	$1\frac{1}{2}$ (31.8)	$1\frac{3}{4}$ (38.1)	1 (25.4)	680	800	920	800	750
6	$1\frac{3}{4}$ (38.1)	2 (38.1)	$1\frac{1}{4}$ (31.8)	820	990	1130	990	930
7	2 (38.1)	$2\frac{1}{4}$ (44.4)	$1\frac{3}{4}$ (31.8)	1070	1370	1610	1370	1290
8	$2\frac{1}{2}$ (38.1)	$2\frac{3}{4}$ (44.4)	2 (31.8)	1400	1800	2100	1800	1700
9	3 (50.8)	$3\frac{1}{4}$ (57.2)	$2\frac{1}{2}$ (38.1)	1700	2100	2500	2100	2000
10	$3\frac{1}{2}$ (50.8)	$3\frac{3}{4}$ (57.2)	3 (38.1)	2000	2500	2900	2500	2300
11	4 (63.5)	$4\frac{1}{4}$ (69.8)	$3\frac{1}{2}$ (50.8)	2500	3000	3500	3000	2800
12	$4\frac{1}{2}$ (63.5)	$4\frac{3}{4}$ (69.8)	4 (50.8)	3000	3600	4300	3600	3400
14	5 (76.2)	$5\frac{1}{4}$ (82.6)	$4\frac{1}{2}$ (63.5)	3900	4600	5500	4600	4400
16	6 (76.2)	$6\frac{1}{4}$ (82.6)	5 (63.5)	5000	6000	7000	6000	5700

^agph $\times$ 0.00379 = m³/h

Source: Schutte and Koerting.

C • H • A • P • T • E • R • 8

**MATERIALS
OF
CONSTRUCTION**

SECTION 8.1

METALLIC MATERIALS AND DAMAGE MECHANISMS

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A successful pump installation depends on how well it satisfies the requirements of *performance* and *life*. Performance is the rating of the pump head, capacity, and efficiency. Life is the total number of hours of operation before one or more pump components must be replaced to maintain an acceptable performance. The initial performance is the responsibility of the pump manufacturer and is inherent in the pump design. Life is primarily a measure of the resistance of the materials of construction to corrosion, erosion, wear, and other factors that can influence the materials when the pump has been placed in service. The need to maximize reliability and extend the pump life makes the selection of appropriate materials of construction crucial.

The selection of materials that are both cost-effective and technically suitable for the application requires a knowledge not only of the pump design and manufacturing processes, but also of the engineering properties of the material, particularly its corrosion and wear resistance properties when subjected to the conditions encountered in the pump. Sufficient information is available in the corrosion and metallurgical literature as well as from the experience of pump manufacturers to make appropriate material choices for virtually any pumping application.

It is known that several factors lead to a long pump life. These include:

- Neutral liquids at near-ambient temperatures
- Appropriate material selections for pumps in aggressive services
- The absence of abrasive particles
- Continuous operation at or near the maximum efficiency capacity of the pump
- An adequate margin of available NPSH over NPSH required as stated on the manufacturer's rating curve
- A low velocity (developed head/rotative speed)

Pumping installations that satisfy all these criteria should have a long life. A typical example would be a waterworks pump. Some waterworks pumps with bronze impellers and cast-iron casings have a life of 50 years or more. At the other extreme might be a chemical pump handling a hot corrosive liquid with abrasive particles carried in suspension. The life of this pump might be measured in months rather than in years, despite the fact that construction was based on the most resistant materials available.

Most pumping applications fall somewhere between these two extremes. The pump designer needs to be familiar with the various types of degradation that can affect the components of the pump and reduce its useful life. These can be grouped into the general categories of corrosion, wear, and fatigue, with corrosion and wear being the predominant life-limiting mechanisms.

TYPES OF CORROSION

General Corrosion General corrosion is corrosion that proceeds without an appreciable localization of attack. This type of corrosion occurs on metals or alloys that do not develop an effective passive film on the surface. Usually, the corrosion mechanism is oxidation with the formation of metal oxide corrosion products. General corrosion is most often encountered in pumps with carbon steels and copper-based alloys. Cast irons also experience a specialized form of general corrosion, known as *graphitic corrosion*, which will be considered separately.

Carbon steel does not develop a protective oxide film and will corrode at a rate dependent upon several characteristics of the water or other fluid, including temperature, oxygen content, pH, and fluid chemistry. Several empirical indices based on water chemistry exist and can be used to calculate the relative corrosivity of natural waters to carbon steel and similar ferrous alloys. The Langelier Index is best known. The rate of corrosion is also very dependent on velocity and increases with an increasing velocity. In most pump applications, with the notable exception of hydrocarbons, the corrosion rate of carbon steel is too high for this material to provide a useful life. However, carbon steel is frequently used, particularly in vertical pumps, with some form of protective coating to prevent corrosion. Coal tar epoxy is a preferred coating for many water services, with the notable exception of potable water, where there are health-related concerns.

Copper alloys, including both brasses and bronzes, are also subject to general corrosion in the water applications where they are most commonly used in the pump industry. The corrosion rate will be increased by the presence of small amounts of sulfides in the water. Copper alloys gradually develop a protective copper oxide corrosion film in most applications. The corrosion rate gradually decreases over time as this film develops. The rate of general corrosion varies with the specific type or grade of copper alloy. Among the alloys commonly used in pumps, nickel aluminum bronzes have the lowest corrosion rate and best tolerance for higher velocities.

The general corrosion of a Ductile Ni-Resist casing from a vertical pump is shown in Figure 1. A metallographic cross section was removed to show the depth of the corrosion attack.

Localized Corrosion There are generally considered to be two types of localized corrosion, pitting and crevice corrosion. Localized corrosion is most commonly found in materials that tend to develop a relatively stable oxide film, such as stainless steels, nickel-based alloys, and aluminum alloys. Pitting, or crevice corrosion, will develop at microscopic flaws or discontinuities in the oxide film. Some corrodent, usually chloride or another halogen ion, initiates the corrosion. Very often, a microgalvanic couple is set up between the interior of the pit and the external unattacked surface. The interior contains acidic, hydrolyzed salts, which are quite corrosive in comparison with the bulk solution. An anode is established within the pit, with the surrounding surface acting as the cathodes. This is particularly the case for alloys that rely on a resistant, passive film for protection, such as stainless steel, titanium, and aluminum.

Stainless steels are extensively used in pumps. Localized corrosion is the most common type of corrosion experienced by these materials. Attack by aqueous chlorides is the usual

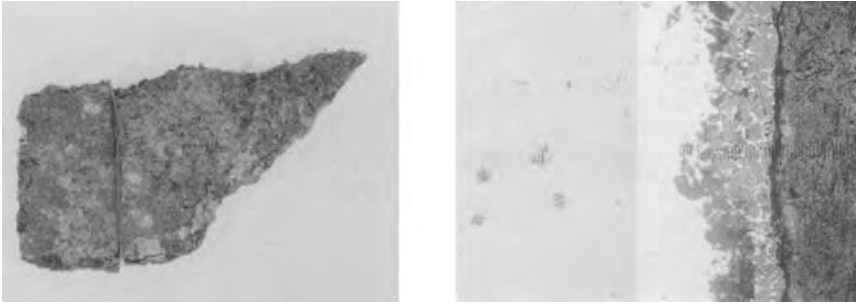


FIGURE 1 A small fragment of Ductile Ni-Resist from the lower casing of a vertical pump. The microstructure is also shown on the right side of this figure, illustrating the depth of the corrosion's penetration. This is a classic example of general corrosion (right photo at 100×).

cause for the corrosion. Susceptibility to localized corrosion is directly related to the alloy content of these steels. An empirical relationship, known as the pitting resistance equivalent (PRE), has been developed that relates the pitting resistance in seawater to the chemical composition of the stainless steel by the following relationship:

$$\text{PRE} = \% \text{Cr} + 3.3\% \text{Mo} + 15\% \text{N}$$

where Cr = chromium

Mo = molybdenum

N = nickel

This formula is applied primarily to austenitic and duplex grades of stainless steel. Test data and field experience indicate that stainless steels with a PRE of 40 or higher will be fully resistant to pitting corrosion in seawater at ambient temperature.

Crevice corrosion occurs in constricted areas where free access to the surrounding liquid is restricted. Among the more common locations for crevice corrosion to occur in pumps are various types of metal-to-metal surfaces, such as threaded fasteners, couplings, or joints. Crevice corrosion also occurs at metal to nonmetal surfaces, such as under gaskets. A specialized form of crevice corrosion is deposit attack, which suggests that the crevice has been formed by a discontinuous deposit of a foreign substance from the environment adhering to the surface.

There are several commonly used methods for evaluating the resistance of alloys to crevice corrosion. The ASTM G78 standard describes a method for conducting crevice corrosion tests, and identifies factors that may affect results and influence conclusions. This method uses segmented nonmetallic washers, having a series of grooves and plateaus, which are attached to a corrosion coupon to create a multiple crevice assembly. Samples are evaluated based on the number of sites showing initiation of crevice corrosion, and the depth of this corrosion. The results can be used to determine a crevice corrosion index, which indicates the relative resistance of the alloy to crevice corrosion.

Another technique for determining resistance to crevice corrosion is determination of the critical crevice temperature (CCT). This is the temperature at which crevice corrosion initiates in a ferric chloride solution in a precisely controlled twenty-four hour test. The results of this test correlate well with crevice corrosion resistance in aerated seawater.

Dealloying Dealloying is the preferential removal of one phase from a multiphase alloy, or one element from a material. Several types of dealloying occur in the pump industry. One of the most common is the graphitic corrosion of gray cast iron. This material is low cost, easy to machine, and well suited for a variety of applications, especially in the waterworks industry. It is probably the most widely used material in the pump industry.

Gray cast iron corrodes by a fundamentally different mechanism than carbon steel or ductile cast iron. The structure of gray cast iron consists of interconnected graphite flakes



FIGURE 2 The interface between the advancing graphitized front and the sound base metal. Graphitic corrosion propagates along the path of the graphite flakes (50 $\times$).

in a matrix that is predominantly iron. In the presence of an electrolyte, which is usually water, a galvanic cell is established between the iron and graphite. The iron corrodes, and the corrosion products are largely flushed away with the fluid passing through the pump. The original casting is gradually reduced to a porous graphite structure that may contain some iron oxide corrosion product. This is frequently referred to as *graphitization*. The surface of a gray iron casting that has suffered graphitic corrosion will retain its original shape and dimensions, but the surface will be largely graphite, which can be cut with a knife. The casting will lose some fraction of its mechanical properties and become increasingly susceptible to brittle failure, resulting from modest shock or impact loads. This is also the corrosion mechanism for Ni-Resist in seawater. Figure 2 shows the interface between the sound base metal and the graphitized surface layer.

It is important to recognize that the rate of graphitic corrosion varies with the water chemistry, and that this type of corrosion can occur in both fresh and salt waters. The high conductivity of salt water corresponds to a higher corrosion rate. Graphitic corrosion will proceed at a slower pace in waters that have a high mineral content. Minerals tend to plug the graphitic layer on the surface, sealing off the base metal from exposure to the fluid, thereby reducing the corrosion rate.

As the surface of a cast-iron component, such as a pump casing, gradually graphitizes, the galvanic relationships with other components within the pump will be altered. It has been observed that the bronze impeller originally supplied in a cast-iron pump handling seawater will provide a significantly longer life than bronze impellers that are installed after the pump has been in service for several years. The reduced life of the replacement impellers is caused by an altered galvanic relationship with the pump casing. Initially, the casing was cast iron, which is anodic to a bronze impeller. With time, as the casing graphitizes, it gradually becomes cathodic, due to the influence of the graphite. The bronze impeller is now the anode and corrodes at a much higher rate. This example highlights the influence that graphitic corrosion can have on other components within the pump and the importance of carefully selecting materials for use in conductive fluids, such as saltwater.

Several other types of dealloying can also occur in pumps. Brass and bronze alloys containing more than about 14% zinc are subject to a form of dealloying known as *dezincification*. The zinc is preferentially corroded from the matrix of the material, leaving a spongy, copper-rich residue. Dezincification can occur either uniformly in a shallow layer over the surface of the casting or as a distinct plug confined to a small area. Plug-type dezincification is a more serious problem because the plug is weak and will cause leakage if it penetrates a pressure boundary. It should be emphasized that copper alloys containing less than 14% zinc are not susceptible to this form of corrosion. Consequently, the requirement often imposed upon pump manufacturers for zinc-free bronzes to avoid dezincification is without technical justification. Figure 3 shows the dealloying of an impeller.

The final type of dealloying that occasionally occurs in pumps is dealuminification in aluminum bronzes. These are metallurgically complex materials. Some compositions can form an aluminum-rich phase that can be preferentially corroded in aggressive fluids,

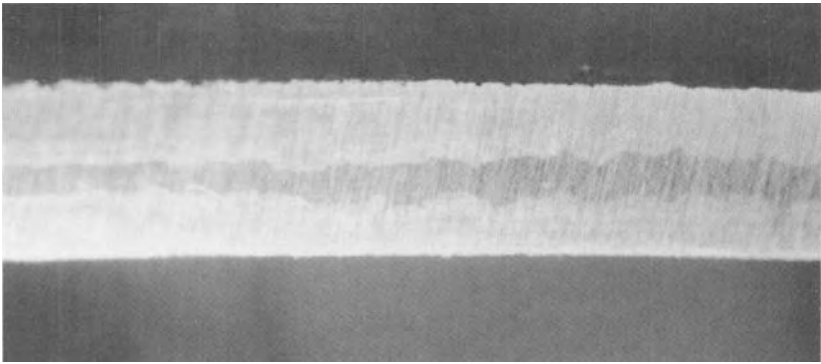


FIGURE 3 The dealloying of a vertical turbine pump impeller. Note the change in color across the cross section. The unaffected bronze (light color) material is surrounded by a dezincified layer (1.3×).

especially seawater. The detrimental phase can be mitigated by a special heat treatment known as *temper annealing*. This heat treatment must be specified by the designer for susceptible compositions, because it is not a mandatory requirement of national material specifications. The chemistry of some aluminum bronze alloys from Europe has been adjusted to preclude the formation of the detrimental aluminum-rich phase without the need for the temper annealing heat treatment. The temper anneal can serve as a stress-relief operation for fabricated aluminum bronze structures, which is a secondary benefit for products in this category.

Galvanic Corrosion *Galvanic corrosion* refers to the corrosion that occurs when one alloy is electrically coupled to another and exposed in a conductive liquid. Usually, the corrosion rate of the more noble alloy will be less than if it were exposed uncoupled. The corrosion rate of the less noble material will be greater than if it were exposed uncoupled.

Several factors influence the rate of galvanic corrosion of both metals. This corrosion is greatly influenced by the conductivity of the fluid. In a fluid such as freshwater, which has a low conductivity, galvanic corrosion will be less severe and generally confined to the immediate location where the metals contact one another. However, in a highly conductive fluid, such as seawater, galvanic corrosion will be more severe and will occur over a wider area. The pump designer needs to consider the possibility of such corrosion when using dissimilar metals in a conductive fluid.

Galvanic corrosion problems in seawater and other conductive fluids can be avoided by the careful use of materials. Galvanic corrosion is related to the area ratios of the coupled metals. It is always desirable to have the area of the anode, or less noble metal, equal to or greater than that of the more noble metal. In this way, the additional corrosion experienced by the less noble metal will be spread over a relatively large area and will not be excessive because of being coupled. An example of the effective use of this galvanic relationship involves centrifugal pumps having a Ni-Resist casing and austenitic stainless steel internals. This combination is often specified for seawater services. The Ni-Resist is anodic to the stainless steel and will protect it from localized corrosion when the pump is shut down and contains stagnant water. The area of Ni-Resist is considerably larger than that of stainless steel. The increased galvanic corrosion of the Ni-Resist is spread over a large area and is negligible.

The amount of corrosion that will occur in a galvanic couple also depends on the freely corroding potentials of the coupled metals. Less corrosion-resistant metals, such as zinc, cast iron, and steel will usually have more negative potentials when measured against a standard reference electrode. More corrosion-resistant metals, such as stainless steels, will have less negative potentials.

The corrosion potentials for many commonly used engineering alloys in slowly moving seawater are shown in Table 1. The alloys are listed in the order of the potential that they

TABLE 1 Corrosion potentials in flowing seawater (8 to 13 ft/s, 50 to 80°F/2.4 to 4.0 m/s, 10 to 26°C)

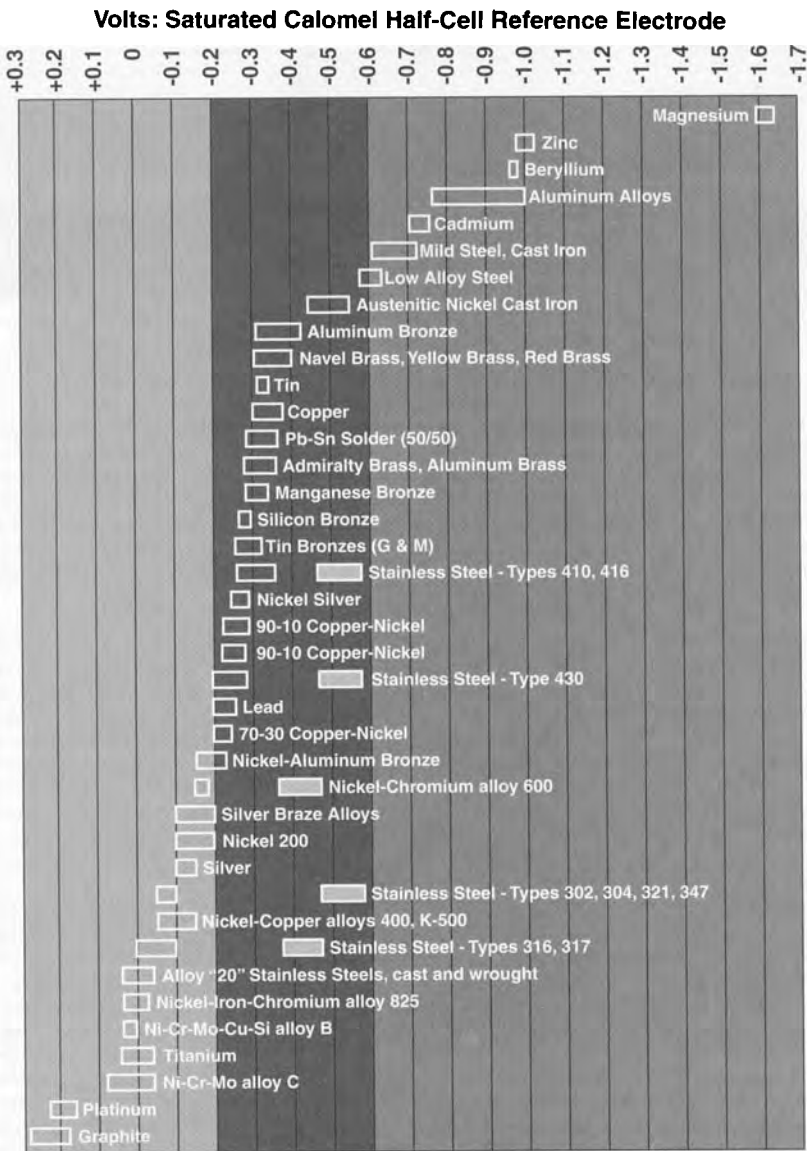


exhibit in flowing seawater. Certain alloys (indicated by solid colored boxes preceding the name of the alloy) in low-velocity or poorly aerated water and at shielded areas may become active and exhibit a potential near - 0.5 volts. The extent of galvanic corrosion that will occur when two metals are electrically coupled will depend on the potential difference between the metals. The corrosion rate of zinc coupled to stainless steel will increase

dramatically because of the large potential difference between these two metals. A nickel aluminum bronze coupled to austenitic stainless steel will experience little galvanic corrosion because the potentials of these two metals are close to one another. The pump designer needs to be aware of the corrosion potentials of dissimilar metals used in conductive fluids in order to avoid unanticipated galvanic corrosion problems.

The use of coatings can decisively alter the galvanic relationships in a pump. If the more anodic component, such as a steel casing, is coated, one can expect a high rate of corrosion at those locations where the coating eventually begins to fail. This will be caused by a very unfavorable area ratio, with a small area of exposed carbon steel coupled to a large area of some more noble metal, such as stainless steel or bronze. Experience has shown that this type of galvanic corrosion problem can occur even in relatively low conductivity fluids, such as freshwater. For this reason, coatings should be employed with caution in pumps handling conductive fluids that are constructed of dissimilar metals. It is generally advisable in these applications not to coat the anodic component, usually the casing. If coatings are used, periodic maintenance will be necessary to avoid galvanic corrosion. Figure 4 documents the galvanic corrosion on the interior diameter of a carbon steel flange connected to a stainless steel shroud. The accelerated corrosion is due to the unfavorable ratio of stainless steel to carbon steel in this component.

Stress Corrosion Cracking *Stress corrosion cracking* (SCC) is a particularly dangerous form of corrosion because it is not easily detected before it has progressed to such an extent that it can cause sudden catastrophic damage. Although relatively uncommon in pumps, it can occur in several classes of materials. The pump designer should be aware of the potential combinations of material and environment that can cause SCC.

Stress corrosion requires that several factors be present. These include tensile stress, which can be either residual or applied, a susceptible material, an environment capable of causing stress corrosion, and time.

The materials used in the pump industry that may experience SCC include austenitic and martensitic stainless steels, some copper-based alloys, and, occasionally, Ni-Resist. The austenitic stainless steels are susceptible to stress corrosion in aqueous chlorides at temperatures above about 140°F (60°C). Cast alloys, which contain some fraction of ferrite in the microstructure, are significantly more resistant to stress corrosion than their wrought counterparts. The possibility of cracking is increased in situations where chlorides are concentrated, as by evaporation. High residual stress, often present in as-welded



FIGURE 4 Galvanic corrosion is evident on this pump section. Note the high corrosion rate on the interior diameter of the carbon steel flange that is attached to the stainless steel shroud.

structures, also enhances the possibility of cracking. Increasing nickel content in austenitic stainless alloys enhances the resistance to SCC. The high nickel grade, commonly known as Alloy 20, is often used in chemical applications where the optimum resistance to stress corrosion is necessary. The SCC of austenitic stainless steels in pumps is relatively uncommon.

Martensitic stainless steels are susceptible to cracking in the presence of hydrogen sulfide and is often referred to as *sulfide stress corrosion cracking* (SSC). These steels, particularly CA-15 and CA-6NM, are commonly used in pumping applications in oil production and refining where hydrogen sulfide can be present. SCC can be avoided by giving these materials a special heat treatment intended to reduce hardness below a certain threshold level, below which cracking will not occur. This has also been correlated to the yield strength of a material. It is often seen in literature that ferrous materials used in these services should have a hardness no greater than $22 R_c$, or a yield strength no higher than 90,000 lb/in² (620 MPa). Technical standards, including API 610 and NACE MR-01-75, can be used to specify appropriate requirements for martensitic steels, which will be used in environments containing hydrogen sulfide.

Copper alloys are susceptible to SCC in the presence of ammonia, although considerable variations take place in the susceptibility of the various types of bronzes, with aluminum bronzes being the most resistant. Polluted natural waters can contain ammonia, and for this reason, bronze pumps are usually not a good choice for these applications.

High-strength manganese bronzes are susceptible to cracking in natural waters. Cast impellers in these alloys have been known to suffer severe cracking. Residual stress in the casting may also be sufficient to induce cracking. These alloys should not be used in pumps because of their susceptibility to such problems.

Ni-Resist is an austenitic cast iron that contains 15 to 20% nickel. This material is commonly used in large, seawater vertical pumps. Experience has shown that it is subject to SCC, especially in the diffuser section of these pumps, unless the castings are furnace stress-relieved. This must be specified by the purchaser, as it is not a requirement of national material specifications.

Hydrogen Embrittlement Hydrogen damage is a form of environmentally assisted failure that results from the combined action of hydrogen and residual or applied tensile stress. Hydrogen damage to specific alloys or groups of alloys manifests itself in many ways, such as cracking, blistering, hydriding, or as a loss of tensile ductility. Collectively, these various forms of damage are often referred to as *hydrogen embrittlement*.

Damage caused by hydrogen is occasionally encountered in pumps. Some plating processes, such as chrome plating, which is often used to rebuild pump shafts, generate hydrogen. This hydrogen can enter the surface of the metal. Microscopic cracks can occur in higher strength steels (greater than a 90,000-lb/in² or 620-MPa yield strength). Abusive grinding can work-harden the surface of lower strength steels and increase the probability that hydrogen will cause cracking. Microscopic cracks resulting from hydrogen damage act as stress risers and can propagate failure catastrophically by mechanical fatigue. This problem can be avoided by utilizing proper grinding practices before plating. Higher strength steels should be baked, to drive off hydrogen, immediately after plating.

Hydrogen can also be introduced into metals during welding. In order to avoid the hydrogen damage associated with welding, ferritic and martensitic steels should be welded with low hydrogen electrodes. Coated electrodes should be baked, in accordance with manufacturer's instructions, prior to usage in order to drive off moisture, which is the major source of hydrogen contamination of welds.

Microbiologically Induced Corrosion Living organisms can promote corrosion in many different environments. A variety of biological organisms thrive in both aerobic and anaerobic environments. Corrosion attributable to microbiological activity occurs most frequently in stagnant water, which remains in a pump when it is shut down for an extended length of time.

Sulfate-reducing bacteria are found in many waters. They will form slimy, reddish, hemispherical-shaped mounds or colonies on cast iron or carbon steel. These are known as *tubercles*. If scraped off, there will invariably be a saucer-shaped pit beneath the tubercle.

The inside of the pit will contain a wet, black deposit. The pitting is caused by traces of sulfuric acid excreted by the bacteria. This type of corrosion will usually not result in premature failure.

Several more serious types of microbiologically induced corrosion afflict stainless steels. A certain class of metal ion concentrating/oxidizing microbes appears to concentrate ferric and manganic chlorides, both of which are potent pitting agents. These bacteria form colonies preferentially at welds in austenitic stainless steels and are capable of causing severe pitting corrosion in a relatively short time. This problem has been encountered in a variety of equipment in both salt and freshwater. It is often discovered only when the welds begin leaking. Pumps employing welded stainless steel fabrications can be afflicted by this problem if permitted to sit idle with stagnant water, either fresh or salt, for an extended period. Biocides can be used to mitigate this problem in some instances.

Finally, the decay of biological organisms can generate hydrogen sulfide, which adversely affects the protective oxide film on copper-based alloys. The enhanced biological activity in warmer tropical waters, especially under stagnant conditions, can impair the corrosion resistance of bronzes and reduce the threshold velocity at which accelerated corrosion will occur. Bronzes should be used with caution in applications where microbiological activity is anticipated and the possibility of extended shutdowns is possible.

Stray Current Corrosion This type of corrosion is sometimes assumed, incorrectly, to be synonymous with galvanic corrosion. In fact, while the resulting corrosion can have some similarities, the cause for stray current corrosion is much different. Stray current corrosion is caused by an extraneous source of dc current passing through the pump. The most frequent source is some defect in the motor, which allows current to escape, pass through the pump shaft, and exit into the fluid. Other possible sources of stray current include defective or poorly designed cathodic protection systems, and grounding of a heavy-duty electrical device, such as a welding torch, through a pump. In the special case of submersible pumps, there have been instances where the insulation on the power cable is cut or damaged, usually during installation of the pump. Current will escape at this location, enter the pump, and cause extensive corrosion. Figure 5 shows a submersible pump bowl perforated in several locations as a result of stray current emanating from a damaged power cable.



FIGURE 5 A submersible pump bowl perforated in several locations as a result of stray current emanating from a damaged power cable

Stray current is often severe. The corrosion occurs at the location where the current exits a metal surface and enters the fluid. The extent of corrosion can be calculated using Faraday's law:

$$\text{Weight of metal reacting} = KIT$$

where I is in amperes, T in seconds, and K is a constant known as the electrical equivalent.

The corrosion often occurs as large, saucer-shaped pits. The base of the pit is smooth and shiny, with no buildup of corrosion products. Stray current should be suspected when severe pitting develops in a pump that is handling a noncorrosive fluid. One should also consider the possibility of stray current corrosion when only one of several pumps develops corrosion, or when corrosion develops in an application where it had not occurred previously, or where there is no other plausible explanation for the corrosion. Although stray current is more often seen in submersible or vertical pumps, it has also been found in horizontal split case pumps. The nominal corrosion resistance of the alloy plays no role in determining the extent of stray current corrosion, which is governed solely by Faraday's law.

Intergranular Corrosion This infrequent type of corrosion preferentially attacks a material at the grain boundaries. This is caused by local chemical differences such as the chrome-depleted regions of an austenitic stainless steel. Bronze alloys susceptible to this type of corrosion include aluminum bronzes, silicon bronzes, Muntz metal, and admiralty metal. Two things are necessary: a sensitized material and a corrosive media, such as seawater. Sensitization refers to chrome carbide precipitation at the grain boundaries in austenitic stainless steels and can occur during heat treatment or more commonly during weld repair. This type of corrosion often leads to corrosion-assisted fatigue cracks when cyclic loading is present.

The improper heat treatment of 300 series austenitic stainless steels can result in sensitization and subsequent intergranular corrosion. Sensitization occurs when stainless steels that contain more than 0.03% carbon are held at temperatures between 800 and 1550°F (between 425 and 850°C). At these temperatures, chrome carbides precipitate along the grain boundaries, resulting in chrome depletion in the adjacent areas. These adjacent areas have reduced corrosion resistance. Austenitic stainless steels contain approximately 16 to 18% chrome. The chromium content in the areas surrounding a chrome carbide particle can drop below the 12% necessary to maintain a passive state. A galvanic cell is set up with a large cathode (grains) and a small anode (grain boundaries). In this undesirable scenario, corrosion occurs along the anodic grain boundaries. The extent of the corrosion damage depends on the length of time held within the sensitization temperature range. The degree of sensitization is a function of the carbon content; the higher the carbon content, the shorter the period of time the material can be held within this range without sensitization occurring. A graph of the temperature versus time for various carbon contents illustrates this point in Figure 6. Intergranular corrosion of an improperly heat-treated stuffing box cover is shown in Figure 7.

Austenitic stainless steels can also be sensitized during normal welding procedures. Care must be taken to avoid the sensitization range during welding followed by proper postweld heat treatment when necessary.

Sensitization can be avoided or corrected by several methods:

- Heat the material to a temperature high enough to dissolve the chrome carbides, typically 1900 to 2100°F (1040 to 1150°C), followed by rapid cooling through the sensitization range. Localized heat treatment of welded areas will not desensitize a material.
- Use a stainless steel that is stabilized by the addition of niobium or titanium. These two elements will tie up the carbon, thus preventing chrome carbides.
- Reduce the carbon content to a low level (less than 0.03%). The lower the carbon content, the longer it takes chrome carbide precipitation to occur.

Austenitic stainless steel castings should always be supplied in the solution annealed and quenched condition. If low carbon grade material is used, (i.e., CF-3M), postweld heat

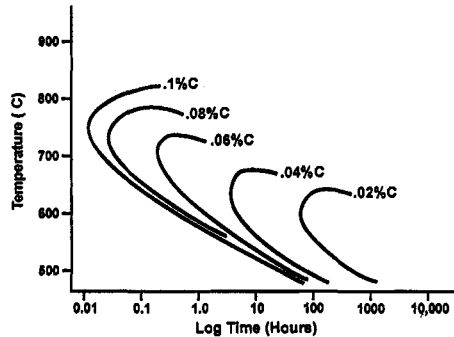


FIGURE 6 Time-temperature sensitization curves as determined by the Strauss test for 18-8 stainless steel. Note that a low carbon grade of stainless (0.03% C) requires 5 to 10 h exposure, while a standard grade (0.08%) needs only minutes of exposure time ($^{\circ}\text{F} = 1.8 \times ^{\circ}\text{C} + 32$).

treatment (to eliminate sensitization) will not be necessary following repair welding. This is a significant benefit in the event welding is required on a finish-machined casting.

Cavitation Erosion Cavitation erosion is primarily a mechanical process, although it acts synergistically with corrosion and is often considered with other forms of corrosion. Cavitation erosion can be defined as metal removal from the surface caused by high stresses associated with the collapse of vapor bubbles in the fluid. Cavitation occurs in a pump when the local pressure of the fluid is reduced to the vapor pressure. In a multi-stage pump, vapor bubbles form in the low-pressure areas at the impeller inlet and are swept by the flow into regions of higher pressure where they collapse. A great many bubbles may form and collapse in a small area, producing many microjets of high kinetic energy. The energy released by the bubble collapse is expended as impact loading on the metal surface. This situation is aggravated if protective oxide films are present because these are damaged, exposing fresh metal to the corrosive action of the fluid. This cyclic loading eventually causes the formation of microscopic fatigue cracks. These cracks propagate and intersect, resulting in the removal of metal from the surface and the characteristic



FIGURE 7 The surface of a stuffing box cover that experienced intergranular corrosion due to sensitization. The grains are clearly evident on the interior of the bore as well.

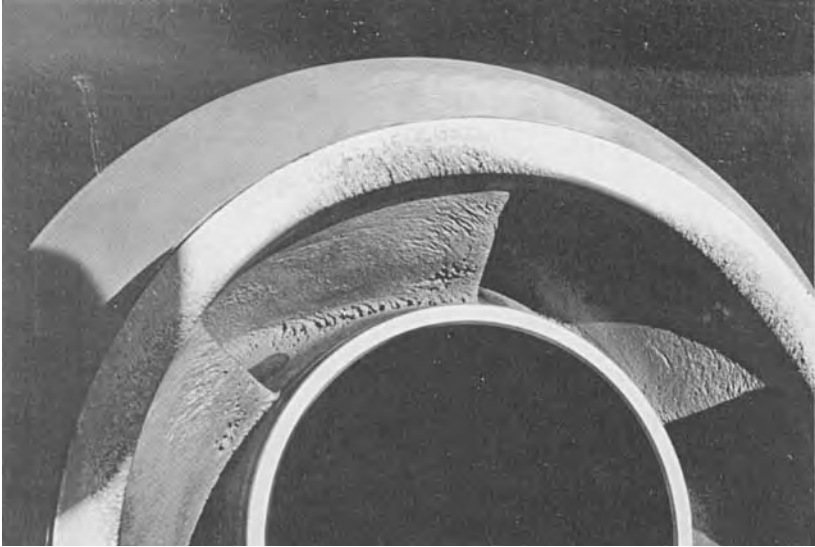


FIGURE 8 Cavitation erosion of an impeller, indicated by the porous appearance of cavitated regions on the surface

spongy or porous appearance of cavitation damage. An example of a cavitated impeller is shown in Figure 8.

It was conventional wisdom in the pump industry until recent years that the cavitation resistance of a material was directly related to its hardness. A more sophisticated understanding has been developed in recent years, leading to the development of a new class of nonstandard stainless steels with exceptional cavitation resistance.

The relationship between cavitation resistance and hardness was first critically investigated in the 1970s when it was observed that cobalt-based alloys of a modest hardness developed a very high resistance to cavitation damage. Cavitation resistance was related to the capability of the material to transform at the surface when subject to cavitation loading into a harder, more resistant metallurgical phase. This work was extended to austenitic stainless steels, whose chemical composition was adjusted to promote the formation of a stress-induced martensite under cavitation loading. New alloys were developed initially as weld filler metals to repair cavitation damage and later as impeller castings for pumps. These alloys have relatively low hardness in the solution-annealed condition, comparable to standard austenitic grades, but transform to a much harder martensite at the surface upon exposure to cavitation loading. The hard surface layer resists the initiation of fatigue cracks. If these cracks eventually develop after extended exposure to cavitation bubbles, propagation into the soft ductile base metal is difficult. Cavitation-resistant austenitic stainless steel castings, alloyed with chrome and manganese, develop cavitation resistance similar to that of cobalt-based alloys.

Extensive laboratory tests of the resistance of a wide range of materials to cavitation erosion have produced data for all the materials commonly used in centrifugal pump construction. It is possible to make a good correlation between the laboratory data and field experience to develop the following tabulation of the cavitation-resistance properties of pump materials, listed in order of decreasing cavitation resistance:

- Stellite
- Chrome-manganese austenitic stainless
- Carburized 12% chrome stainless casting
- Titanium 6AL-4V

- Cast nickel-aluminum bronze
- Cast duplex stainless steel
- Cast precipitation hardening stainless steel
- Ductile Ni-Resist
- Cast CF-8M
- Cast CA6-NM
- Cast CA-15
- Monel
- Manganese bronze
- Carbon steel (cast)
- Leaded bronze
- Cast iron

Selecting materials with adequate cavitation resistance will afford the pump designer much greater leeway in the range of conditions under which the pump can be operated. It also permits the design of smaller, lighter pumps that can be operated at higher speeds. The judicious use of materials significantly extends the time between outages caused by cavitation damage and can dramatically reduce maintenance costs.

TYPES OF WEAR

Rotating equipment, including pumps, can suffer from damage as a result of mechanisms unrelated to corrosion. The relative motion between parts that are in close proximity to each other can produce wear when these components come into contact with one another.

Catastrophic damage may occur if the parts make contact under high loading conditions or when foreign bodies are entrapped between the rotating and stationary components. An accelerated material loss or catastrophic seizure of these components can result in costly repairs or replacements. Erosion, due to the presence of solid particles in the liquid being pumped, can also limit the life of internal pump components.

Wear mechanisms have been categorized into more than 20 individual processes.¹ However, only a few mechanisms are frequently recognized as damaging to a pump:

- Adhesive wear: material-to-material contact
- Abrasive wear: solids interacting with internal components
- Erosion: solid particle impingement
- Fretting: small amplitude motion of parts causing oxidation damage

Identifying the wear mechanism is somewhat difficult at times as wear, or the loss of material, within a pump can result from more than one mechanism at a time.

The study of friction and wear as a science, known as *tribology*, had its beginning in the late 1930s. These early studies fostered an increased awareness of wear damage mechanisms that, in addition to corrosion and material fatigue, account for the life-limiting factors of pumps. Additional information on the study of wear can be found in current trade journals and texts.

Adhesive Wear One of the primary causes of material loss on rotating components in a pump handling clear liquids (with no solids entrained in the fluid stream) is adhesive wear. This material loss is due to metal-to-metal contact producing mechanical damage on the surface, a transfer of material, and eventually galling, which is considered a severe case of adhesive wear. Two important characteristics to consider for a pair of materials that may come into contact are their adhesive wear traits and their galling threshold.

The wear of two surfaces in relative motion is complex. Some alternative theories of sliding wear have been proposed in addition to the adhesive wear model. They are the delamination theory, the oxidation theory, the surface delamination theory, a fatigue model, and combinations of several of these theories. However, only the adhesive wear theory offers a general wear equation that quantitatively predicts wear, thus providing a means to rank materials with respect to their wear characteristics.

A multitude of adhesive wear tests exist, including ring and block, pin and vee block, 4-ball, and pin on disk. Wear tests are designed to simulate, as closely as possible, the actual service conditions and parameters, and are useful for developing a quantitative ranking of various alloys.

Wear testing and many years of field experience in pumping applications has led to several guidelines for the selection of materials for close clearance rotating and stationary components that will have adequate resistance to adhesive wear.

1. Austenitic stainless steels have poor galling resistance. Wide clearances are advisable, unless one of the two components is Nitronic 60, a special high manganese austenitic grade with antigalling characteristics.
2. Martensitic stainless steels will also have poor galling resistance. These are heat treatable materials. If heat treated to a minimum hardness difference of 50 BHN, galling resistance will be adequate. Also, if both parts are heat treated to a minimum hardness of 400 BHN, galling resistance will be acceptable.
3. Bronzes have reasonably good galling resistance, especially those containing several percent lead, which acts as a lubricant.
4. Cast iron also has good antigalling characteristics, with the graphite in the material promoting lubricity.
5. Several nonmetallics, including graphite composites and reinforced polyetheretherketone (PEEK) are excellent antigalling alloys, and useful for low specific gravity fluids.

Based upon extensive empirical testing and field experiences, several sound rules of thumb have been developed through the years when selecting pump wear ring materials. Three factors are used to select materials for wear surfaces in clear liquid environments:

- Corrosiveness of the fluid
- Amount of wear allowed
- Galling stress

Corrosion determines the class of material to be used. These classes generally fall into three groupings: noncorrosive, mildly corrosive, and corrosive. Of course, additional constraints occur when selecting an appropriate material within the corrosive material grouping that will need to be addressed by application experience.

A general guide for selection of suitable antigalling material combinations in several environments is as follows:

Environment	Materials	Hardness
Noncorrosive	Cast iron/leaded bronze	Unimportant
Mildly corrosive	Martensitic stainless steels (locally or through hardened)	Less than 400 BHN, 50-point differential
Corrosive	Corrosion-resistant, nongalling austenitic stainless steel (Nitronic 50/Nitronic 60 or Waukesha 88/Nitronic 50)	Greater than 400 BHN, same hardness acceptable
Severely corrosive	Highly alloyed austenitic stainless steel with hard-faced materials such as stellite or colmonoy	Not applicable

Using these industry-wide accepted rules of thumb will help avoid catastrophic damage normally resulting in costly repairs.

Some applications are particularly susceptible to adhesive wear problems and require special materials to resist wear. These include low specific gravity applications where the use of mechanical carbon materials is desirable because of the nonlubricating nature of these fluids. Common practice is to make the stationary component metal-filled graphite if the specific gravity is 0.5 or less. Stationary mechanical carbon components are also used in liquid CO₂ services and other potential dry start applications, such as the upper bearing in vertical pumps. Currently, nonmetallic wear components, such as advanced polymers and especially reinforced versions of PEEK are being employed to solve nagging problems in a variety of applications..

Fretting Fretting can be considered a special case of adhesive wear. It occurs when two parts in contact experience a repeated, small amplitude relative motion between close-fitting surfaces such as a loose impeller on a shaft. Researchers have described fretting damage as a four-stage event:²

1. Adhesive wear of the asperities on the mating materials
2. Abrasive wear caused by the wear debris produced in step one
3. Abraded particles filling the asperity valleys
4. Elastic contact producing cold working of the surface and micropitting

In a pump, there is the potential for small amplitude motion at loose fitting impellers, beneath loose bearings, and between impeller wear rings and the impeller hub. The design engineer does not intentionally create a circumstance that will generate this type of motion, but when it occurs, fretting damage can lead to other problems.

Fretting can be identified by a red powdery oxide that forms along the fretted surface. In a pump, the red-colored debris is often washed away, but a distinct damaged surface appearance will develop on the fretted surfaces. This damage is often described as having a mottled appearance and is best depicted as a flat, eroded surface with no directionality to the damage. Although the oxide may be washed from the surface, some staining of the adjacent component can be observed after disassembly of the pump. This has led to the misinterpretation that fretting is a corrosion mechanism, but it is actually a special wear phenomenon.

Figure 9 shows the fretting damage of a pump shaft along the impeller-fitted area where a loose fit enables the oscillation of the impeller during operation. Since the motion necessary to cause fretting can be of a small amplitude, large vibrations in the pump may not be present. This makes the detection of fretting during any operation impossible. The impeller in this example would have similar damage along its bore.

Fretting damage can be avoided with a few relatively simple guidelines. One should eliminate or prevent the possibility of motion between the two components by either tighter clearances, or shrink fitting the assembly, which increases the clamping force. If fretting is unavoidable in a particular design, methods of mitigation can be used. These include various coatings or providing the contact zone with an appropriate lubricant. Coatings that may be used include flame-sprayed high-nickel alloys, silver plating, or possibly adding a thin, dense chrome plating to one or both of the faces in contact.

Abrasive Wear Abrasive wear is often categorized into two main classifications: *two-body* and *three-body wear*. The name indicates the mechanism of wear. Usually, three-body abrasive wear is the primary mechanism of damage in centrifugal pumps. This can occur when hard solid particles entrained in the fluid enter between ring fit areas or impeller keyway faces. In fluids with high concentrations of solids, another form of three-body wear is produced. Solids carried in the fluid stream can strike the internal pump surfaces. This is more commonly referred to as *erosion*. This type of damage is observed in the impeller and cutwaters of the casing. The degree of material damage, due to this mechanism, depends upon the bulk hardness of the material, the carbon content, and the characteristics of the solids present. Important particle characteristics include size, shape, hardness, and mass.

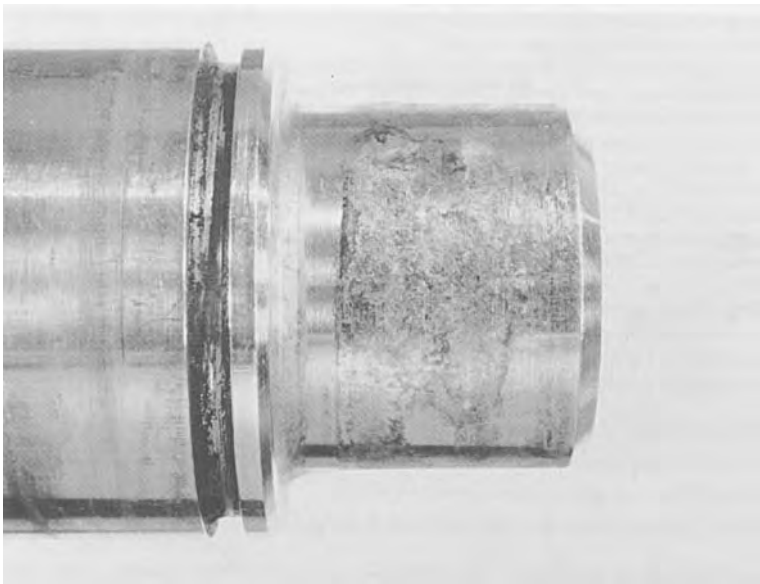


FIGURE 9 The fretting damage of a shaft beneath an impeller that experienced small amplitude motion. The mottled appearance is typical of the damage caused by fretting (2.2×).

To minimize three-body abrasive wear, a couple of variables must be taken into consideration. The wear ring clearance influences damage. The relationship between the size of the particles in the fluid stream and the gap into which they can enter is important. This is graphically illustrated in Figure 10, which shows three types of particle-to-gap relationships. Condition A is logically the most damaging three-body abrasive case. A high rate of damage will result as these particles are entrapped between the two components. In condition B,

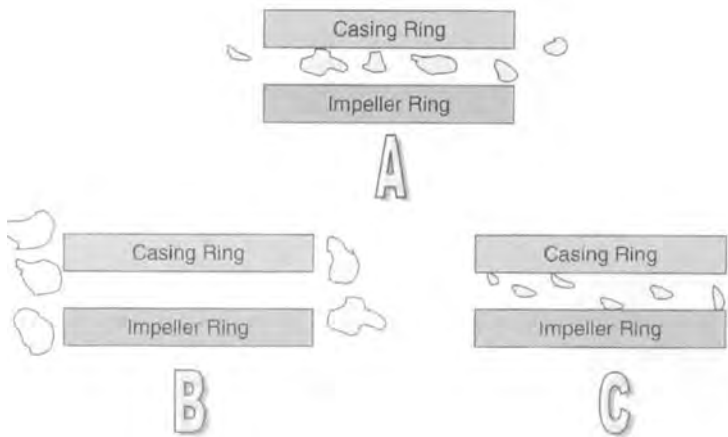


FIGURE 10 Three possible conditions between wear surface clearances and solid particle size. Condition “A” is conducive to maximum three-body abrasive wear.

large particles relative to the ring clearance will not enter and produce damage. This condition enables the particles to flow with the fluid stream through the eye of the impeller and exit the pump. In condition C, very fine or relatively small particles will not be entrapped and ground between the rings and will not result in collateral damage of the components.

Usually, particles in a fluid service will be in a range of sizes, so all the conditions will exist. Typically, a particle size and distribution analysis is performed to characterize the amount of particles that will cause condition A to exist. This is relatively simple to accomplish by extracting the solids from a fluid sample and performing a sieve analysis. The percentage of solids present in the fluid stream is extremely important for determining the appropriate material and design considerations. This will be addressed later with guidelines given for appropriate material selections.

The hardness of entrained solids is also extremely important. If particles are soft and friable, such as talc, little damage would be expected to occur on metal pump components because of three-body abrasive wear. The amount of damage is expected to be greater if the particles are extremely hard. These particles include welding scale or silicon dioxide (SiO_2), which is sand. The particle geometry also contributes to the amount of damage that can result in three-body abrasive wear. Often, particles of SiO_2 are found in a rounded condition. Pumps used to handle river water or seawater on ships frequently encounter these configurations. Hard, round particles are less damaging than particles of equal hardness with sharp, angular configurations. Fly ash, a very hard, sharp, angular particle, is one of the most abrasive services encountered in the pump industry.

A material's resistance to abrasive wear can be characterized by a standard ASTM test procedure. Each testing procedure attempts to simulate the mechanism that most appropriately addresses the class of abrasive wear. In general, materials that are resistant to two-body abrasive wear are resistant to three-body abrasive wear also.

Test results show that the primary property responsible for increasing resistance to abrasive wear is the hardness of a metal alloy. Small microstructural differences, alloying, and surface-condition differences within alloy groups also can influence the abrasion resistance of a material.

An example of three-body abrasive wear is shown in Figure 11. It shows a laser-hardened shaft sleeve after approximately one year of service in a mine dewatering operation where abrasive wear caused a significant wear of other material combinations. The abrasive wear was caused by fine tailings in this gold mine application. To increase the life of rings in services like this, the use of hardened wear rings is recommended. As an alternative to through hardening, pump producers will often use coated rings in applications where significant abrasive wear is anticipated. Depending upon the severity of the service, a coating containing carbides or other hard intermetallic compounds may be necessary.

For mildly abrasive services, the following materials should be considered:

- Ni-Resist—Its resistance is due to chromium carbides in the matrix. It has good adhesive wear resistance also.
- Selectively hardening the surface of AISI 420 (laser hardened 50 to 55 Rc). Surface hardening is not susceptible to hydrogen embrittlement or SCC.
- Carburized and hardened 12% chromium stainless steel.

For more abrasive services, the following is often considered:

- Hardened AISI-440C (50 to 55 Rc)
- Stellite or colmonoy-coated (hard-faced) austenitic stainless steel
- Solid stellite
- Tungsten carbide
- Silicon carbide
- Partially stabilized zirconia (PSZ)



FIGURE 11 The three-body abrasive wear of a laser-hardened shaft sleeve in an abrasive service. Note the fine concentric scoring of the hardened surface. The helix pattern is the laser-beam overlapped zone produced by the laser process.

Recent advances involving the use of ceramics, metal-matrix composite materials, laser-surface alloying, and laser-surface modifications to a substrate that normally could not survive in an abrasive service are examples of ongoing material developments.

Erosion Most fluids handled by pumps are considered clear liquids, meaning they do not have significant amounts of solid particulates present. The corrosive nature of these fluids dictates the required pump materials. Guidelines for many of these services are embodied in the “Corrosion Data Survey.”³

However, many fluid-handling applications requiring pumps are far from clear liquids. Solid particulates can be removed with costly filtration systems that must work flawlessly at all times. Fabricated piping systems may introduce suspended solids from weld slag and pipe burn. Naturally occurring suspended solids are those found in water sources such as river water or seawater, as mentioned previously in the abrasive wear section.

The following factors should be considered during the material and pump selection phase of the procurement process:

- The hardness of the particles
- The quantity of particles
- Size distribution
- Nature (geometry)
- The velocity of the pumpage
- The angle of fluid impingement

The first four items listed deal with the suspended solids. These variables can vary from application to application. The hardness of the particles is important to understand

in determining the materials necessary to yield an acceptable pump life span. Hardness can range from relatively soft substances, such as cellulose fiber in pulp and paper applications, to very hard abrasive particles such as silicon or rock in mining pumps. The Miller number index, as described in ASTM G75,⁴ is used to characterize the abrasivity of hard particles.

The Miller number was developed to determine the relative abrasivity and attrition of solid particles making up a slurry. In a closed loop test, the abrasivity of the particles becomes less damaging with time due to the fracturing and rounding (or friability) of the particles as they strike each other and/or impinge on a pump or casing wall.

The Miller number is therefore reported with two numbers. The first number characterizes the abrasivity of the particles and the second is the loss of abrasivity (attrition) of the particles during the slurry test. The abrasivity portion of the Miller number is useful in practical applications because this more closely characterizes a slurry's damaging potential. The attrition number has found little use other than characterizing a test loop's influence on a slurry. A slurry with a Miller number less than 50 is not considered abrasive in a reciprocating pump. Examples of slurries with a Miller number below 50 are limestone, sulfur, and detergent. It has been determined that a slurry consisting of finer particles is less abrasive than one containing larger particles. Test data show that Corundum at 220 mesh is about four times as abrasive as the same material at 400 mesh.⁴

Particle velocity plays a major role in the degree of damage that occurs in a pump handling slurries. In this case, the potential energy is converted into kinetic energy, producing a material loss by the transfer of energy from the particle to the component. The amount of material damage on an individual particle scale depends specifically upon particle velocity, v , and mass, m (kinetic energy, defined as mv^2). This is demonstrated by Finnie's equation⁵ for hard materials:

$$\text{Wear rate} = (\# \text{ of impinging particles}) \times (\text{average particle mass}) \times (\text{impingement velocity})^2 \times (\text{angle of impingement})$$

Of course, the pump part that absorbs the kinetic energy resulting from the particle impact has a role to play also. The material hardness and/or resilience of the pump component in absorbing the particle's impact energy will also determine the amount of material loss.

Chen and Hu⁶ have performed laboratory tests on materials while changing the particle variables previously described. Their test results show the following:

- An increased particle hardness increases the material loss up to 1700-kg/mm² microhardness (greater than 75 R_c). Beyond this hardness, a decrease in wear occurs. This is most likely the result of the hard, brittle particle fracturing, which absorbs some of the kinetic energy.
- Sharp, angular particles increase the erosion rate over round particles.
- Erosion increases with increasing concentrations of abrasive particles.
- An increased fluid (and particle) velocity increases the erosion rate.
- Minimal erosion occurs at an impingement angle of 0° (tangent to the target surface) and increases to a maximum amount of wear at a 65° angle.

A review of the literature shows that several authors have plotted the solid particle impingement angle versus the amount of erosion.^{7,8} These plots show that for ductile materials, erosion increases with the increasing impingement angle to a maximum material loss at an angle of 25°. Then the erosion damage decreases to the 65° impingement angle previously mentioned. Brittle materials, such as glass, are quite different. As the impingement angle increases from 0° to 90°, the volume of material loss continuously increases.

The characteristic features of erosion damage due to solid particle impingement are usually recognizable. However, when an aggressive fluid is present, the effects of solid particle impingement may not be easily identified. These effects can appear very much like corrosion-erosion, which is a fluid velocity-controlled damage mechanism where entrained solids are not present. If this damage is misdiagnosed, an improper material substitution can be made that may not solve the real problem. Conversely, a more likely situation is

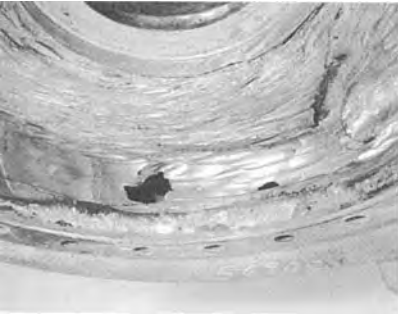


FIGURE 12 The severe erosion of a carbon steel casing in a 17% bauxite and sand service. Note the gouging due to the local turbulence of the slurry.

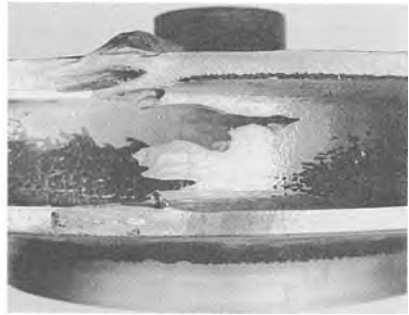


FIGURE 13 Erosion at the exit vane tips of a duplex, stainless steel CD4MCu impeller in a bauxite service

that the observed damage resulting from the erosion-corrosion is misinterpreted as solid particle erosion. A full understanding of the pumpage including the fluid velocity, fluid corrosiveness, content, and nature of the solid particles present is necessary for the appropriate action to be taken in improving the life of a damaged pump.

An example of solid particle erosion in a pump is shown in Figures 12 and 13. The severe erosion damage of a casing is illustrated by the gouging of surfaces that were directly impinged or scoured by glancing blows of the solid particles in the fluid stream. This pump handles a bauxite slurry where the percentage and velocity of alumina (Al_2O_3) and sand are too high for the carbon steel casing and CD4MCu impeller.

Figure 14 shows a CF3M impeller in a fly ash service. This shows that the greatest damage to the impeller is at the outer periphery, which corresponds to the highest velocity of the slurry. The least amount of damage is near the impeller inlet eye. Figure 14 also shows that the lower velocity region of the impeller inlet eye has the least damage. This confirms the laboratory data that show an increased erosion with an increased slurry velocity. Note that the damage increases near the outside of the inlet eye and is almost nonexistent at the impeller hub where the fluid velocities are lower.

Erosion damage can also be encountered in reciprocating pumps. Figure 15 shows extensive erosion of an AISI-type, 440C stainless steel ball from a ball valve after it became stuck and unable to rotate in a coal slurry application. This caused a slurry impingement on a concentrated region of the ball.

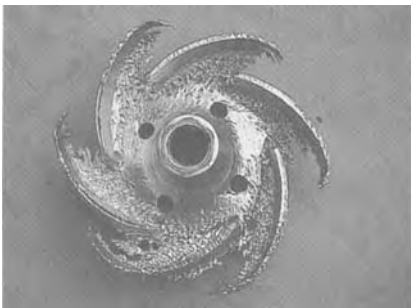


FIGURE 14 An austenitic stainless steel impeller in an abrasive fly ash service that shows severe erosion. Increased erosion occurs with an increasing fluid velocity near the periphery of the impeller.



FIGURE 15 Erosion damage of an AISI-type, 440C stainless steel ball valve in a coal slurry service

Erosion damage, once identified, has a limited number of solutions to prolong the longevity of pump materials. This can be accomplished by the selection of hard, wear-resistant replaceable liners, elastomeric liners, or, in cases where liners cannot be utilized, hard materials. Such metallic materials include white cast iron (such as Ni-Hard), high chromium (13 to 28%) alloy steels, cobalt-based super alloys (such as stellite), and nickel-based alloys. Slurry pumps tend to be large single stage pumps operated at low speed to minimize erosion damage. These pumps may use rubber liners to absorb the impingement energy of the abrasive particles, or be made of abrasion-resistant alloys.

FATIGUE

Centrifugal and reciprocating pumps are subjected to cyclical loading, which, if not considered during design, will result in a limited life due to material fatigue. In combination with a corrosive environment, material fatigue can be accelerated due to what is commonly referred to as *environmentally assisted fatigue*.

The one essential parameter in component fatigue is the presence of an alternating or cyclic load. In general, pumps are machines that have either fluid or mechanically induced cyclic loading on their components. Although centrifugal pumps are steady-state rotating equipment, pulsations or fluctuating applied stresses are encountered. The source of these cyclic stresses can be from fluid interaction between impeller exit vanes and diffuser vanes or, in a volute pump, the impeller vanes and the casing cutwater. Mechanically induced forces are due to bending moments acting on the pump shaft or possibly a component imbalance in the rotor assembly. Reciprocating pumps experience a cyclic loading of the internal and external components from the action of the machinery. In fact, these pumps can be thought of as large fatigue-testing machines due to the pulsating action of the pumping process.

When cyclic forces are applied to materials in a pump over a period of time, a crack may initiate, usually at a geometric stress concentration. After initiation, the crack will grow with continued cyclic loading until the part finally fractures. Fractures can occur, even though the loading produces stresses that are far less than the tensile strength of the material. Engineers have been aware of this potential mode of component fracture for many years and have developed design criteria that take this anomaly into account. The study of cyclic loading and material behavior based upon cyclic stress history and flaw size is beyond the scope of this text. It should be noted, however, that the field of fracture mechanics offers an engineering design tool that can predict the life of an engineered component.

Fatigue is a three-stage process consisting of: (1) crack initiation, sometimes associated with preexisting defects, (2) crack propagation, and (3) the final fracture, associated with crack instability, as suggested by Wohler.⁹ The applied stress level, sample geometry, flaw size, and mechanical properties determine the existence and extent of these stages.

Fatigue was first studied by August Wohler in 1852.¹⁰ Wohler's work included the concept of alternating applied stress, S , and the number of cycles, N , applied to a sample until a fracture occurs. This work is the basis for today's S/N curves used by design engineers. A laboratory-generated S/N curve is shown in Figure 16. This curve was generated by

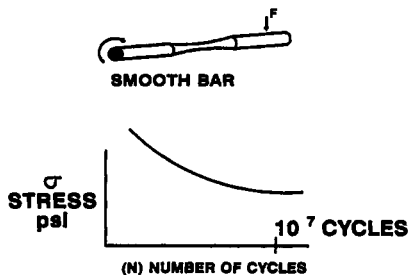


FIGURE 16 A laboratory-generated S/N curve for a smooth bar rotating beam test specimen

smooth, rotating-beam test specimens. These specimens are machined carefully to avoid metallurgical notches on their surfaces that would lower the applied stresses required to produce a failure during testing.

When the fluid handled is corrosive, pitting, or other types of corrosion develop and act as stress risers. Since corrosion over time can increasingly damage the material, the endurance limit will correspondingly decrease with the exposure time. No true fatigue limit exists for materials in a corrosive environment. For this reason, the corrosion-assisted fatigue life of a material is usually published with cautionary statements. Given enough time, corrosion can penetrate completely through a fatigue test specimen, resulting in a data point of zero load and zero cycles. For this reason, corrosion-influenced fatigue test results usually specify the corrosive media, the test temperature, the details of the sample preexposure to the corrosive media, and the test frequency with respect to the applied cyclic loading.

Published data vary because laboratories that use a low frequency of applied stresses increase the influence that corrosion has upon the test specimens. This is in comparison to laboratories that conduct these tests at a high frequency that minimizes the influence of the corrosive media. Published values for the ultimate tensile strength and corrosion fatigue strength of various alloys are shown in Table 2.¹¹

Wohler's investigation shows that a mechanical notch can reduce a material's fatigue.⁹ This is shown in Figure 17 as a shift in the *S/N* curve below the curve produced by a smooth bar specimen. The severity of the notch determines the amount of divergence from the smooth bar curve. As shown, the surface degradation mechanisms lower the stress needed to produce the specimen failure after a certain number of cycles. This in turn reflects a lowering of the material's endurance limit.

The three stages of fatigue cracking can be observed on the fracture face, rendering them easily identifiable. This is especially true if no other secondary damage masks the characteristic appearance. Especially in fractures that occur over long periods, lines are visible on the fracture surface. These bands are sometimes referred to as *clamshell markings*, *crack arrest lines*, or *beach marks* and reflect different periods of crack growth. Ratchet lines, which represent the joining of two different crack fronts on different planes into one, are observed in multiple origin fatigue cracks. Multiple origin fatigue fractures are often associated with rotating components.

TABLE 2 Corrosion fatigue strength of alloys in seawater*

Alloy	UTS	CFS
Ti-6Al-4V	154	88
Inconel 718	189	60
Inconel 625	149	50
Hastelloy C	108	32
Monel alloy K-500	176	26
Ni Al bronze (cast)	115	15
304 Stainless	79	15
316 Stainless	85	14
304L Stainless	75	14
316L Stainless	79	13
17-4PH - cast		10
70-30 Cu-Ni (cast)	83	9
Ni Mn Bronze	82	9
Mn Bronze	73	8
D-2 Ni-Resist		7.5
Mild steel		2

*Test parameters: ambient temperature, 1750 rpm, 2 to 3 ft/s (0.6 to 0.9 m/s). Corrosion fatigue strength (CFS) given at 100,000,000 cycles. All values are in ksi; 1 ksi = 6.894759 MPa. (UTS is ultimate tensile strength of material in air.)

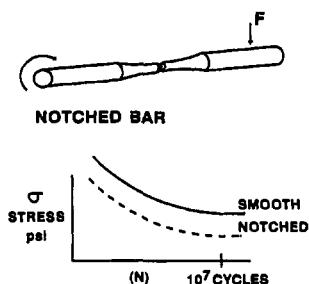


FIGURE 17 A shift in the S/N curve below the curve produced by a smooth bar specimen. The severity of the notch determines the amount of divergence from the smooth bar curve.



FIGURE 18 SEM photo of a fatigue smooth fatigue test specimen with a single origin. The arrow indicates the fatigue crack origin.

Figure 18 shows a high magnified view of a smooth bar fatigue specimen after a fracture. The arrow shows a single origin of this specimen. This fatigue crack propagated across the entire specimen diameter until the final fracture occurred. The final fracture is sometimes referred to as the *ductile overload zone* or the *fast fracture zone*. An example of a fatigue fracture on a pump shaft is shown in Figure 19. The arrows in this figure indicate the location of many crack origins. The flat, smooth surface appearance of this fracture face is characteristic of most fatigue fractures. This flat fracture appearance is sometimes mistaken for a brittle fracture because no evidence of plastic deformation is observed on or near the break. Shown almost directly in the center of the fractured shaft section is a small area of ductile overload, marked A and B, which corresponds to the final fracture area. The relatively small size of this area indicates the crack propagated under low alternating loads. In other words, the only material holding the two halves of the shaft together was the last area to fracture. This is a mere fraction of the total cross-sectional area of this shaft. The multiple arrows at the OD of the shaft shows the many

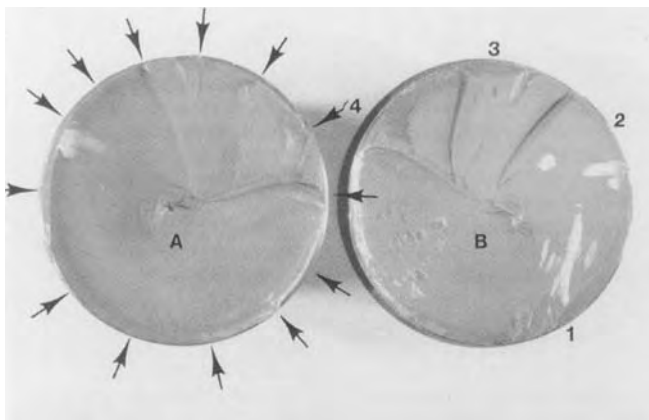


FIGURE 19 Multiple origin fatigue fracture of a pump shaft. The arrows show the locations of the many fatigue crack origins. A and B correspond to the final fracture zone of each fracture face.

crack origins. Fatigue ratchet marks are at each of these locations. This type of fatigue fracture is referred to as *multiple origin, high cycle fatigue*.

As mentioned before, an investigator uses the relative size of each fatigue crack stage to determine the magnitude of the loads acting on the component. The identification of the crack origin is also of prime concern in conducting a failure analysis. The crack origin is important to determine if the fatigue crack initiated from a flaw in the material or a notch produced in service or during manufacturing.

Corrosion, often the primary cause of pump material damage, can increase the likelihood of fatigue cracking. *Corrosion-assisted fatigue* is the name given to this special type of cracking. Corrosion damage can change the surface texture and significantly increase the local stresses acting on the pump component. If the corrosion damage is severe enough to produce a sharp notch in a region of high cyclic loading, then fatigue cracking of the component is inevitable. In some cases, the propagation phase is also influenced by oxidation, which can mask the telltale features of the fatigue mechanism. Corrosion oxides, which form along the crack face, can produce a wedging effect, which mechanically increases the local tensile forces acting on the crack tip. This increases the crack propagation rate.

An example of corrosion-assisted fatigue at two locations in the front shroud wall of an impeller is shown in Figures 20 and 21. The evidence of corrosion pitting on the surface indicates a strong possibility that corrosion influenced the fracture mode. Further investigation shows that both shroud wall fatigue fractures were initiated at corrosion pits located in highly stressed areas of the impeller. The fluid pulsations acting on the exit vane tip result in alternating loading.

Corrosion is not the only mechanism of surface degradation that can promote this form of cracking. Surface disruptions through fretting or wear can also provide sites for fatigue crack initiation. Sharp radii and defects at the material surface such as porosity can act as stress risers. Abusive grinding can alter the microstructure at the surface and induce shallow microscopic cracks, which can propagate by a fatigue mechanism.

Once the mechanism of fatigue cracking has been identified, suitable corrective actions can be implemented. These include the following:

- **Higher strength materials** A good approximation for the endurance limit of a metal is 50% of the material's tensile strength. This is for high-cycle fatigue where no macroplastic loading is experienced. A graph published in *Deformation & Fracture of Engineering Materials*⁹ shows this rule of thumb.
- **Design modification** The stress acting upon a component can be reduced with an increased section size. Reducing the stress on a component will increase its life. The design criteria for mean stress in an alternating loading environment can be determined



FIGURE 20 Overall view of a CF-3M impeller that has two corrosion-assisted fatigue fractures in the front shroud wall



FIGURE 21 A higher magnification of one of the fatigue fractures that originated at a corrosion pit at the exit vane tip and shroud intersection. Additional corrosion pitting can be seen on the impeller in this figure.

using several analytical models. Since components are subjected to a range of loading (not a constant amplitude), a fluctuating mean stress is encountered. The anticipated load history can aid in the design process to avoid fatigue fractures. The prediction of potential component life can be based upon a fluctuating mean stress design criterion, referred to as the *Pamgren-Miner cumulative damage law*.⁹

- **Surface treatments** The introduction of compressive stresses to the surface of a part increases the fatigue life of a component. This is usually performed at crack-sensitive regions such as sharp corners or notches. If compressive stresses are introduced into the surface of a material, cyclic tensile stresses in excess of the compressive stress value are needed to cancel their effect before fatigue damage can occur. Therefore, any form of compressive stress will benefit a component with respect to fatigue cracking. Compressive stresses can be introduced by (1) cold working, (2) shot peening, or (3) a local heat treatment that introduces beneficial, compressive residual stresses (such as laser hardening or induction hardening).
- **Increased corrosion-resistant materials** The use of more highly corrosion-resistant materials is beneficial in cases where corrosion has decreased a component's life by degradation of its surface condition.

MATERIALS OF CONSTRUCTION

Impellers The pump designer needs to consider several criteria when selecting the material for the impeller:

- Corrosion resistance
- Abrasive wear resistance
- Cavitation resistance
- Casting and machining properties
- Weldability (for repair)
- Cost

For many water and other noncorrosive services, bronze satisfies these criteria and, as a result, is the most widely used impeller material for these services. Bronze impellers should not be used for pumping temperatures in excess of 250°F (120°C). This is a limitation imposed primarily because of the differential rate of expansion between the bronze impeller and the steel shaft. Above 250°F (120°C), the differential rate of expansion between bronze and steel will produce an unacceptable clearance between the impeller and the shaft. The result will be a loose impeller on the shaft.

Leaded bronzes have been used extensively in the past as impellers, especially in less demanding applications. The lead addition to bronze enhances its castability and machinability. In recent years, environmental concerns associated with lead have caused many nonferrous foundries to stop producing these alloys and pump manufacturers are increasing their use of nonleaded bronzes for impeller applications.

It should be noted that bronzes have velocity limitations above which they will suffer accelerated erosion corrosion. The maximum velocity, which will correspond with the periphery of the impeller, is higher in freshwater than in saltwater. The most resistant bronzes, able to tolerate the highest velocities, are the nickel aluminum bronzes. These alloys are often used as impellers in saltwater applications because they combine high mechanical properties, good corrosion resistance, and the capability to be weld-repaired. A nickel aluminum bronze impeller can be designed for a higher speed than any other bronze impeller alloy.

Cast-iron impellers are used to a limited extent in small, low-cost pumps. Cast iron is inferior to bronze in corrosion, erosion, and cavitation resistance. It also cannot be welded to repair damage due to wear or erosion. For these reasons, a low initial cost is usually the only justification for selecting a cast-iron impeller.

Martensitic stainless steel impellers are widely used where bronze will not satisfy the requirements for corrosion, erosion, or cavitation resistance. The alloy most commonly used is CA-6NM. This alloy can be used for pumping temperatures above 250°F (120°C), as the thermal expansion is comparable to that of a steel shaft. Martensitic stainless steel impellers are used in a wide range of applications, including boiler feedwater, many cooling waters, and a variety of hydrocarbon applications. It does not have sufficient resistance to pitting corrosion for use in seawater.

Martensitic stainless steels are heat-treatable alloys. The specified mechanical properties are developed through a quench and temper heat treatment. Quenching can be in oil or, as is more common, in air. The cooling rate in air is sufficiently rapid that the high-temperature austenitic structure will transform to the metastable martensitic structure, which can subsequently be tempered to the desired hardness. The designer should specify that tempering be done at a minimum temperature of 1100°F (600°C) in order to assure that the casting has adequate toughness. It is also important that these alloys be heat-treated after weld repairs. This can present a problem in the case of a finish machined casting, which would suffer distortion if heat-treated. Welding techniques have been developed, however, that do not require a postweld heat treatment, but these are, in most cases, unsuitable for use on martensitic stainless impellers, particularly in higher stress areas near the periphery.

Oil and refining industry applications often involve exposure to hydrogen sulfide, which may be present as a trace contaminant in hydrocarbon fluids. Martensitic stainless steels are susceptible to a form of SCC in this environment and should be specified with a special double-temper heat treatment designed to limit hardness and thereby prevent cracking.

Austenitic stainless steels are used for impellers in applications requiring a higher level of corrosion resistance than can be obtained from the martensitic grades. A number of different alloys make up this group. The most widely used are CF-8M and CF-3M, which are the cast versions of the well-known 316 and 316L wrought materials. The cast alloys have a slightly different chemistry than the wrought grades. This difference accounts for the presence of 5 to 15% ferrite in the castings, which makes them slightly magnetic. The ferrite also enhances the resistance to SCC and hot shortness, a casting problem associated with fully austenitic cast grades. These alloys provide corrosion resistance over a wide range of pH and have reasonably good resistance to pitting and crevice corrosion in aqueous chlorides.

Higher alloyed austenitic cast grades are also available for applications requiring a greater degree of corrosion resistance. Alloy 20 contains about 30% nickel and was developed for sulfuric acid applications. The high nickel makes the alloy fully austenitic (without ferrite). Consequently, it is difficult to cast and suffers from hot shortness, which may manifest itself as fine cracking at the intersection between the vane and the shroud in an impeller. The high nickel content also makes Alloy 20 very resistant to SCC.

Austenitic grades containing 6% molybdenum have been developed for use in salt water and other high-chloride applications such as acidic brines used in oil field waterflood injection. The high level of molybdenum makes these alloys fully resistant to pitting in stagnant seawater, which will be present when a pump is not in operation. The 6% molybdenum grades are more expensive and therefore not frequently used for most applications. These alloys are usually considered only for critical, demanding applications where a high level of corrosion resistance is needed.

Austenitic stainless steels with unique properties have been developed for specific applications. A chrome-manganese alloy, discussed in the section on cavitation erosion, can be employed to mitigate or entirely eliminate cavitation damage in problem applications.

A high-strength austenitic stainless grade, CF10SMnN, can be used where the mechanical properties of CF-8M are inadequate. Some pump manufacturers also offer nitrogen-enriched austenitic grades that have corrosion resistance and mechanical properties better than CF-8M.

Duplex stainless steels offer a combination of higher mechanical properties and better corrosion resistance than the standard austenitic grades. The original duplex casting grade, CD4MCu, was developed in the 1950s. Use of this material was limited by problems with castability and weldability. Improved steelmaking technologies now enable the addition

of precise amounts of nitrogen to duplex stainless steel. The nitrogen addition improves castability, weldability, and also corrosion resistance. Numerous duplex stainless grades have been developed in recent years, all having a specified nitrogen addition. These duplex grades all outperform the old CD4MCu grade, which did not have a nitrogen addition. Many foundries now make CD4MCu with nitrogen.

Duplex stainless impellers are extensively used in mining, flue gas desulfurization, and similar applications that require a combination of resistance to corrosion and abrasion. Duplex stainless steels also have better corrosion resistance than the standard austenitic grades and are used in a variety of applications in the chemical industry, the pulp and paper industry, and the marine industry. Duplex stainless pumps are standard for offshore high-pressure water injection pumps in the oil industry. Published corrosion data indicate that, for acceptable resistance to seawater, a duplex stainless should contain a minimum of 25% chrome, 3% molybdenum, and 0.15% nitrogen.

Casings The following criteria should be considered when selecting material for centrifugal pump casings:

- Strength
- Corrosion resistance
- Abrasive wear resistance
- Casting and machining properties
- Weldability (for repair)
- Cost

For many pumping applications, cast iron is the preferred material for pump casings when evaluated on the basis of cost. For single-stage pumps, cast iron usually has sufficient strength for the pressures developed. For corrosive or hazardous petroleum products, it may be necessary to specify cast steel or cast stainless steel. The concern with cast iron when handling hazardous fluids is that the material is inherently brittle and could fail suddenly in a catastrophic manner with no prior indication of distress.

Cast-iron casings for multistage pumps are limited to approximately 1000 lb/in² (6.9 MPa) discharge pressure and 350°F (177°C). For temperatures above 350°F (177°C) and pressures up to 2000 lb/in² (13.8 MPa) discharge pressure, a cast steel is usually specified for split-case, multistage pumps. For pressures higher than 2000 lb/in² (13.8 MPa), a cast or forged steel barrel-type casing is usually required.

In any evaluation of cast-iron versus steel casings, consideration should be given to the problem of casing erosion during operation. Erosion can occur either from abrasive particles in the fluid or from wire drawing across the flange of a split-case pump. Although the initial cost of a steel casing is higher than that of a cast-iron casing, a steel casing can often be salvaged by welding the eroded portions and remachining. Salvaging a cast-iron casing by welding is much more difficult, and the casting usually must be replaced.

The ductile irons are useful casing materials for pressure and temperature ratings between cast irons and steels. Although the modulus of elasticity for the ductile irons is essentially the same as that for cast iron, the tensile strength of the former is approximately double that of the latter. In any evaluation of the ductile irons as a substitute for the steels in the intermediate pressure and temperature range, it should be remembered that ductile iron casings cannot be effectively repair-welded.

Austenitic irons, commonly known by the tradename *Ni-Resist*, are used for pump casings in applications where gray and ductile irons have insufficient corrosion resistance. Austenitic irons typically contain 15 to 20% nickel. They are frequently used in brackish and saltwater applications where they are considerably more resistant to both corrosion and erosion than unalloyed gray iron. A preferred combination for this service is a Ni-Resist casing and stainless internals. The stainless steel is galvanically protected from pitting when the pump is made idle by the more anodic Ni-Resist casing.

The traditional Ni-Resist alloys have poor weldability in common with other types of cast iron. In recent years, a new and more readily weldable Ni-Resist grade has been

developed. This grade, designated D2W, contains a small columbium addition that enhances weldability. This new D2W grade is gaining popularity as the preferred grade for pump casings.

Bronzes are also used for pump casings in many water applications. Several bronzes are used, with the choice depending upon the specific application. Leaded bronzes, specifically leaded red brass, are used for small low-pressure pumps. This material is the least costly and easiest to cast of the bronzes. Tin bronzes, with or without lead, are used for larger centrifugal pump casings. The lead contributes to the pressure tightness of the casting. Unleaded bronzes often have to be impregnated in order to obtain adequate pressure tightness. Unleaded tin bronze can be weld-repaired, whereas the leaded version is not weldable. Nickel aluminum bronze has the highest mechanical properties and the best corrosion resistance of the bronze alloys normally considered for pump casings. It can also be repaired by welding. Nickel aluminum bronze casings are expensive and usually not competitive on a cost basis with Ni-Resist or other alternatives.

Stainless steels are selected for pump casings when required due to corrosion considerations. Martensitic stainless steels are commonly used to handle boiler feedwaters as well as many hydrocarbon applications. These materials have good mechanical properties and are suited for high-pressure applications. Their corrosion resistance is less than that of other categories of stainless steel and for this reason, they are unsuited for more aggressive waters or other fluids.

The austenitic stainless steels, particularly CF-8M and CF-3M, are frequently used for pump casings in chemical applications and other corrosive services. These materials can handle a wide range of pH. They are resistant to erosions by high velocity and can be field-weld-repaired with relative ease.

Duplex stainless steels (stainless steels having a metallurgical structure that is approximately 50% ferrite and 50% austenite) are used for pump casings in some applications that require a combination of corrosion resistance and mechanical properties superior to that of the standard austenitic grades. These materials have become the preferred choice for high-pressure, offshore injection pumps handling seawater. The higher mechanical properties permit the design of thinner wall, lighter pumps. The weight savings is an important factor in this application.

Shafts The following criteria should be considered in the selection of material for a centrifugal pump shaft:

- Endurance limit
- Corrosion resistance
- Notch sensitivity

The endurance limit is the stress below which the shaft will withstand an infinite number of stress reversals without failure. Since one stress reversal occurs for each revolution of the shaft, this means that ideally the shaft will never fail if the maximum bending stress in the shaft is less than the endurance limit of the shaft material.

In practice, however, the endurance limit is substantially reduced because of corrosion and stress raisers, such as threads, keyways, and shoulders on the shaft. In selecting the shaft material, consideration must be given to the notch sensitivity of the material, and to its resistance to corrosion in the pumped fluid. Corrosion will substantially lower the fatigue limit of the material. Fatigue cracks can initiate at corrosion pits or other surface discontinuities that act as stress risers.

In the absence of corrosion, an approximate relationship exists between fatigue endurance limits and mechanical properties. The endurance limit is equal to roughly half the tensile strength of the material. Depending upon the application, the pump designer will usually select the least expensive shaft material that will satisfy the three criteria noted previously. Carbon steel is used when corrosion resistance is not required, and relative low mechanical properties can be tolerated. A low alloy steel, often AISI 4140, is used when the mechanical properties of carbon steel are not adequate. Martensitic stainless steels, usually type 410, are a common choice when some measure of corrosion resistance,

combined with reasonably good mechanical properties, is required. The resistance of type 410 stainless to fatigue crack initiation is related to the toughness of the material, which can vary over a wide range in commercial bar stock. The best measure of material toughness is the Charpy impact test, which is not a requirement of most relevant material specifications. In order to ensure the optimum toughness and resistance to fatigue cracking, type 410 stainless should be tempered at a temperature of 1100°F (593°C) minimum.

Stainless steels offering improved corrosion resistance and mechanical properties are also used in pump shafting. These include Nitronic 50, an austenitic grade, and 17-4PH, a precipitation hardening grade.

Several manufacturing issues involving pump shafts also need to be considered. Difficulty has frequently been experienced achieving the stringent tolerances for straightness required for long, thin shafts that are used in multistage pumps. This may necessitate intermediate stress relief during the machining process. The shaft may be stress relieved in the vertical position or, if horizontal, with supports every few feet. Type 410 pump shafting can be specially heat-treated to eliminate residual stress and maintain straightness tolerances during pump operation. Heat spot straightening of shafts is not recommended because it often adversely affects the microstructure and could lead to fatigue cracking.

Shafts can be plated or coated in specific areas for an improved resistance to wear or corrosion. Chrome plating is commonly used in this manner. The designer needs to be aware that plating reduces the fatigue endurance limit because of the fine microcracking associated with this process. An improvement in fatigue life can be achieved on plated surfaces if the substrate is shot peened. Normally, this reduction in endurance limits is not critical, because it is the flat surfaces, rather than the shoulders or keyways that are being plated. Consequently, the largest stress raisers are elsewhere in the shaft. There is also the potential for the plating process to cause hydrogen embrittlement cracks in high-strength steels. To avoid this problem, the steel should be baked at 300 to 400°F (150 to 200°C) after the plating process.

Finally, the manufacturing process needs to be carefully controlled to avoid the inadvertent introduction of stress raisers, which could shorten the life of the shaft. Abusive grinding has been identified as the root cause of some shaft fatigue failures. Heavy grinding will heat the surface and cause hardenable steels to form a thin layer of untempered martensite. This is a brittle structure and likely to develop fine cracks. These are stress raisers that can be propagated by a fatigue mechanism once the pump is placed in service. This type of problem is avoided by proper controls on the manufacturing process.

Wear Rings The following criteria should be considered in the selection of the material for the wear rings:

- Corrosion resistance
- Abrasive wear resistance
- Galling characteristics
- Casting and machining properties
- Suitability for coating

A centrifugal pump will often have both case and impeller wear rings. The impeller wear ring rotates within the bore of the stationary or case wear ring. These rings provide a close running clearance to minimize leakage from the discharge to the suction of the impeller. As the rings wear with use, leakage will gradually increase, affecting the head, capacity, and efficiency of the pump. In multistage (flexible rotor) pumps, increased wear ring clearance may also affect rotor stiffness.

To reduce the rate of wear of the wear rings, and thereby increase the life of the pump, special considerations must be given to the corrosion and abrasive wear characteristics of the ring material. Since the impeller and case rings may occasionally touch one another, the combination should also be selected to have antigalling characteristics.

Bronze is a widely used material for wear rings because it exhibits good corrosion resistance for a wide range of water services. In addition, bronze exhibits good wear characteristics in clear liquids but tends to wear rapidly when abrasive particles are present. The bronzes

also have a relatively good resistance to galling. The leaded bronzes offer excellent galling resistance but use of these grades has been reduced due to environmental concerns associated with lead. The casting and machining properties of most grades of bronze are excellent.

In applications where bronze is not suitable because of either corrosion or abrasive wear limitations, or where pumping temperatures exceed 250°F (120°C), stainless steel rings are used. Unlike bronze, the stainless steels of the 300 and 400 series have poor galling resistance. Several options are available to minimize the possibility of galling between stainless steel rings. The clearance between the rings can be increased, serrations can be machined into one of the rings, or a minimum hardness differential of 50 to 100 Brinell points can be established between the rings, if made from a hardenable grade of stainless, such as the martensitic grades. If both rings are hardened to above 400 BHN, it is not necessary to maintain a hardness differential.

Martensitic stainless steel rings are usually hardened in a furnace. A laser hardening process is also available that involves heating only the surface of the material, which is rapidly quenched by the base metal, resulting in a precisely controlled surface hardness and a soft ductile core. Rings hardened in this manner are more resistant to cracking in some environments and provide increased wear resistance because of increased surface hardness (50 to 55 R_c). A summary of adhesive wear test results can be found in Table 3.

Increasing the clearance between rings is the least costly method for reducing the risk of galling or seizures. However, increasing the clearance will reduce the output and efficiency of the pump. In large, low-head pumps, the loss in efficiency is less than 1%, but in small, high-head pumps, the loss in efficiency can be significant. Serrated rings can be used on smaller pumps to help maintain the efficiency level, but only at an increase in manufacturing costs.

For cast-iron, bronze, hardened 11 to 13% chromium steels, and materials with similar low-galling tendencies, the recommended minimum running clearances between rings are given in API Standard 610, "Centrifugal Pumps for Petroleum, Petrochemical and Natural Gas Industries." For materials with higher galling tendencies, such as austenitic stainless steels and for all materials operating at temperatures above 500°F (260°C), it is suggested that 0.005 in (0.125 mm) be added to the recommended minimum diametrical clearances.

TABLE 3 Calculated wear factor

Material (Ring/Block)	Hardness DPH (Ring/Block)	Wear Factor K
In Distilled Water Specific Gravity 1.0		
Leaded Bronze/ASTM A48 Class 30 CI	80/205	0.17×10^{-4}
Ni-Resist/Ni-Resist	120/120	0.41×10^{-4}
Nitronic 50/Nitronic 60	195/190	0.76×10^{-4}
90-10 CuNi/ASTM A48 Class 30	87/210	1.14×10^{-4}
Stellite 12/Stellite 6	440/395	1.71×10^{-4}
Ampco 18/Ampco 18	155/155	2.40×10^{-4}
AISI 410/ASTM A743-CA6NM	300/270	2.45×10^{-4}
AISI 410/AISI 416	290/430	2.97×10^{-4}
AISI 416/AISI 416	430/360	3.57×10^{-4}
In Alcohol Specific Gravity 0.87		
Nitronic 50/Nitronic 60	195/190	0.62×10^{-4}
Leaded Bronze/ASTM A48 Class 30	80/205	1.54×10^{-4}
AISI 410/AISI 416	290/430	7.38×10^{-4}
In Iso-octane Specific Gravity 0.69		
Leaded Bronze/ASTM A48 Class 30 CI	80/205	0.41×10^{-4}
Nitronic 50/Nitronic 60	195/190	0.69×10^{-4}

TABLE 4 Galling resistance of alloys

Metals in Contact			Threshold Galling Stress (ksi)	Threshold Galling Stress (mPa)
Silicon Bronze	BHN 200 vs. Silicon Bronze	BHN 200	4	27.6
Silicon Bronze	BHN 200 vs. Type 304	BHN 140	44	303
Waukesha 88	BHN 141 vs. Type 303	BHN 180	50+	345+
Waukesha 88	BHN 141 vs. Type 316	BHN 200	50+	345+
Waukesha 88	BHN 141 vs. S17400	BHN 405	50+	345+
Type 410	BHN 322 vs. Type 420	BHN 472	3	20.7
Type 416	BHN 342 vs. Type 316	BHN 372	13	89.6
Type 416	BHN 372 vs. Type 410	BHN 322	4	27.6
Type 440C	BHN 560 vs. Type 440C	BHN 604	11	75.8
S17400	BHN 311 vs. Type 304	BHN 140	2	13.8
S17400	BHN 435 vs. Type 304	BHN 140	2	13.8
Nitronic 50	BHN 205 vs. Nitronic 50	BHN 205	2	13.8
Nitronic 60	BHN 213 vs. S17400	BHN 313	50+	345+
Nitronic 60	BHN 205 vs. Nitronic 50	BHN 205	50+	345+
Nitronic 60	BHN 205 vs. Stellite 6B	BHN 415	50+	345+

Several antigalling material combinations have been used that do not compromise either wear or corrosion resistance. Nitronic 60 is a nonstandard austenitic grade that contains both manganese and nitrogen. This alloy has been developed for antigalling characteristics and will resist galling when mated with standard 300-series grades, Nitronic 50, and other alloys having poor galling resistance. Nitronic 60 is available in both wrought and cast forms. A cast nickel-based alloy, Waukesha 88, offers equally good antigalling characteristics and can be used in brine and other corrosive environments. A summary of galling thresholds for commonly used material combinations is shown in Tables 4 and 5.

Colmonoy- or stellite-coated rings are used in some critical applications and provide a high degree of resistance to abrasive wear, corrosion, and galling. These coatings are applied by welding, which can produce unacceptable distortion in rings, especially in larger diameters. The pump designer needs to exercise caution in specifying rings having a weld overlay because of potential manufacturing problems, which can involve cracking of the weld overlay or distortion of the ring. Some large rings for sewage pumps have performed well with tungsten carbide coatings applied using a high-velocity plasma spray process. The advantage of this process is that it does not heat the substrate, thereby avoiding distortion. Tungsten carbide is used for its high hardness and resistance to abrasion.

SELECTION OF MATERIALS OF CONSTRUCTION

The selection of materials for pumps is often not a simple and straightforward matter because of the weight assigned to various technical and economic factors by different users. A final selection may involve a compromise between the manufacturing cost and the anticipated maintenance costs. This is particularly true for fluids like seawater, for which a wide range of materials have been used, from cast-iron pumps with bronze internals to 6% molybdenum austenitic stainless steels. Despite the fact that economic factors unique to a particular application may influence the choice of materials for that application, some general guidelines can be used for some of the more common fluids.

Boiler Feedwater/Condensate High-purity water at a high temperature can be very corrosive to cast iron and carbon steel despite the fact that it usually has a low oxygen content. These materials will suffer erosion corrosion, sometimes within months, if incorrectly

TABLE 5 Gallling resistance of alloys

Metals in Contact				Threshold Gallling Stress (ksi)
Class 30 Cast Iron	(BHN 159–172)	vs. UNS C93200—Leaded Tin Bronze	(RB 41–43)	14.3
ASTM A487 CA6NM	(RC 29–31)	vs. Type 416	(BHN 352–415)	5.9
ASTM A487 CA6NM	(RC 29–31)	vs. Type 410	(BHN 353–415)	2.1
ASTM A487 CA6NM	(RC 29–31)	vs. Type 420—Laser Hardened	(RC 53–55)	1.7
Laser hardened Type 420	(RC 50–51)	vs. Type 420—Laser Hardened	(RC 53–55)	15.9
Type 420	(RC 50–55)	vs. Type 420	(RC 50–55)	21.1
Type 420	(RC 48–49)	vs. As-received nodular iron	(RC 24–26)	2.5
Type 420	(RC 48–49)	vs. Nitrided nodular iron	(RC 45)	10.9
Type 420	(RC 48–49)	vs. Laser glazed nodular iron	(RC 57–60)	11.3
Type 420	(RC 48–49)	vs. Laser hardened nodular iron	(RC 57–60)	6.8
Type 420	(RC 48–49)	vs. CA15	(RC 44–45)	1.8
Type 420	(RC 48–49)	vs. Zirconia	(R45N 74–79)	25*
Type 420	(RC 48–49)	vs. TDC on AISI 4140	(RC 70–80)	5.1
Type 420	(RC 48–49)	vs. Electroless nickel on nodular iron	(RC 45–49)	5.1
Type 420F	(BHN 262–302)	vs. Graphalloy bronze	(45–50 Schleroscope)	4.3
Type 410	(BHN 262–302)	vs. Type 420F	(RC 50–55)	3.8
Type 416	(BHN 262–302)	vs. Type 410	(BHN 353–415)	1.2
Nodular iron	(RC 27)	vs. Type 416	(BHN 352–415)	2.7
Nodular iron	(RC 27)	vs. Graphalloy bronze	(45–50 Schleroscope)	3.6
Nitrided nodular iron	(RC 45)	vs. Graphalloy nickel	(45–50 Schleroscope)	2.0
Nitrided nodular iron	(RC 45)	vs. Graphalloy bronze	(45–50 Schleroscope)	3.6
Laser glazed nodular iron	(RC 57–60)	vs. Graphalloy nickel	(45–50 Schleroscope)	3.6
Laser glazed nodular iron	(RC 57–60)	vs. Graphalloy bronze	(45–50 Schleroscope)	3.6
Laser hardened nodular iron	(RC 57–60)	vs. Graphalloy nickel	(45–50 Schleroscope)	5.0
Laser hardened nodular iron	(RC 57–60)	vs. Graphalloy bronze	(45–50 Schleroscope)	7.0
CA15	(RC 44–45)	vs. Graphalloy nickel	(45–50 Schleroscope)	2.0
CA15	(RC 44–45)	vs. Graphalloy bronze	(45–50 Schleroscope)	3.6
TDC on AISI 4140	(RC 70–80)	vs. Graphalloy nickel	(45–50 Schleroscope)	3.8

Electroless nickel on nodular iron	(RC 45-49)	vs. Graphalloy bronze	(45-50 Schleroscope)	2.5
Graphalloy bronze	(45-50 Schleroscope)	vs. Graphalloy bronze	(45-50 Schleroscope)	3.6**
TDC on AISI 4140	(RC 70-80)	vs. Graphalloy nickel	(45-50 Schleroscope)	5.0
Electroless nickel on nodular iron	(RC 45-49)	vs. Graphalloy nickel	(45-50 Schleroscope)	3.8
Graphalloy bronze	(45-50 Schleroscope)	vs. Graphalloy nickel	(45-50 Schleroscope)	3.6**
Nodular iron	(RC 24-26)	vs. Zirconia	(R45N 74-79)	12.7*
Nitrided nodular iron	(RC 45)	vs. Zirconia	(R45N 74-79)	14.0
Laser glazed nodular iron	(RC 57-60)	vs. Zirconia	(R45N 74-79)	25.5*
Laser hardened nodular iron	(RC 57-60)	vs. Zirconia	(R45N 74-79)	25.5*
CA15	(RC 44-45)	vs. Zirconia	(R45N 74-79)	15.3*
Nodular iron	(RC 24-26)	vs. TDC on AISI 4140	(RC 70-80)	1.8
Nitrided nodular iron	(RC 45)	vs. TDC on AISI 4140	(RC 70-80)	11.5
Laser glazed nodular iron	(RC 57-60)	vs. TDC on AISI 4140	(RC 70-80)	15.3***
Laser hardened nodular iron	(RC 57-60)	vs. TDC on AISI 4140	(RC 70-80)	20*
CA15	(RC 44-45)	vs. TDC on AISI 4140	(RC 70-80)	5.1
Nodular iron	(RC 24-26)	vs. Electroless nickel on nodular iron	(RC 45-49)	3.8
Nitrided nodular iron	(RC 45)	vs. Electroless nickel on nodular iron	(RC 45-49)	2.5
Laser glazed nodular iron	(RC 57-60)	vs. Electroless nickel on nodular iron	(RC 45-49)	5.1
Laser hardened nodular iron	(RC 57-60)	vs. Electroless nickel on nodular iron	(RC 45-49)	5.1
CA15	(RC 44-45)	vs. Electroless nickel on nodular iron	(RC 45-49)	5.1

* No galling at this stress. Testing is limited by torque capabilities.

** No severe damage

*** Inconsistent results

Note: All stress values in ksi (1 ksi = 6.894759 mPa)

applied. High-purity water is defined as that having a conductivity of 20 micromhos/cm or less, which is equal to a dissolved solids content of 10 ppm or less. At these low levels, carbon steel and cast iron are incapable of developing surface films such as magnetite that minimize erosion corrosion. The precise mechanism is not well defined, but the relationships between conductivity, oxygen content, pH, temperature, and the influence of these variables on erosion corrosion were established by pioneering work conducted at the Detroit Edison Power Plants in the 1950s.

The purity of feedwaters has increased over the years as water treatments have become more effective in removing dissolved solids to meet the demands of modern boilers. As a consequence, these waters have also become more corrosive. The early work at Detroit Edison showed that chromium additions to carbon steel dramatically enhance the resistance to erosion corrosion in high-purity water. Chrome additions of 1 to 1.25% are sufficient in most boiler feedwaters to impart acceptable corrosion resistance. Pump manufacturers initially used 5% chromium alloy steels for these applications. Difficulties in casting and welding 5% chromium steels led to the more recent use of 12 to 13% chrome steels, initially CA-15 and now CA-6NM, which is commonly used in boiler feed pumps, condensate pumps, and others that handle high-purity waters.

Water purity is directly related to conductivity, measured in units of micromhos, or microsiemens. It has been determined that, for high-temperature boiler feedwater services greater than 200°F (93°C), and with a conductivity of less than 20 micromhos, a chrome addition is usually required to avoid corrosion erosion at high velocity areas. Figure 22 shows a carbon steel shaft that was severely damaged in high-purity water.

Some applications still remain in which cast iron, bronze, and other less costly materials can be used. Materials are selected based on four critical water chemistry parameters: pH, temperature, conductivity, and dissolved oxygen. Table 6 represents an industry consensus on the recommended materials choices for the full range of high-purity waters.

Saline Water Saline waters have been defined as those that are sufficiently electrically conductive to enable an appropriate pump casing material to galvanically protect the pump internals when the pump is shut down. This corresponds to waters with more than about 1000 ppm chloride. Many pump applications involve saline waters. Among the more common saline water applications are the following:

- **Tidal river water** The chloride level here can fluctuate significantly with the season and the ingress of saltwater from a bay or estuary.
- **Groundwater** The chloride level and corrosivity can vary over a wide range. Some groundwaters, which are injected by high-pressure pumps into oil formations to enhance output, are very corrosive, due to low pH and very high chloride levels.



FIGURE 22 The corrosion erosion of a carbon steel shaft that was damaged in an ultra-pure water service after approximately 6 months

TABLE 6 Materials for boiler feedwater services

pH range	Feedwater Parameters				
	0-14	4.5-14	6-9	6-14	9-14
Temperature $\leq 200^{\circ}\text{F}$ (93°C)	S	G-S	B*-J* H-G-S	H-G-S	C*-L*-DR* H-G-S
Temperature $> 200^{\circ}\text{F}$ (93°C)	S	G-S			
Conductivity $< 20 \mu\text{mho/cm}$					
Dissolved oxygen $> 0.04 \text{ ppm}$					
Temperature $> 200^{\circ}\text{F}$ (93°C)	S	G-S		NM-G-S	
Conductivity $< 20 \mu\text{mho/cm}$					
Dissolved oxygen $< 0.04 \text{ ppm}$					
Temperature $> 200^{\circ}\text{F}$ (93°C)	S	G-S	B*-J* H-G-S	H-G-S	C*-L*-DR*
Conductivity $> 20 \mu\text{mho/cm}$					
Dissolved oxygen $< 0.04 \text{ ppm}$					

* Head must be less than 600 ft/stage (180 m/stage)

Materials:

B—Cast iron casing—bronze impeller—cast iron/bronze rings

C—All cast iron

DR—Carbon steel casing—cast iron impellers

G—All 12% chrome (CA-15 or CA-6NM)

H—Carbon steel case—12% chrome impellers (CA-15 or CA-6NM)

J—Cast iron case—bronze impellers—bronze rings

L—All cast iron with 12% chrome rings

NM—1- or 2-chrome barrels—12% chrome impeller (CA-15 or CA-6NM) S—All 316 stainless (CF-3M)

- **Geothermal water** This type may contain high levels of hydrogen sulfide, carbon dioxide, and other gases in addition to chlorides.
- **Oilfield brines** These are often deaerated, greatly reducing their corrosivity. Less corrosion-resistant pump materials may be used but are susceptible to corrosion during shutdown periods when oxygen cannot be effectively excluded from the water.
- **Seawater** The chemical composition of seawater is relatively uniform throughout the world. Other factors, including temperature, microbiological activity, and the presence of pollutants can alter the corrosivity of the seawater to pump materials of construction.

For each of the saline waters, a variety of materials has been used for pumps. The choice of materials for a particular application will depend on the water chemistry and other factors including the expected life of the pump, whether it will operate continuously or sit idle for long periods, and user preferences based on previous experiences. Some general considerations will influence material selections.

The materials for saline water pumps must resist erosion corrosion. Ni-Resist and copper-based alloys are frequently specified but have velocity limits, above which the protective oxide film is stripped off and accelerated corrosion occurs. Among copper-based alloys, nickel, aluminum, bronze can tolerate the highest velocity. The pump designer needs to be aware of these limitations and use bronze and Ni-Resist only for components when the velocity limits of the materials will not be exceeded.

Stainless steels develop a more tenacious oxide film than bronzes and can tolerate velocities much higher than those seen in pumps without suffering erosion corrosion. However, stainless steels are susceptible to pitting and crevice corrosion in stagnant seawater. These problems are exacerbated if marine biofouling occurs. Several methods exist for handling this problem. The stainless internals of a pump can be effectively protected by galvanic coupling with Ni-Resist. The combination of a Ni-Resist case and stainless steel internals is widely used because of this favorable galvanic relationship.

To avoid localized corrosion during shutdown in an all-stainless pump, some form of cathodic protection is required. This can be either sacrificial anodes or an impressed current system. It is also possible to construct the pump of stainless grades that are highly alloyed and develop adequate corrosion resistance. This approach requires either 6% molybdenum austenitic grades or 25% chromium, 3% molybdenum duplex grades. These materials are considerably more expensive than standard 300-series austenitic stainlesses are frequently used in critical applications. Higher alloyed duplex grades have become the universal standard for high-pressure injection pumps, especially those used in offshore locations. The high mechanical properties enable the design of lighter, smaller pumps. The weight saving is an important factor in offshore applications.

Many large seawater pumps are constructed of cast iron with bronze internals. Provided the velocities are not too high, this combination of materials has been known to provide approximately 20 years of service in some large vertical pumps. Nickel aluminum bronze is preferred over tin bronze for the impeller because it is stronger, has better resistance to high velocity, and is more easily weld-repaired.

Monel shafting is no longer commonly specified for seawater pumping applications. This material is expensive and will develop pitting in stagnant water. Several grades of stainless steel will provide a combination of strength and corrosion resistance equivalent to Monel at a significantly lower cost. These include Nitronic 50 and several of the higher alloyed duplex grades, such as Ferralium.

When specifying materials for seawater pumps, the designer should also consider whether the water will be chlorinated, and, if so, where the chlorine is to be added. Chlorine is added to cooling waters to kill marine organisms that cause biofouling. The chlorine may be added continuously at low levels or as a shock treatment at periodic intervals. Chlorination at normal levels of up to 2 ppm does not appear to be detrimental to alloys commonly used in saline water pumps. However, an injection should be made far enough upstream of the pump intake so that the dilution occurs ahead of the pump. When an injection is made at or near the pump intake, copper alloys, stainless steels, and Ni-Resist may suffer accelerated corrosion. Recent work has shown that the corrosion rate of stainless steels will begin to increase at chlorine levels of about 5 ppm.

Galvanic considerations will also play a role in the material selection for saline water pumps. In general, the pump internals should be cathodic to the pump case. Coatings should be avoided, especially on the anodic component. Flaws or defects in a coating will expose a small area of base metal. Corrosion will then proceed at a high rate due to the extremely unfavorable area ratio. It is also inadvisable to use carbon or graphite bearings in seawater pumps. These are at the noble end of the galvanic series and are likely to cause a galvanic corrosion of stainless steels or other alloys with which they come in contact.

Table 7 indicates a number of material combinations commonly specified for seawater pumps.

Hydrocarbons Pure hydrocarbons are not corrosive, but they frequently contain small amounts of water or other substances that make them corrosive. Material selection guidelines for a variety of hydrocarbon services have been developed by the American Petroleum Institute and are reproduced in Tables 8 and 9. These tables provide general guidelines that do not necessarily apply to every situation.

Most corrosion problems in hydrocarbon production and refining applications are caused by contaminants in the fluid. These can include water, hydrogen sulfide, hydrochloric acid, hydrofluoric acid, sulfuric acid, and caustic. Some of these contaminants are present in the feedstock. Others may be process chemicals, including solvents, neutralizers, and catalysts. There are generally considered to be two types of corrosion in refineries, including low-temperature corrosion, below about 500°F, in the presence of water, and high-temperature corrosion, above 500°F, and not requiring the presence of water.

Contaminants present in crude oil are the primary source of low-temperature corrosion problems. Water is almost always present in crude oil, and is responsible for several types of corrosion. Chloride salts are often present as well. These salts may be dissolved in water that is emulsified in the crude oil, or as suspended solids. It is also possible that salts can originate from brines that are injected for secondary recovery. In either case, water hydrolyzes certain inorganic chlorides to hydrogen chloride, which is corrosive to carbon steel.

TABLE 7 Materials for saline water pumps

	Component	Reference	Alternative	Nonferrous
Vertical pumps	Column/head	Ni-Resist ¹	316L	C614 ⁴
	Diffuser	Ni-Resist ¹	CF-8M/CF-3M ²	C952 ⁴
	Bowl	CF-8M/CF-3M ²	Ni-Resist ³	C952 (4)
	Inlet ball	Ni-Resist	Ni-Resist	C952 ⁴
	Impeller	CF-8M/CF-3M ²	CF-8M/CF-3M ²	C958 ⁴
Centrifugal pumps	Case	Ni-Resist	CF-3M/CF-8M ⁵	C952 ⁴
				G/M bronze
	Impeller	CF-3M/CF-8M ² CD4MCu ⁴	CF-3M/CF-8M ² CD4MCu ⁴	C958 ⁴ Monel
	Component	Deaerated Brine	Low pH Brine – H ₂ S	
High pressure multistage—oilfield brines	Case	Duplex ⁶	5-6% Mo alloy ⁷	
		C952 ⁴	CF-8M	
	Impeller	Duplex ⁶	5-6% Mo alloy ⁷	
		CF-8M C952 ⁴		

¹Furnace stress relieved²CF-8M should be postweld heat treated³Erosion-corrosion may be high⁴Postweld heat treat required⁵Galvanic protection desirable to prevent crevice corrosion⁶High-alloy grade with 25 Cr, 5-6 Ni, 3 Mo, and N⁷20 Cr, 19-25 Ni, 5-6 Mo, and N

The water associated with crude oil is often sour, meaning that it contains hydrogen sulfide, and possibly other contaminants, including ammonia, hydrogen cyanide, phenols, and chlorides. While sour water tends not to cause general corrosion of carbon steel or cast iron (because it contains no dissolved oxygen), hydrogen sulfide is capable of inducing a form of stress corrosion cracking in hardenable steels and other alloys. Additional precautions must be taken when hydrogen sulfide is present, consisting of special heat treatments to limit hardness. Details concerning the requirements for specific materials can be found in NACE standards MR0175 (applicable to oil and gas production environments) and MR0103 (petroleum refining environments). Pumps furnished to API Standard 610 (ISO 13709) must comply with these requirements whenever the fluid contains hydrogen sulfide.

A variety of process chemicals can occasionally cause low-temperature corrosion problems in refinery pumps. These include sulfuric acid and hydrofluoric acid, both associated with different alkylation processes. Carbon steel pumps handling hydrofluoric acid have experienced leakage on occasion. Hydrofluoric acid will slowly attack nonmetallic inclusions in a carbon steel casting, eventually producing a leakage path through the casting. Enhanced nondestructive examination, usually involving a helium leak test, is usually required to assure the pressure-tight integrity of these pump casings. Monel inlays are used in high velocity areas.

Pump internals in concentrated or spent sulfuric acid service are often made of alloy 20, or similar high alloys. Hydrocarbons containing only traces of concentrated or dilute sulfuric acid have been successfully handled by carbon steel pump casings, with an aluminum bronze weld overlay.

High-temperature corrosion in refinery pumps is usually associated with sulfur compounds found in crude oil. These sulfur compounds can include polysulfides, hydrogen sulfide, mercaptans, aliphatic sulfides, and disulfides. All of these react with metal surfaces at elevated temperatures, causing uniform thinning, localized attack, or erosion-corrosion.

TABLE 8 Material classes for centrifugal pump services (Courtesy of the American Petroleum Institute, reference 14)

CAUTION: This table is intended as a general guide for on-plot process plants and off-plot transfer and loading services. It should not be used without a knowledgeable review of the specific services involved.

Service	Temperature Range		Pressure Range	Materials Class	Ref. Note
	°C	°(F)			
Fresh water, condensate, cooling tower water	<100	<212	All	I-1 or I-2	
Boiling water and process water	<120	<250	All	I-1 or I-2	a
	120 to 175	250 to 350	All	S-5	a
	>175	>350	All	S-6, C-6	a
Boiler feedwater					
Axially split	>95	>200	All	C-6	
Double-casing (barrel)	>95	>200	All	S-6	
Boiler circulator	>95	>200	All	C-6	
Foul water, reflux drum water, water draw, and hydrocarbons containing these waters, including reflux streams	<175	<350	All	S-3 or S-6	b
	>175	>350	All	C-6	
Propane, butane, liquefied petroleum gas, ammonia, ethylene, low-temperature services (minimum metal temperature)	230	<450	All	S-1	
	> - 46	> - 50	All	S-1(LCB)	h
	> - 73	> - 100	All	S-1(LC2)	h
	> - 100	> - 150	All	S-1(LC3)	h, i
	> - 196	> - 320	All	A-7 or A-8	h, i
Diesel oil; gasoline; naphtha; kerosene; gas oils; light, medium and heavy lubricating oils; fuel oil; residuum; crude oil; asphalt; synthetic crude bottoms	<230	<450	All	S-1	
	230 to 370	450 to 700	All	S-6	b, c
	>370	>700	All	C-6	b

Noncorrosive hydrocarbons, e.g., catalytic reformat, isomaxate, desulfurized oils	230 to 370	450 to 700	All	S-4	c
Xylene, toluene, acetone, benzene, furfural, MEK, cumene	<230	<450	All	S-1	
Sodium carbonate	<175	<350	All	I-1	
Caustic (sodium hydroxide), concentration <20%	<100	<212	All	S-1	d
	>100	>200	All	—	e
Seawater	<95	<200	All	—	f
Sour water	<260	<470	All	D-1	
Produced water, formation water and brine	All	All	All	D-1 or D-2	f
Sulfur (liquid state)	All	All	All	S-1	
FCC slurry	<370	<700	All	C-6	
Potassium carbonate	<175	<350	All	C-6	
	<370	<700	All	A-8	
MEA, DEA, TEA stock solutions	<120	<250	All	S-1	
DEA, TEA-lean solutions	<120	<250	All	S-1 or S-8	d, g
MEA-lean solution (CO ₂ only)	80 to 150	175 to 300	All	S-9	d
MEA-lean solution (CO ₂ and H ₂ S)	80 to 150	175 to 300	All	S-8	d, g
MEA-, DEA-, TEA-, rich solutions	<80	175	All	S-1 or S-8	d

(Continued)

TABLE 8 Material classes for centrifugal pump services (Courtesy of the American Petroleum Institute, reference 14) (*Continued*)

CAUTION: This table is intended as a general guide for on-plot process plants and off-plot transfer and loading services. It should not be used without a knowledgeable review of the specific services involved.

	Service	Temperature Range °C	Pressure Range °(F)	Materials Class	Ref. Note
Sulfuric acid concentration >85%	<38	<100	All	S-1	b
85% to <1%	<230	<450	All	A-8	b
Hydrofluoric acid concentration >96%	<38	<100	All	S-9	b

The materials for pump parts for each material class are given in Table 9.

Specific materials recommendations should be obtained for services not clearly identified by the service descriptions listed in this table.

Cast-iron casings, where recommended for chemical services, are for nonhazardous locations only. Steel casings (5.12.1.6) should be used for pumps in services located near process plants or in any location where released vapor from a failure could create a hazardous situation or where pumps could be subjected to hydraulic shock, for example, in loading services.

*Oxygen content and buffering of water should be considered in material selection.

*The corrosiveness of foul waters, hydrocarbons over 230°C (450°F), acids, and acid sludges may vary widely. Material recommendations should be obtained for each service. The material class indicated above is satisfactory for many of these services, but shall be verified. S-8 materials may also be considered for operating temperatures below 95°C (200°F).

*If product corrosivity is low, Class S-4 materials may be used for services at 231°C to 370°C (451°F to 700°F). Specific material recommendations should be obtained in each instance.

*All welds shall be stress relieved.

*UNS N08007 or Ni-Cu alloy pump material should be used.

*For seawater, produced water, formation water, and brine services, the purchaser and the vendor should agree on the construction materials that best suit the intended use.

*The vendor shall consider the effects of differential material expansion between casing and rotor and confirm suitability if operating temperatures are to exceed 95°C (200°F).

*Materials for low-temperature services shall be selected with caution. Impact testing is recommended below -30°C (-20°F). Casting alloy ASTM A 352, Grades LCB, LC2 & LC3 is shown only for reference. Use equivalent materials for wrought alloys.

*Material alloys based on aluminium, bronze, aluminium bronze and nickel, may also be considered for temperatures as low as -196°C (-320°F).

TABLE 9 Materials for pump parts (courtesy of the American Petroleum Institute, reference 14)

Part	Full compliance materials ^b	Material classes and abbreviations													
		I-1 CI ^a CI	I-2 CI BRZ	S-1 STL CI	S-3 STL NI-RESIST	S-4 STL STL	S-5 STL STL 12% CHR	S-6 STL 12% CHR	S-8 ⁱ STL 316 AUS	S-9 ⁱ STL Ni-Cu alloy	C-6 12% CHR	A-7 AUS AUS ^{c,d}	A-8 316 AUS 316 AUS ^d	D-1j Duplex	D-2i Super duplex Super duplex
Pressure casing	Yes	Cast iron	Cast iron	Carbon steel	Carbon steel	Carbon steel	Carbon steel	Carbon steel	Carbon steel	Carbon steel	12% CHR	AUS	316 AUS	Duplex	Super duplex
Inner case parts: (bowls, diffusers, diaphragms)	No	Cast iron	Bronze	Cast iron	Ni-Resist	Cast iron	Carbon steel	12% CHR	316 AUS	Ni-Cu alloy	12% CHR	AUS	316 AUS	Duplex	Super duplex
Impeller	Yes	Cast iron	Bronze	Cast iron	Ni-Resist	Carbon steel	Carbon steel	12% CHR	316 AUS	Ni-Cu alloy	12% CHR	AUS	316 AUS	Duplex	Super duplex
Case wear rings ^k	No	Cast iron	Bronze	Cast iron	Ni-Resist	Cast iron	12% CHR hardened	12% CHR hardened	Hard-faced 316AUS ^e	Ni-Cu alloy	12% CHR hardened	Hard-faced AUS ^e	Hard-faced 316 AUS ^e	Hard-faced duplex ^e	Hard-faced super duplex ^e
Impeller wear rings ^k	No	Cast iron	Bronze	Cast iron	Ni-Resist	Cast iron	12% CHR hardened	12% CHR hardened	Hard-faced 316AUS ^e	Ni-Cu alloy	12% CHR hardened	Hard-faced AUS ^e	Hard-faced 316 AUS ^e	Hard-faced duplex ^e	Hard-faced super duplex ^e
Shaft ^d	Yes	Carbon steel	Carbon steel	Carbon steel	Carbon steel	Carbon steel	AISI 4140	AISI 4140 ^f	316 AUS	Ni-Cu alloy	12% CHR	AUS	316 AUS	Duplex	Super duplex
Throat bushings ^k	No	Cast iron	Bronze	Cast iron	Ni-Resist	Cast iron	12 % CHR hardened	12% CHR hardened	316 AUS	Ni-Cu alloy	12% CHR hardened	AUS	316 AUS	Duplex	Super duplex

(Continued)

TABLE 9 Materials for pump parts (courtesy of the American Petroleum Institute, reference 14) (Continued)

Part	Full compliance materials ^b	Material classes and abbreviations													
		I-1 CI ^a CI	I-2 CI BRZ	S-1 STL CI	S-3 STL NI-RESIST	S-4 STL STL	S-5 STL STL	S-6 STL 12% CHR	S-8 ¹ STL 316 AUS	S-9 ¹ STL Ni-Cu alloy	C-6 12% CHR 12% CHR	A-7 AUS AUS ^{c,d}	A-8 316 AUS 316 AUS ^d	D-1j Duplex Duplex	D-2j Super duplex Super duplex
Interstage sleeves ^a	No	Cast iron	Bronze	Cast iron	Ni-Resist	Cast iron	12% CHR hardened	12 % CHR hardened	Hard-faced 316AUS ^e	Ni-Cu alloy	12% CHR hardened	Hard-faced AUS ^e	Hard-faced 316 AUS ^e	Hard-faced duplex ^e	Hard-faced super duplex ^e
Interstage bushings ^a	No	Cast iron	Bronze	Cast iron	Ni-Resist	Cast iron	12% CHR hardened	12% CHR hardened	Hard-faced 316AUS ^e	Ni-Cu alloy	12% CHR hardened	Hard-faced AUS ^e	Hard-faced 316 AUS ^e	Hard-faced duplex ^e	Hard-faced super duplex ^e
Case and gland studs	Yes	Carbon steel	Carbon steel	AISI 4140 steel	AISI 4140 steel	AISI 4140 steel	AISI 4140 steel	AISI 4140 steel	AISI 4140 steel	Ni-Cu alloy hardened ^f	AISI 4140 steel	AISI 4140 steel	AISI 4140 steel	Duplex ⁱ	Super duplex ⁱ
Case gasket	No	AUS, spiral wound ^g	AUS, spiral wound ^g	AUS, spiral wound ^g	AUS, spiral wound ^g	AUS, spiral wound ^g	AUS, spiral wound ^g	AUS, spiral wound ^g	316 AUS spiral wound ^g	Ni-Cu alloy, spiral wound, PTFE filled ^g	AUS, spiral wound ^g	AUS, spiral wound ^g	316 AUS spiral wound ^g	Duplex SS spiral wound ^g	Duplex SS spiral wound ^g
Discharge head/suction can	Yes	Carbon steel	Carbon steel	Carbon steel	Carbon steel	Carbon steel	Carbon steel	Carbon steel	Carbon steel	Carbon steel	AUS	AUS	316 AUS	Duplex	Super duplex

Column / bowl shaft bushings	No	Nitrile butadiene ^a	Bronze	Filled carbon	Nitrile butadiene ^a	Filled carbon	Filled carbon	Filled carbon	Filled carbon	Filled carbon	Filled carbon	Filled carbon	Filled carbon	Filled carbon	Filled carbon
Wetted fasteners (bolts)	Yes	Carbon steel	Carbon steel	Carbon steel	Carbon steel	Carbon steel	316 AUS ^m	316 AUS ^m	316 AUS	Ni-Cu alloy	316 AUS ^m	316 AUS	316 AUS	Duplex	Super duplex

^aThe abbreviations in the upper part of the second row indicate the case material; the abbreviations in the lower part of the second row indicate trim material.

Abbreviations are as follows: BRZ = bronze, STL = steel, 12%, CHR = 12% chromium, AUS = austenitic stainless steel, CI = cast iron, 316 AUS = Type 316 austenitic stainless steel

^bSee 5.12.1.4

^cAustenitic stainless steels include ISO Types 683-13-10/19 (AISI Standard Types 302, 303, 304, 316, 321, and 347).

^dFor vertically suspended pumps with shafts exposed to liquid and running in bushings, the standard shaft material is 12% chrome, except for Classes S-9, A7, A-8, and D-1. The standard shaft material for cantilever pumps (Type VS5) is AISI 4140 where the service liquid allows (see Annex G, Table G.1).

^eUnless otherwise specified, the need for hard-facing and the specific hard-facing material for each application is determined by the vendor and described in the proposal. Alternatives to hard-facing may include opening running clearances (5.7.4) or the use of nongalling materials or nonmetallic materials, depending on the corrosiveness of the pumped liquid.

^fFor Class S-6, the standard shaft material for boiler feed service and for liquid temperatures above 175°C (350°F) is 12% chrome (see Annex G, Table G.1).

^gIf pumps with axially split casings are furnished, a sheet gasket suitable for the service is acceptable. Spiral-wound gaskets should contain a filler material suitable for the service. Gaskets other than spiral wound, may be proposed and furnished if proven suitable for service and specifically approved by the purchaser.

^hAlternative materials may be substituted for liquid temperatures greater than 45°C (110°F) or for other special services.

ⁱUnless otherwise specified, AISI 4140 steel may be used for nonwetted case and gland studs.

^jSome applications may require alloy grades higher than the Duplex materials given in Table H.2.

"Super Duplex" material grades with pitting resistance equivalency (PRE) values greater than 40 may be necessary.

PRE W 40, where PRE is based on actual chemical analysis.

PRE = %Cr_{free} + (3,3 × % molybdenum) + (2 × % copper) + (2 × % tungsten) + (16 × % nitrogen)

= [(% chromium - (14,5 × % carbon))] + (3,3 × % molybdenum) + (2 × % copper) + (2 × % tungsten) + (16 × % nitrogen)

Note that alternative materials such as "super austenitic" may also be considered.

^kNonmetallic wear part materials, proven to be compatible with the specified process fluid, may be proposed within the applicable limits shown in Table H.4. Also see 5.7.4 c).

^lThe vendor shall consider the effects of differential material expansion between casing and rotor and confirm suitability if operating temperatures are to exceed 95°C (200°F).

References in the notes above are to paragraphs within the API 610 standard.

This is generally referred to as sulfidic corrosion. Chromium additions to steel reduce the corrosion rate of steel at high temperature in crude oils containing significant amounts of sulfur. The relationship between corrosion rate, temperature, and chromium content is shown in the McConomy curves. The corrosion rate is reduced to a negligible amount when the chromium content is increased to 12%. Consequently, the 12 to 13% chrome martensitic grades find considerable use in high-temperature refinery applications.

Naphthenic acids are organic acids that are found in many crude oils. They are corrosive at temperatures above about 450°F (232°C), with the corrosion rate tripling with each 100°F rise in temperature. Unlike sulfidic corrosion, no protective scale is formed, and low alloy and stainless steels containing up to 12% chromium provide no benefit over carbon steel. Damage is usually in the form of pitting and other localized attack. Austenitic stainless steels containing molybdenum are resistant to naphthenic acid corrosion. Type 316 stainless steel is adequate for applications involving naphthenic acid at lower temperature. Austenitic grades containing at least 3.5% molybdenum, such as 317, are required for more severe applications.

It should also be noted that there are several environments that are capable of inducing stress corrosion cracking in pump materials in refinery environments. In addition to aqueous chlorides, and hydrogen sulfide, which have been discussed previously, aqueous amines, and polythionic acid can cause cracking. Amines are used to remove hydrogen sulfide and carbon dioxide from refinery streams. Cracking has been found at welds in carbon steel components. It is thought to be similar to caustic stress corrosion cracking. Postweld heat treatment of carbon steel welds has been found to be an effective method of preventing cracking, and should be required for all fabrication welds, and repair welds, in carbon steel pumps that will handle amines.

Polythionic acids are formed by the reaction of oxygen and water with the iron/chromium scale that covers the surface of austenitic stainless steel components as a result of high-temperature sulfidic corrosion. Since oxygen and water are not present during normal operation in situations where austenitic stainless steels would be used, conditions suitable for polythionic acid to form occur only during shutdowns. Procedures to prevent cracking, including nitrogen purging, have been developed, and this form of cracking, which affects only sensitized materials, is relatively uncommon.

MATERIALS FOR SLURRY AND ABRASIVE SERVICES

The construction materials for pumps that handle high concentrations of suspended solids are often based upon high bulk hardness. In many applications, coatings, hard liners, and weld overlays are used to specifically increase the surface hardness of the internal wetted portions within the pump. However, many of the slurry applications use nonmetallics that do not have high bulk hardness because of their unique qualities.

Nonmetallics Contrary to “harder is better,” a good number of slurry pumps use non-metallic materials, such as rubber, that absorb the kinetic energy of the solid particles through a large elastic deformation of the surface. Natural rubber is the most commonly used material since it provides good wear resistance with abrasive particles less than approximately 0.25 to 0.38 in (6 to 9 mm). Rubber linings pose a problem in the bonding of this outer shell to a metallic substrate. This is particularly true for cutwater areas of a casing and, of course, attachment to metal skeletons of an impeller. Care must be taken also in considering the liquid phase of the slurry and the temperature of the application, both of which can degrade the rubber.

Other factors to consider when using elastomeric liners include pressure in the flow passage versus the pressure between the liner and the wall, and temperature, since rubber softens around 240°F (115°C). It is also important to consider the size of the particles greater than 0.2 in (5 mm) if they are dull or greater than 0.08 in (2 mm) if they are sharp, and the head per stage.

Metals In mildly abrasive services, carburized steels are sometimes used to increase the wear life of components. Carbon is diffused into the surface of carbon steel, which, after

a hardening heat treatment, can achieve a surface hardness of 60 R_c . This gas diffusion heat treatment can produce high hardness layers that penetrate the outside surface of the pump component to a depth of approximately 0.080 to 0.090 in (2.0 to 2.3 mm). However, after carburization, the materials are impossible to weld-repair without cracking. Using a special process, usually a vacuum furnace, carburizing has been employed in the surface hardening of the 12% chromium stainless steels, such as CA15, for abrasive services where mild corrosion is expected.

The most commonly used materials for severe slurry services are the abrasion-resistant cast irons found in ASTM A532. Essentially, three main classes and several types of alloys are covered in this specification. The most widely employed material in slurry applications is the Class III hard irons. A brief description of this class of abrasion-resistant iron is as follows.

All three classes contain a martensitic matrix with secondary hard phases of chrome and iron carbides that increase the wear resistance. The molybdenum in Class II increases the material's hardenability for thicker cross sections.

It should be noted that machining and welding these three classes of material is impossible. Another important consideration is the role of carbon content on corrosion, erosion, and fracture resistance. High carbon contents reduce corrosion resistance because any chromium tied up as chrome carbide is no longer available to form a protective chrome oxide layer. Although beneficial with regard to erosion and abrasion resistance, high carbon content increases the susceptibility to breakage by thermal and mechanical shocks.

To counteract this problem, a number of precautionary measures must be adopted to enhance the serviceability of this class of material. First, slow warm-up cycles must be instituted, typically around 100 to 150°F per hour.¹² Another strategy is to lower the hardness from about 600 to 400 Brinell by a partial anneal. This measure reduces brittleness, but at the expense of erosion resistance.

Linings, Inserts, and Coatings The low ductility and toughness of A532 cast irons do not permit their usage with primary pressure boundaries according to ASME code and API regulations. This restricts the use of hard irons to internal wetted parts. Therefore, it is necessary to use steel pressure casings with hard materials as liners. A common slurry pump consists of ASTM A532, Class III, Type A (HC-250) impellers and replaceable HC-250 wear liners for the volute and for both the inlet and outlet ends of the pump casing.

Since pump erosion is often quite localized, in some instances it is more practical to install replaceable, mechanically attached inserts at high wear areas, such as the cutwater. These are typically made of sintered tungsten carbide or some other hard material. Newer materials, such as ceramic composites and toughened ceramics, should perform better than the "cermets" used in the past.

Several problems, however, can occur with mechanically attached inserts. One problem is protecting the fastening device against erosive wear. Another problem is the insert's tendency to act as a turbulence riser due to an imperfect fit or erosion-induced crevices and offsets. Limited success has been achieved with weld-applied overlays of stellite and other hard-facing materials. Weld overlays are extensively utilized, but several problems may occur. These include a propensity for cracking, debonding resulting from preferential corrosion of the bond line, dilution of the hard-facing material with the substrate, and potential uneven thickness after machining.

ASTM A532 Grade	Composition (Cr, Ni, and Mo)	Hardness (BHN)
Class I: Type A (Ni-Hard)	1 to 11% Cr, 3 to 7% Ni	500 to 600
Class II: Type A, B, C, D, E	11 to 23% Cr, 0.5–3.5% Mo	450 (annealed for machining) 600 (hardened)
Class III: Type A (26% chrome iron, original trade name of HC-250)	23 to 28% Cr	400 to 600 depending on desired properties

Thermal spray coatings as well as diffusion surface treatments have been used in pump applications for fluids containing high concentrations of suspended solids. Spray coatings are restricted to areas within the pump, accessible by the line of sight. Diffusion-produced coatings are not limited by this constraint. A disadvantage of diffusion processes is that they are performed at high temperatures that can negatively influence the base material properties. Diffusion coatings can range from traditional gas carburizing to the diffusion of high chromium alloys. These coatings increase the surface hardness of the component and, depending upon the process, can increase the material's surface hardness to values in excess of 60 *R_c*. Diffusion layers can be produced to a depth of approximately 0.100 in (2.54 mm). One item of caution: these coatings usually render the material unweldable after application. For this reason, steps must be taken to protect areas of anticipated welding, such as attachment piping. Future weld repairs are not possible unless the coating is completely worn off or removed.

Through the years, developments in thermal spray equipment have enhanced the acceptability of this surface modification process. Thermal spray processes employ the transfer of a material onto another by raising the temperature of the hard-facing material, usually in powder form, and projecting it against the component that requires the additional erosion resistance. The bond strength between the hard-faced material and the substrate material is directly influenced by the maximum velocity that the particles of molten material achieve in a given thermal spray process. The greatest bond strength is achieved by the highest velocity process. The typical thermally sprayed materials used in pumps to resist solid particle erosion damage are as follows:

- Nickel chromium boride coatings
- Cobalt-based hard-facing coatings
- Tungsten carbide coatings
- Solid particle tungsten carbide loaded (1) or (2)

The processes used to apply the above hard-facing materials are as follows:¹³

Process	Typical Particle Velocity
Flame spray	100 ft/s (30.5 m/s)
Plasma spray	800 ft/s (244 m/s)
D-Gun (Union Carbide trade name for detonation gun process)	800 ft/s (244 m/s)
HVOF (high velocity oxy-fuel)	3000 ft/s (915 m/s)

The severity of the service usually dictates the process. In the past, thermal spray coatings, for the most part, were tungsten carbide and the diffusion coatings were high in borides. It was found that for spray coatings an increased performance could be achieved by applying them over erosion-resistant substrates. This is a challenge because the high-chromium, carbon, abrasion-resistant materials are thermal crack sensitive. Overlay coatings, if applied several times, are thicker but are more prone to cracking, chipping, and spalling. An alternative is to use carburized carbon steel or 12% chromium stainless steel centrifugal pump components for mildly corrosive environments. Coatings are frequently used to increase the life of plungers in reciprocating pumps for slurry services.

Another process for applying hard-faced materials is by laser consolidation. This process can be accomplished in two different ways. The first case is one in which a laser beam is used to melt a thermally sprayed coating applied upon a substrate. The other process is to simultaneously melt the substrate while applying a hard-faced material. In either case, the principle is to use the hard-faced material as a consumable in a laser-welding operation. Since laser welding is a rapid process, very little dilution of the hard-facing material is produced. This allows for much thinner coatings that are less prone to thermally induced cracking during an operation. In addition, since there is little dilution, the hardness and chemistry of the coating are very consistent. This provides for uniform erosion resistance throughout the entire coating thickness.

SUMMARY

It should be clear from the discussion of materials for saline waters and hydrocarbon applications that the selection of materials for a pump is a complicated exercise, requiring knowledge of the engineering properties of the material, its fabrication characteristics, and corrosion and erosion resistances. An in-depth discussion of the considerations governing material selections for other pump applications, such as the chemical industry, mining industry, and others, is beyond the scope of this section, but a number of references are included to provide additional guidance. Past experience is often helpful, but it is recommended that the materials selections be reviewed by a metallurgist or corrosion engineer with experience in this area, especially for critical applications or those applications with which you are not familiar.

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SECTION 8.2

MATERIALS OF CONSTRUCTION FOR NONMETALLIC (COMPOSITE) PUMPS

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The use of composite materials is playing an increasing role in the manufacture of industrial pumps due to the cost savings they offer in manufacturing, installation, and operation. Composite parts can be molded to near "net shape." This eliminates the cost of secondary machining to attain the final part. Because one composite material can replace two or three different grades of metal part, the combining of several parts into one assembly is possible. This part's integration reduces assembly time, reduces inventory, and ultimately reduces manufacturing costs.

Because composite pumps weigh less than metal pumps, they are easier to handle during installation and maintenance. Further, because composites naturally dampen vibration, a composite pump operates more quietly than a comparable metallic pump.

The corrosion resistance of composites is superior to metals. For this reason, a composite pump's life can be significantly greater than a metallic pump. When the amortization costs of both composite and metallic pumps are calculated, the composite pump has the cost advantage.

FACTORS TO CONSIDER

Factors to consider in using composite materials for pump parts are mechanical properties, abrasion and corrosion resistance, temperature, costs, weight and insulating properties, exposure to sun, fire resistance, and chemical resistance.

Mechanical Properties The composite pump or parts must be designed to withstand the loads/stresses of the operating conditions. If the existing metal part design were

simply replicated in a composite, the composite part could fail. Excessive stresses for a composite pump at the nozzle area or at the bearing housing can cause the pump housing to creep. The result is part distortion or bending and shaft misalignment.

A solution to this problem is to calculate the stress levels in the composite part and design within the composite's elastic limit. These complex stress calculations are now done on a routine basis with the aid of a computer and nonlinear FEA (finite element analysis) programs.

Abrasion and Corrosion Resistance The abrasive wear on 316 stainless steel is caused by the continuing removal of the metal's protective oxidation layer on the metal's surface. Composites do not need a protective layer and therefore have better resistance to mild abrasives. However, some composites are subject to corrosion by caustic liquids.

As background, metallic materials deteriorate from any one or combination of the following electrochemical reactions: galvanic action, pitting, corrosive attack, crevice corrosion, intergranular corrosion, and stress corrosion. The temperature, pH, formation, removal of an oxide corrosion barrier, or the velocity of the solution affects the rate of corrosion in a metal's surface.

Temperature Most composite pump materials are limited to operating temperatures below 300°F (150°C) for noncorrosive liquids and under 250°F (120°C) for corrosive liquids. Like metals, there is a reduction of mechanical properties for composites with increasing temperature.

Costs There are three areas of cost to consider when evaluating a composite part: development costs, initial capital costs, and cost savings.

Developing costs would include the following:

- Designing the part for composites
- Evaluating and testing alternative composite materials
- Prototype molding the parts

Initial capital costs are in the cost of the molds to produce the parts. Cost savings in manufacture and use include the following:

- Little or no corrosion increases the time between maintenance and extends the useful life of the pump.
- Reduced secondary machining cost at manufacture (the parts are near "net shape").
- Integrating several parts into one assembly results in a less expensive part, lower inventories, and less assembly time.
- Quality control cost is reduced as part tolerance is primarily determined by the mold.

The initial cost of a composite pump versus a metallic pump is typically as follows:

- Equal or slightly more expensive than an all 316 stainless steel pump
- 80% the cost of an Alloy 20 pump
- 50% the cost of a Hastelloy C pump

Weight and Insulation Properties With a significant reduction in weight compared to metal, the composite pump is easier to handle during installation and maintenance removal. As an example, an immersion sump pump that is usually 3 to 10 ft long weighs only 10 to 30% of a comparable metal pump.

Added insulation on the pump to conserve high temperature process heat is usually not needed on a composite pump, as the composite is a natural insulator when compared to metal. This saves the cost to fabricate an insulation cover over the pump and provides easier access for maintenance.

Exposure to the Sun Some composites are susceptible to the UV (ultraviolet) rays of the sun. Two percent carbon black in the resin is an effective UV blocker. Hindered amine light stabilizers (HALS) can also be added to the composite to effectively prevent damage by the UV rays of the sun.

Fire Resistance In a severe fire, thermoplastic composites will melt and burn and thermoset composites will char and ash. If the composite reaches 400 to 500°F (204 to 260°C), there will be distortion of the pump and the pump will be lost. Because of the insulating properties of a composite pump housing, however, the liquids inside the pump will be cooler than if the pump were metal.

Fire retardant additives can be used with composite materials, but they may affect optimum mechanical properties. Most thermoset composites are self-extinguishing in a fire, but the pump manufacturer should be consulted to determine if there is a potential for toxic fumes if the composite is burned.

CHEMICAL RESISTANCE OF COMPOSITES

Composites do not corrode because they are nonconductive. Therefore, an electrochemical reaction does not take place. (An exception to this may be when electrically conductive long carbon fibers are used in the wrapping of a pressure vessel or a pump housing.) Their lack of resistance to different chemical fluids can degrade composite materials. This degrading can be in the resin, the reinforcing fiber, or in the coupling agent between the resin and fiber. If a composite material is going to be degraded, it will usually be immediate. Fluid temperature and concentrations are vital when selecting the appropriate composite material. With some composites, such as nylon, the pH can be a factor when selecting the best composite material for the application.

As an initial test, a sample coupon of the composite can be immersed in the process liquid and the results noted. A more definitive method is to review the composite manufacturer's informational database on chemical resistance. Many composite manufacturers have run extensive chemical resistance studies of their materials in a variety of liquids at set temperatures over time. The composite's physical properties are measured before and after the test period and the percentage deterioration versus time is documented in the manufacturer's literature.

The Hydraulic Institute, discussed later, has a polymer material selection guide that recommends specific composites for various liquids.

TYPES OF COMPOSITES

Composites can be divided into two general groups of organic compounds: thermoplastic and thermoset.

Thermoplastic Thermoplastic materials can be repeatedly melted and solidified. They do not go through an irreversible chemical process when heated. Amorphous thermoplastics resins have a broad softening range and are suitable for processes that require good melting strength like thermoforming. They can be transparent in color but do have limited chemical resistance.

Crystalline thermoplastic resins have a sharp melt point and are better suited to molding processes like injection molding where melt strength is not required. They have excellent chemical resistance. Their physical properties can be modified and improved with the addition of glass or carbon fiber, tougheners, or mica. Chart 1 shows the effect of various reinforcement materials on selected polymers.

Some of the thermoplastic processes that are used to form pump parts are injection molding, vacuum forming, extrusion, and blow molding. Thermoplastic resins suitable for pump applications include fluoropolymers, acrylics, polyethylene, polypropylene, polyvinyl chloride, and others. The allowable temperature range for these materials is from 130 to 300°F (54 to 150°C), depending on the material. Figure 1 shows the elastic modulus versus service temperature for various polymers.

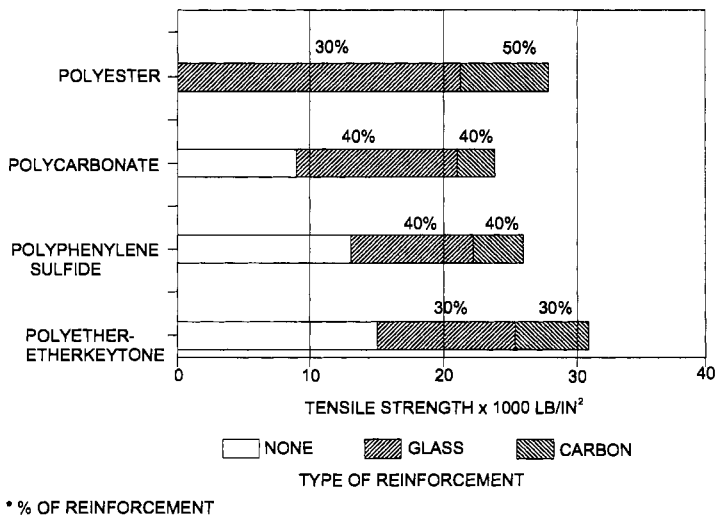


CHART 1 Thermoplastics strength versus reinforcement

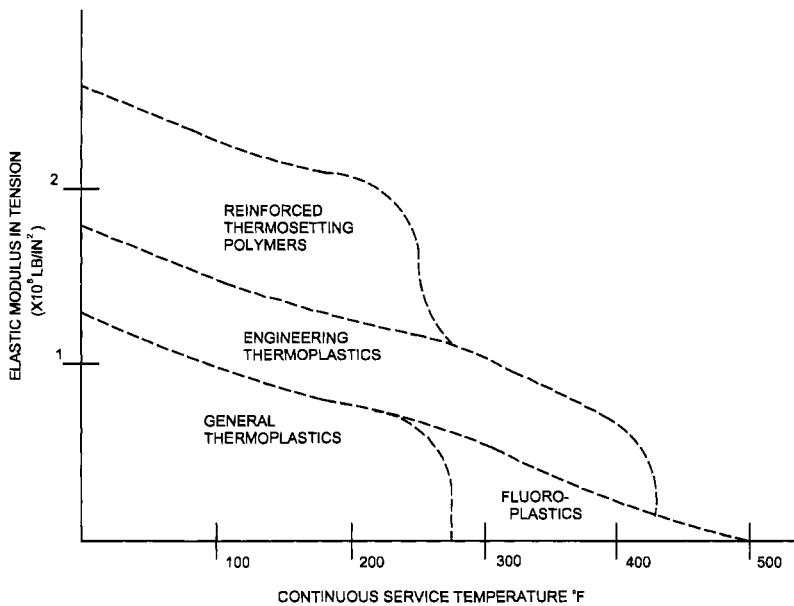


FIGURE 1 The effect of temperature on modules of elasticity

TABLE 1 Differences between thermoset and thermoplastic

Process	Thermoset	Thermoplastic
Average range of molded thickness	0.030 to 2.00 in (0.75 to 50 mm)	0.06 to 0.50 in (1.5 to 13 mm)
Weight range of piece	1 to 500 lb (0.5 to 225 kg)	0.5 to 5.0 lb (0.22 to 2.26 kg)
Glass content by volume	50 to 60%	30 to 40%
Length of glass fiber	0.25 in (6 mm) to several inches (mm)	0.06 in (1.5 mm)
Strength	Not uniform throughout	Uniform
Minimum annual quantities	Less than 1000	10,000
Obtain strength from ribs	Not necessarily	Yes
Tooling	Depends on complexity and size	Generally 20 to 30% higher than compression molding
Process	Compression or resin transfer	Injection, cannot use compression molding

Thermoset Thermoset materials, when heated, go through an irreversible chemical change with the result that the material will not soften when reheated. Thermoset resins are generally reinforced with glass fiber, linen, carbon fiber, graphite, minerals, and so on.

They have an allowable application temperature range of 180 to 250°F (82 to 120°C), depending on the material. Examples of thermosets are epoxies and vinyl esters. They can be stronger than thermoplastics but are usually more brittle. Thermosets are processed by compression molding or transfer molding. They have good chemical resistance to acids, bases, and oxidizing agents.

Table 1 outlines the differences between thermoset and thermoplastic polymers.

LINED PUMPS

To provide chemical resistance for a metallic pump, it is possible to line inside the pump housing. Some of the more popular liner approaches are flexible, rigid, sprayed, or molded. With a flexible-lined pump, a metal casing is made to the design shape and pressure. Then a loose liner is fitted inside the pump housing and bonded to the metal casing with an adhesive. This type of liner can come loose under vacuum conditions. There also can be problems at the termination of the liner to the mating pipe flanges. Depending on the liner material and design of the pump, liners can be molded into a pump housing. Two of the more popular methods are rotomolding, in which the liner material becomes bonded to the metallic armor, and injection molding, in which the liner is locked to the armor by strategically placed locking grooves.

A resin used in some lined pumps is PTFE (polytetrafluoroethylene). By itself, PTFE does not have the physical properties needed for a pump housing. When it is supported by a metal housing, however, it provides an excellent lining material due to its outstanding range of chemical resistance at high temperatures.

Rigid pump liners are machined out of plastic stock shapes or molded into a near net shape. The liner is then mechanically compressed between bolted steel plates to provide the support. Such pumps are usually small in size.

A sprayed-on composite lining is made by taking a metal casing of the required pressure rating, heating it, and then spraying the composite material inside the housing and flanges. At 0.030 in (0.76 mm) thickness, the lining will not have pinholes and is suitable for pump applications. This lining process is usually limited to pumps that require up to 100 horsepower (75 kW) drives.

Table 2 gives an overview of the comparative advantages and disadvantages of the different styles of pumps.

TABLE 2 Comparative advantages and disadvantages of the different styles of pumps

Feature	Thermoset	Thermo-plastic	Flex-lined	Sprayed-lined	Rigid-lined
Cut impellers available	Yes	Yes	Yes & No	Yes	Yes
Potential for damage to liner in operation or repair	Low	Low	High	High	Low
Integral casing flanges	Yes	Yes	Yes	Yes	No
Efficiency	High	Med	Med	Yes	Med/Low
CPI hydraulic coverage	Yes	No	Yes	Yes	No
Conforms to ASME/ANSI B73.1M dimensions	Yes	No	Yes	Yes	No
Cost	Med	Low	Yes	Yes	Med/High
Pressure capability	Med	Low	High	High	High
Vacuum capability	Good	Fair	Low	Good	Good
Temperature capability	Med	Med/Low	High	High	High

TABLE 3 Mechanical properties of some unreinforced composites

Property/Material	Steel	Glass	Vinyl Ester	PVC	PVDF	PTFE	PPS	PP	PEEK	Epoxy
Modules of elasticity 1×10^6 lb/in ²	29	5	3.5	0.4	0.16	0.8	0.6	0.2	0.6	3.5
Tensile strength 1×10^3 lb/in ²	60	300	11	7	7	4	14	6	16	16
Specific gravity	7.8	2.6	1.2	1.3	1.8	2.2	1.3	0.9	1.31	2.2
Coefficient of thermal expansion 1×10^5 in/in/°F	0.6	0.3	2.0	2.8	8.5	10.0	2.8	8.0	4.0	2.2

Lb/in² $\times$ 6.894757 = kPaIn/in/°F $\times$ 1.8 = cm/cm/°C

CERAMIC AND CARBON PUMPS

Solid ceramic or carbon pumps are used for very hostile liquids such as concentrated hydrofluoric acid. Ceramic and carbon pumps are much more expensive than nonmetallic pump materials, but do have the chemical resistance required for extremely aggressive chemicals.

TECHNIQUES FOR MOLDING PUMP COMPONENTS

Injection molding is a process for high volume forming of high quality thermoplastic parts. The part cost is low, but the initial mold costs can be high. Reinforcing materials can be added to an injection molded material. Typical additives are short-strand fibers (0.06 to 0.50 in or 1.5 to 13 mm long) and are glass, graphite, mica, and so on.

Compression or transfer molding uses thermoset resins and can make small quantities of larger parts. The mold costs are less, but the part price is usually more due to trimming and secondary machining steps. The reinforcing materials for these processes can be continuous or chopped mat strands, as well as woven mats and blankets. These long-strand fibers give exceptional physical properties to the final part.

MECHANICAL PROPERTIES OF COMPOSITES

Composites in general have far lower physical properties when compared to steel and lower physical properties than aluminum or brass. Composites can, however, be designed to withstand the operating pressures of a pump by increasing the section modulus of the part; that is, increasing the wall thickness. Table 3 shows some unreinforced composite properties versus steel. Figure 2 illustrates where the various composites and metallic are positioned and applied on a grid of developed pressure versus flow.

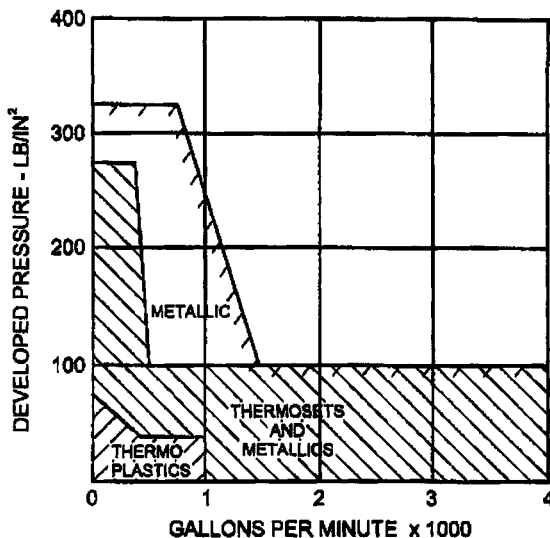


FIGURE 2 Developed pressure versus flow for composite pumps ($\text{lb/in}^2 \times 0.0689 = \text{bar}$; $\text{gpm} \times 0.227 = \text{m}^3/\text{h}$)

TABLE 4 Thermoplastics used for pump parts

Chemical Name	Common Reference	Temperature Limit	
		°F	°C
Polycarbonate	Lexon®	250	120
Phenylene oxide	Noryl®	194	90
Polyphenylene	Ryton®-PPS	250	120
Polyphenylene	PP	150–180	65–82
Chlorinated polyvinyl chloride	CPVC	230	110
Polyvinylidene chloride	PVDC	160	70
Polyvinyl chloride	PVC	140	60
Polyetherether keytone	PEEK	250	120
Polytetrafluoroethylene	Teflon®-PTFE	460	238
Chlorotrifluoroethylene	Teflon®	500	260
Polyvinylidene fluoride	Kynar®-PVDF	300	150

Thermoplastics PVC and CPVC pumps are low cost and suitable for the lower temperature ranges. PVC can be used up to 140°F (60°C) and CPVC, with greater abrasion resistance, is suitable for temperatures up to 210°F (100°C). PVC and CPVC can be solvent-bonded together to attain the best attributes of each resin in a single pump.

Polypropylene (PP) has excellent corrosion resistance and is used in hydrocarbon service up to 185°F (85°C). It is not suitable for strong acids or chlorinated hydrocarbons. The material can be ultrasonic, vibration, and spin welded in assembly.

PVDF (Kynar) is very corrosion and abrasive resistant and suitable for temperatures up to 300°F (150°C). It is also relatively expensive.

PTFE (Teflon) is extremely corrosion resistant, but its low physical properties require it to have a metal reinforcement backing to withstand high pump pressures. Table 4 is a partial list of thermoplastics used for pump parts.

Thermosets Vinyl esters and epoxies are the strongest of the thermoset composites and usually do not require external metal reinforcement. Vinyl esters are resistant to corrosion, whereas the epoxy composites are resistant to solvents. Table 5 is a list of thermosets used for pump parts.

CANDIDATE FLUIDS FOR COMPOSITE PUMPS

This section discusses some of the more general liquids that are suitable for use in composite pumps.

Chlorine and Caustics Chlorine is used in the production of organic chemicals such as vinyl chloride, chlorinated solvents, pesticides, and fluorocarbons. Chlorine is also used in

TABLE 5 Thermostats used for pump parts

Name	Common Reference	Temperature	
		°F	°C
Vinyl ester GL	VE-GI	250	120
Vinyl ester C	VE-C	250	120
Epoxy	Epoxy	250	120

GL—glass reinforced (should not be applied to hydrochloric acid or caustics)

C—carbon reinforced

the pulp and paper industry, municipal water purification, sewage treatment plants, and in the electrolysis of sodium chloride to produce sodium hydroxide. Additional fluids in this category are as follows:

- Potassium hydroxide
- Sodium chloride solutions
- Seawater, brine
- Sulfuric acid diluted up to 70%
- Hydrochloric acid
- Sodium hydrochloride
- Nitric acid up to 5%

Figure 3 shows that a composite pump can handle up to a 40% concentration of hydrochloric acid at 175°F (80°C).

Sulfuric Acid Figure 4 shows the operating range of a composite pump, pumping sulfuric acid.

Ferrous and Ferric Chloride Ferric chloride is used as an etching reagent in the production of printed circuit boards. It is also used as a coagulant in wastewater treatment. In metal pickling operations, ferric is produced when hydrochloric acid reacts with iron and steel. In all of these applications, composite pumps have replaced the more costly titanium pumps.

Ethylene and Propylene Candidate applications for composites of ethylene and propylene are:

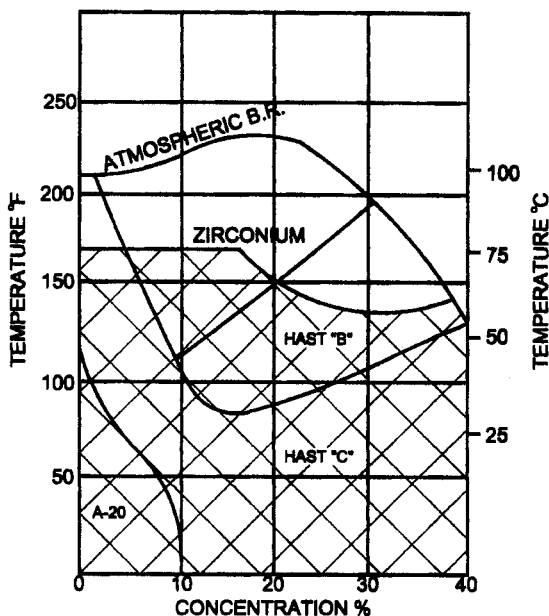


FIGURE 3 Material selection for hydrochloric acid

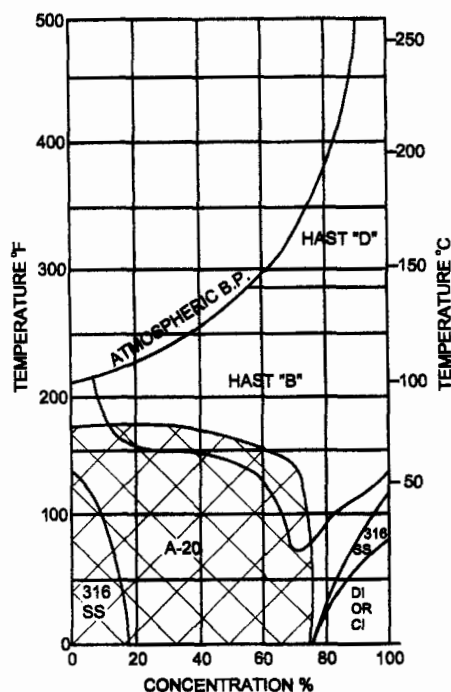


FIGURE 4 Material selection for sulfuric acid

- Propylene glycol
- Ethylene glycol
- Diethylene glycol
- Propylene chlorhydrin

Dyes Acid, sulfur, and Diazo dyes can be used in composite pumps.

Agriculture Chemicals Pesticides comprised of insecticides, herbicides, fungicides, and fertilizers can be severely corrosive to a composite pump. The chemical supplier should be contacted for their recommendations of materials suitable for use with their products.

Table 6 lists the maximum temperature and concentration of several liquids for various polymers.

INDUSTRIES USING COMPOSITE PUMPS

Pulp and Paper Composite pumps are used in bleaching and cooling of liquor preparation. The pumps can handle white, black, and green liquors. Other liquids suitable for composite pumps associated with this industry include:

- Sulfuric acid
- Sodium chlorate

- Sodium hypochlorite
- Sulfur dioxide
- Chlorine
- Methanol
- Peroxide bleach
- Sodium chloride
- Chlorine dioxide
- Sodium hydroxide
- Hydrochloric acid

Metal Finishing Composite pumps are an excellent choice for use in this industry for two reasons:

- The chemicals used in electroplating, steel pickling, etching, anodizing, galvanizing, and plating do not chemically attack composites.
- Metal pumps can generate stray currents and could effect the plating process.

Composite pumps are natural electrical insulators, which makes them well suited for this industry.

Desalinization and Water Purification Composites can replace 316 stainless steel, duplex stainless steel, and Alloy 20 on sea water distillation plants. On reverse osmosis equipment, composite pumps can be used for backwash, membrane blowdown, and intake screen wash.

Aqua Culture Composite pumps can be used for waste removal, seawater transfer, washdown, and filtering without adversely affecting sea life. This adverse reaction can occur with metal pumps. Metal pumps can have a chemical reaction with seawater and stray electrical currents generated by the pump can affect the pump's mechanical seal.

STANDARDS FOR COMPOSITE PUMPS

There are many composite pump designs for low-pressure and low-capacity use in the general-purpose market. For composite pumps in the "industrial chemical market," however, a separate standard ASME B 73.5M has been written. This standard is similar to B73.1M for metal pumps in that it requires the same dimensional interchangeability, shaft deflection, and seal chamber requirements. The composite pump standard differs from the metallic pump standard in these areas:

- Basic working pressure is from 100 to 275 lb/in² (6.9 to 19 bar) depending on pump size.
- Nozzle flanges are Class 150 dimensions but not Class 150 rated.
- Hydrostatic pressure factor above working pressure will depend on size, rpm, and manufacturing process.
- Pressure-temperature limit will be based on a manufacturer-user agreement for the liquid and its concentration.
- The standard applies to both thermoplastic and thermoset composites.
- Casing, casing cover, and gland have a minimum corrosion allowance of two years.

Hydraulic Institute Material Selection Guide A material selection guide has been written with recommendations of what composite should be used with what type of

liquid. This guide lists 150 liquids at various concentrations and temperatures and recommends suitable composites for applications. See References and Further Reading at the end of this section.

PUMP CONSTRUCTION

The pump section views in Figures 5 to 8 illustrate the construction of some composite pumps.

Components Figure 9 shows various pump components made from both thermoplastic and thermoset polymers. The enclosed impellers are made of thermoplastic resins because the front and back shroud can be ultrasonic- or vibration-welded together. Semi-open impellers can be made of either type of composite. On sizes greater than 6 in (152 mm) diameter, thermosets are usually used due to their superior strength over thermoplastics. When thermoplastics are used in larger impellers, a metal reinforcing skeleton is used for reinforcement. Casing and casing covers are made of either composite depending on the pump size and pressure rating.

The seal between the casing and cover is usually a shaped “O-ring” held by a groove running around the perimeter of the casing. The “O-ring” is compressed between the casing and cover with a ring of bolts around the casing’s perimeter.

Shafts are made of AISI 4140 or 316 stainless steel with a thermoplastic shaft sleeve for corrosion protection. Mechanical seals are of “outside” construction to prevent liquid from contacting the metal portion (springs) of the seal.

Bedplates Composite materials have several advantages over metals and other materials when used for bedplates. Composites do not rust, rot, or deteriorate in adverse environments. The composite bedplate maintains its shape over time. This assures that the pump shafts will stay in alignment, and the pump’s nozzle will not be stressed due to a sagging bedplate. The designed in features in a composite bedplate include an integral drip lip, slopping surface to collect drips, and ringed grout holes.

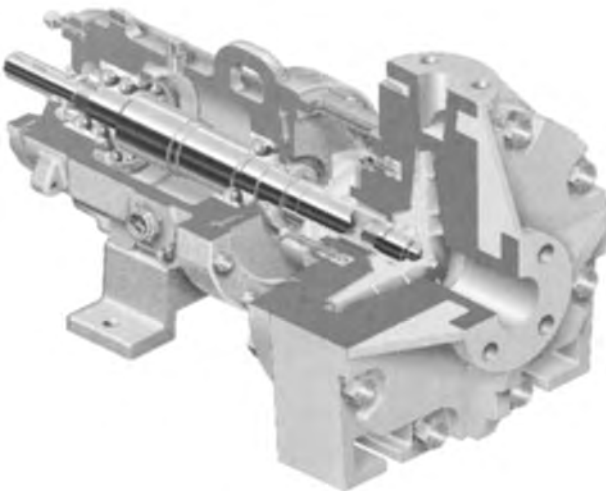


FIGURE 5 Thermoset ASME B73.5M pump (Flowserve Corporation)

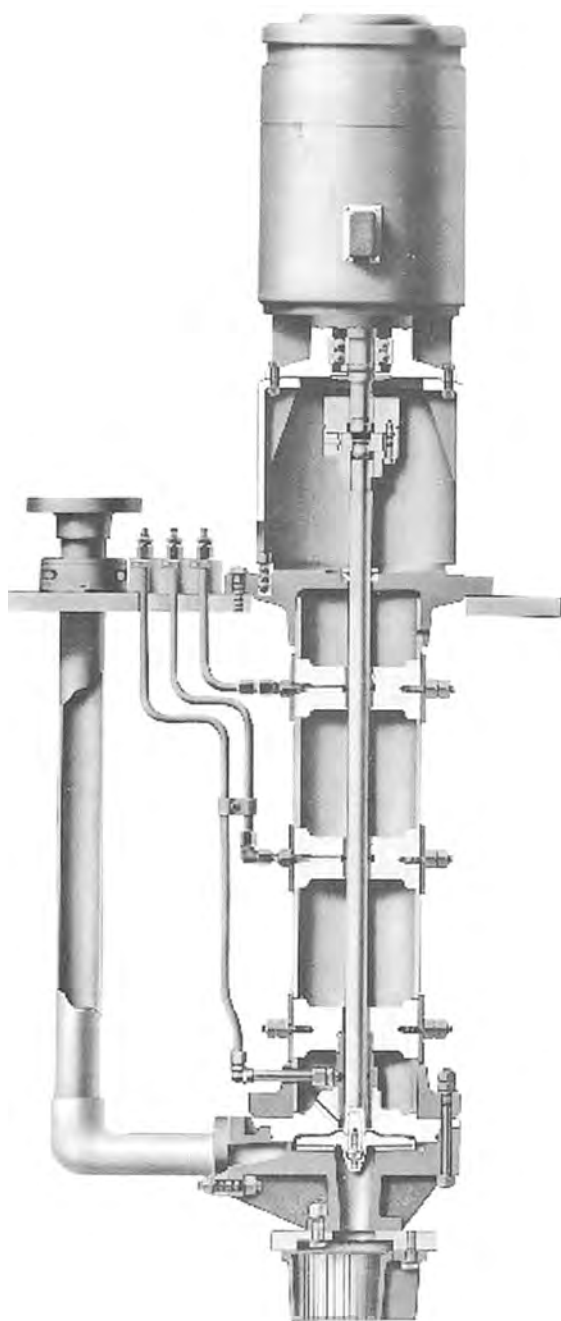


FIGURE 6 Vertical immersion pump (Flowserv Corporation)



FIGURE 7 Thermoplastic coolant pump (Flowserve Corporation)

Disclaimer The data and notes included in this section are only for general guidance. No recommendations are intended as a guarantee by the author or other sources. Pump manufacturers and material suppliers can supply specific application information for their products when used with specific liquids in identified services and environments.

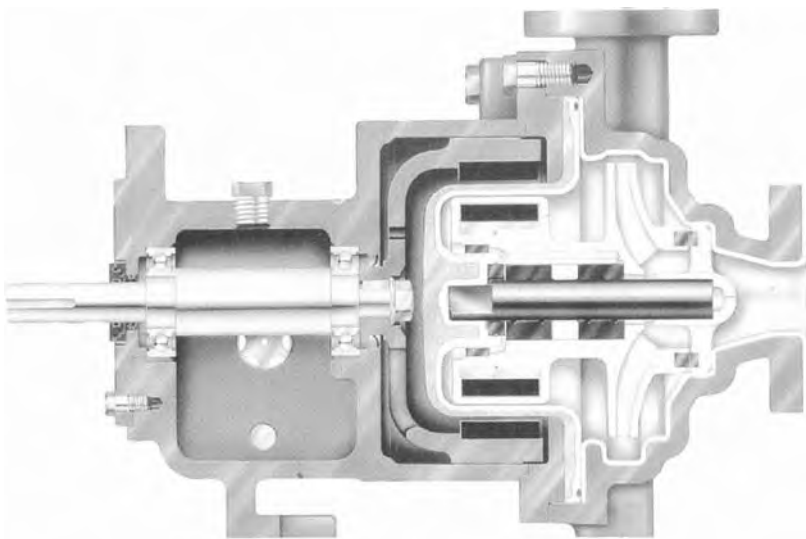


FIGURE 8 Fluoropolymer lined magnetically coupled sealless pump (Goulds Pumps)

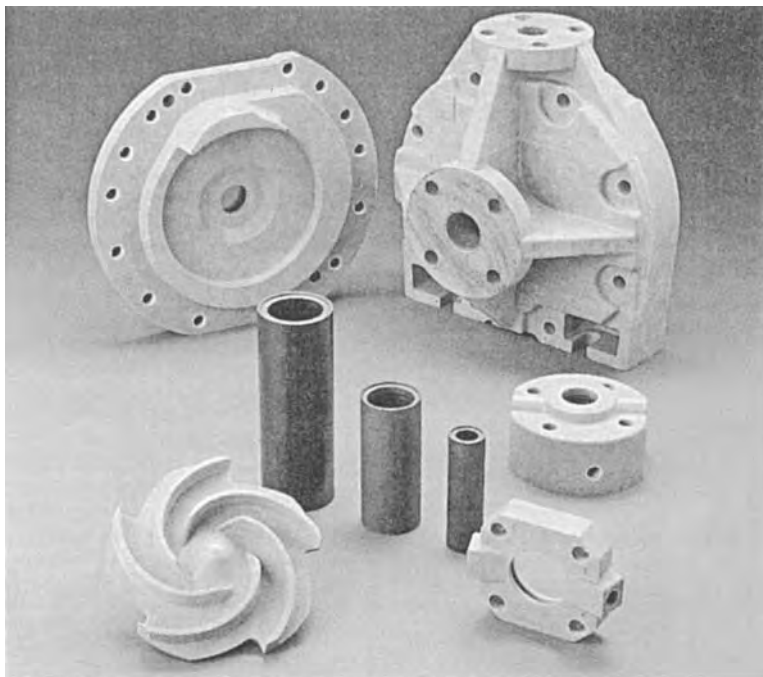


FIGURE 9 Components made of thermoset (Flowserve Corporation)

REFERENCES AND FURTHER READING

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Internet Resources

Injection Molding Magazine's listing of books: <http://www.immbookclub.com>

Plastic technology consulting source: <http://www.rapra.net/intro.htm>

Listing of plastic resources and issues: <http://www.polysort.com>

PUMP DRIVERS AND POWER TRANSMISSION

SECTION 9.1

PUMP DRIVERS

9.1.1

ELECTRIC MOTORS

AND MOTOR CONTROLS

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The two most common types of electric motors are *alternating current (ac) induction* and *direct current (dc) commutated shunt- or series-wound*. All electric motor configurations except one type (reluctance brushless) convert electrical energy to mechanical energy from the magnetic flux linkage of their two magnetic circuits. One of these circuits is in the stator and the other is in the bearing-mounted rotor. This flux linkage between the two magnetic circuits produces a moment of force at the rotor radius that results in a torque on the motor shaft causing shaft rotation. The speed of the rotation times the torque equals the output power at the motor shaft. This is, of course, the power used to drive a pump.

The basic difference between these two types of electric motors has to do with their electrical power source. The first type has been historically powered by 60 cycle (Hz) alternating voltages direct from the public utility power grid (50 Hz in most of Europe and some other parts of the world). For this type of motor, the speed is determined by the number of magnetic poles designed in the motor and the alternating sinusoidal frequency of the voltage from the power grid. Methods have been developed to alter the speed of an ac motor with a fixed number of poles and a fixed line frequency (it is also possible to wind such a motor with several poles for other speeds). However, the squirrel-cage induction motor remains without question the most common type of motor used to drive pumps. The reasons for this are worldwide availability, excellent reliability, excellent performance characteristics, and ease of replacement.

Because of its adaptability to variable speed using ac inverters and vector drives, the ac induction motor will continue to be the number one prime mover for all types of pumps as the pump industry continues to adopt variable speed motor/drives.

The other principle motor type used to drive pumps is the dc motor. The dc electrical machine also contains two magnet fields, one in each of the stator and rotor assemblies of the motor. Dc current creates the rotor field provided through commutator brushes. These brushes have a finite life and are the principal maintenance/failure point in this type of machine. However, in some applications, particularly those where only dc (battery)

power is available, these machines are common. The dc motor, powered by the voltage from storage batteries or from a dc generator, can be speed-adjusted by varying the voltage with a power supply. In a limited number of instances dc motors are used where adjustable speed is required. Its speed relationship to the voltage is linear and very useful for some pump applications such as constant displacement types, which require speed adjustment to set flow. [See Section 9.2.2.] Most dc motor applications are in the smaller sizes such as automotive and off road equipment. The overall use of dc motors for pump drives is predicted to decline.

A third type of pump prime mover is the permanent magnet machine, sometimes called a brushless dc motor. Unlike dc motors permanent magnet motors have no brushes; unlike the induction machine there is no squirrel cage with induced electric current either; the field is generated by permanent magnets, offering a more reliable prime mover for a broad range of applications, with superior performance and long-life benefits. Permanent magnet motors require a motor drive (inverter, variable speed or frequency drive) to operate; very simple so-called "sensoreless" drives can be used for pumping applications.

There are other types of electric motors considered for driving pumps. The reasons for this include the dramatic recent advances in power electronics and microprocessors, advances in motor materials such as permanent magnets, and advances in the pumps themselves. Besides the active interest in adjustable speed pumps, another reason for the interest in some newer motor types for pump applications is because of recent U.S. government regulations enacted to improve energy conservation by implementing motor efficiency mandates along with a time schedule. For example, the *U.S. Energy Policy and Conservation Act* of September 10, 1992 (EPACT) stipulates that all covered electric motor products must meet the efficiency levels per NEMA MG1 1993. The requirement covers all electric motors from 1–200 horsepower (1–150 kW) manufactured after October 24, 1997 that operate from 230/460 VAC power at 60 Hz line frequency. The U.S. Department of Energy approved test method to meet the new efficiency levels is per IEEE-112B. This regulation was originally intended for ac induction motors.

For those pump applications that require the motor torque to increase as the square of speed, these new, more efficient motors will run at higher speeds because their slip is less. This could cause an appreciable increase in overload within the motor. Overload could also result in other mechanical parts of the system. There are other ramifications resulting from these new requirements that must be carefully analyzed when selecting an electric motor, starter, or inverter. The resulting analysis might very well cause the selection to be some other electric motor type, such as a permanent magnet brushless dc, permanent magnet ac, synchronous (sine wave driven version of the brushless dc), or even a switched reluctance brushless dc motor. Each of these types must be powered with an inverter and a controller. However, the result can most likely offer adjustable speed with a very high efficiency over a wide speed range. This use of an inverter is required for each of those other types of motors mentioned. With the availability of the new vector controlled ac inverters (Section 9.2.2), the high-efficiency ac induction motor can also be applied for variable speed. This will eliminate the need for a soft starter frequently required with line fed motors. The elimination of the starter helps somewhat to offset the additional cost of the inverter.

TYPES OF MOTORS

Alternating-Current Motors

SQUIRREL-CAGE INDUCTION MOTOR By far the most common motor used to drive pumps is the squirrel-cage induction motor (Figure 1). This motor consists of a conventional stator wound with a specific number of poles and phases, and a rotor that has either cast bars or brazed bars imbedded in it.

The squirrel-cage induction motor operates at a speed below synchronous speed by a specific slip or revolutions per minute. The synchronous speed is defined as

$$N = \frac{f \times 60 \times 2}{p}$$

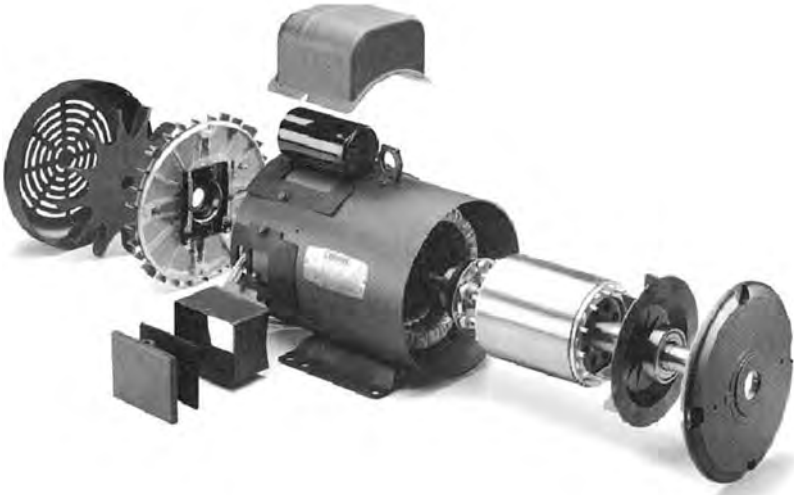


FIGURE 1 Exploded view of a typical squirrel-cage induction motor (Courtesy of Leeson Electric Corp.)

where N = speed, rpm

f = line-power frequency, Hz

p = number of poles

The percent slip is defined as

$$\% \text{ slip} = \frac{(N - s) \times 100}{N}$$

where s = slip, rpm

When the stator winding of a squirrel-cage induction motor is connected to a suitable source of power, a magnetic flux is generated in the air gap between the stator and rotor of the motor. This flux revolves around the perimeter of the air gap and induces a voltage in the rotor bars. Because the rotor bars are short-circuited to each other at their ends (end rings), a current circulates in the rotor bars. This current and the air-gap flux interact, causing the motor to produce a torque.

The squirrel-cage induction motor exhibits a characteristic speed-torque relationship that is determined by the resistance of the rotor bars. Thus, the desired speed-torque characteristics are obtained by selecting a metal of suitable resistance when designing the rotor bars. The slot shape and size for the bars in the rotor can be selected to achieve a certain rotor resistance.

Figure 2 suggests several typical speed-torque characteristics that have been standardized by NEMA (National Electrical Manufacturers Association), covering motor frames 143T through 449T. Motors larger than 449T may not have these same values, but generally have the same characteristic curves. Also, single-phase motors may not exhibit these characteristics and are defined specifically by NEMA with different values.

Most pumps are driven by NEMA B characteristic motors when operated from three-phase power sources.

WOUND ROTOR INDUCTION MOTOR The wound-rotor induction motor is in every respect similar to the squirrel-cage version except that the rotor is wound with insulated wire turns and this winding is terminated at a set of slip rings on the rotor shaft. Connections are made

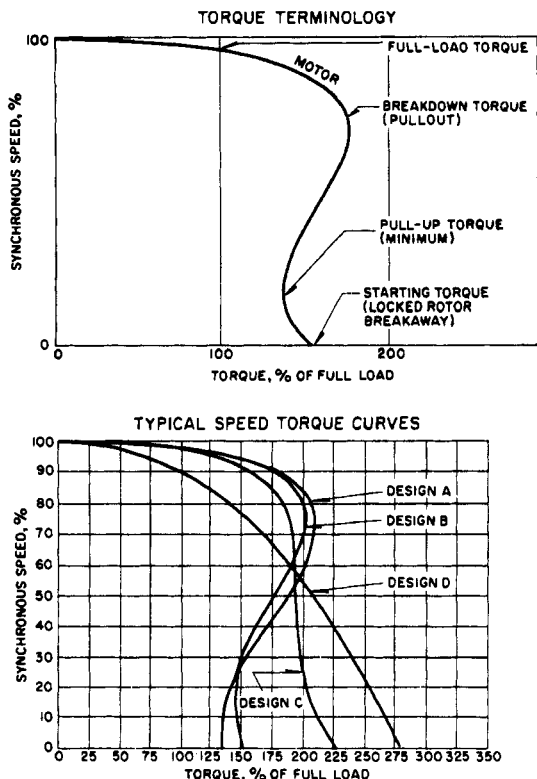


FIGURE 2 These curves are characteristic of NEMA frame size squirrel-cage induction motors through size 449T (typically through 300 hp–224 kW)

to the slip rings through brushes and in turn to an external resistor, which can be adjusted in ohmic value to cause the motor speed-torque characteristics to be changed. These types of motors have been used in some pump applications in the past, but due to the availability of inverter-fed ac squirrel-cage induction motor drives, they are no longer very practical.

Figure 3 demonstrates the speed-torque characteristics of a wound-rotor induction motor for several resistor values. It will be noticed that increasing the external resistance of the control will cause the peak torque of the motor to be developed at lower speeds until the peak torque occurs at zero speed. Increasing the resistance beyond this value will cause the motor to have a limited torque as, for example, curves 4, 5, and 6. This motor can be used where torque control is required or where variable speed is necessary. In the variable-speed application, the rotor resistance is adjusted to produce a motor torque that matches the load torque at the specific speed desired. This system is not as useful and cost-effective as an inverter-fed ac induction motor due to the recent developments of power electronics and microprocessors.

SYNCHRONOUS MOTOR The synchronous motor is also similar to the squirrel-cage induction motor except that it operates at synchronous speed and its rotor is constructed with definite salient poles on which a field coil is wound and connected to a source of direct current for excitation. The most common synchronous motor is constructed with slip rings on the rotor shaft to connect the dc excitation to the field coils.

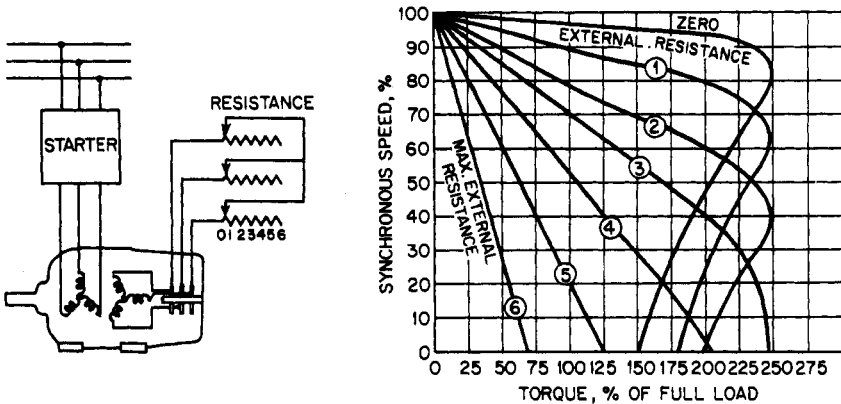


FIGURE 3 Typical speed-torque characteristics of a wound-rotor induction motor (Westinghouse Electric)

There are various means of providing the dc power to the slip rings:

1. **Static excitation** The power to be connected to the slip ring brushes on the motor shaft is obtained from a transformer and rectifier package external to the motor.
2. **Direct-connected exciter** This arrangement has a dc generator directly connected to the synchronous motor shaft (Figure 4). The dc power from this generator is connected to the brushes of the synchronous motor slip rings.
3. **Motor-generated exciter** The dc power for exciting the synchronous motor is generated by means of a remote motor-generator set operating from normal ac power, and the dc voltage from this motor-generator set is connected to the brushes of slip rings of the synchronous motor.

Another form of synchronous motor is known as the *brushless synchronous motor* (Figure 5). As the name implies, this motor has its rotating field excited without the use of slip rings for connecting the external direct current to the motor field. The construction

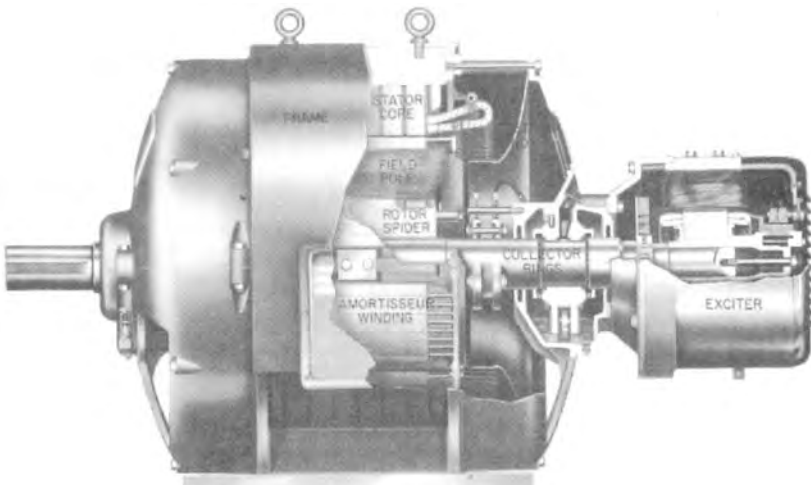


FIGURE 4 Synchronous motor with direct-connected exciter (Electric Machinery Manufacturing)

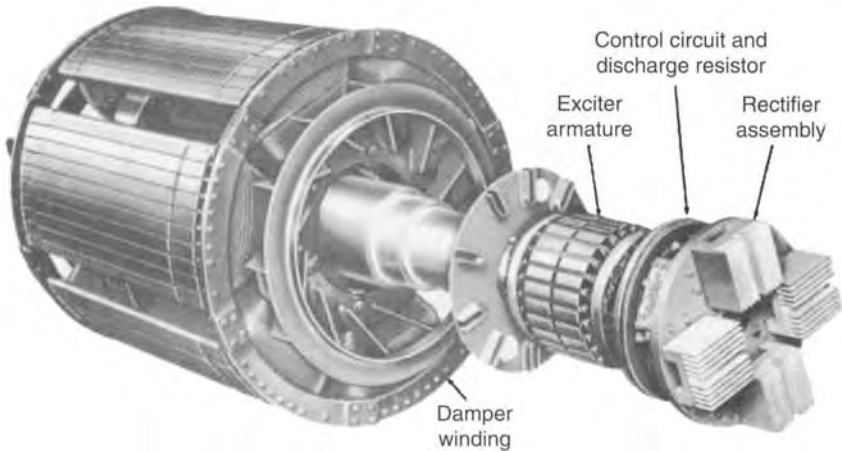


FIGURE 5 Brushless synchronous motor with ac generator mounted on the shaft with rectifier and control devices (Electric Machinery Manufacturing)

of this motor incorporates a shaft-connected ac generator. The field of the ac generator is physically stationary and connected to a source of dc voltage. The rotor of this ac generator is connected through a solid-state controlled rectifier mounted on the synchronous motor rotor and in turn connected to the synchronous motor field. This arrangement facilitates a connection between external excitation power and the rotating field of the synchronous motor through the air gap of the shaft-connected ac generator. The brushless synchronous motor has many advantages over the conventional slip-ring synchronous motor. Among these are the elimination of brushes and slip rings, which are high-maintenance items; the elimination of sparking devices, which are not permissible in certain atmospheres; and the use of static devices for field control, which are more reliable than conventional electromagnetic controls.

Synchronous motors are used for pump applications requiring larger horsepower ratings at lower speed conditions, as illustrated in Figure 6. Also, they are used on applications where a high power factor or a power-factor-correction capability is desired. Of less importance is the characteristic of the synchronous motor that it will always operate at synchronous speed (does not have a slip) regardless of load. Synchronous motors are started on their damper windings (the same as squirrel-cage induction motors), and when they have accelerated to within 5% of synchronous speed, the field is applied and the motor accelerates to synchronous speed (Figure 7). Typical characteristic curves are shown in Figure 8.

Direct-Current Motors Dc motors are only occasionally used to drive pumps. Most of the current applications for dc pump drives are in some form of automobile or off-the-road equipment due to the dc power from storage batteries. There are some other situations that might call for dc motor pump drives such as shipboard duty, railway applications, aircraft, mining installations, and some other emergency battery operations.

There are three types of dc motors available (Figure 9): shunt, series, and compound-connected.

Larger horsepower ratings of shunt-wound dc motors are frequently qualified as “stabilized shuntwound” motors and incorporate a series field similar to that of a compound wound motor. This is necessary to adjust the regulation of the shunt motor so as not to exhibit a rising speed-torque characteristic. It is important to be aware of the speed at which a dc motor will operate on pump applications because of pump performance guarantees.

The windings in the stators of the wound types are connected to the armature windings through brushes and a commutator three different ways to achieve different performance

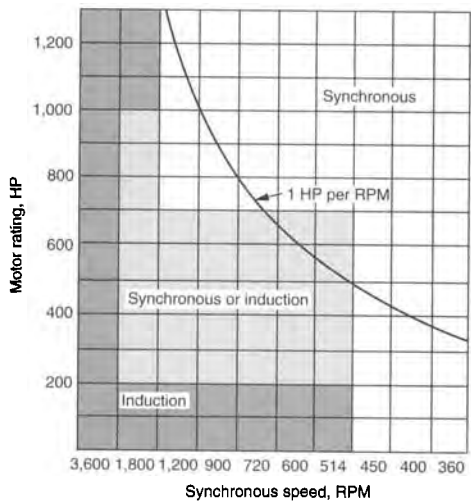


FIGURE 6 Below 514 rpm, or powers greater than approximately 1 hp/rpm (0.746 kW/rpm), synchronous motors are a better selection than squirrel-cage induction motors because higher cost can generally be offset by higher power factor and efficiency (from *Power* special report, "Motors," June 1969)

characteristics (see Figure 3 for details). With the use of pulse width modulation (PWM) or silicon-controlled rectifiers (SCR), variable dc power supplies the armature voltage, which can be controlled along with a separate control for the field current (see Section 9.2.2). Using this control scheme, the dc motor can be used to control speed over a very wide range in a smooth manner. For example, constant voltage can be produced from zero rpm up to base design speed of the motor by armature control and constant power above base speed using flux weakening of the armature field. When operating from a constant voltage, dc motors are available to provide up to a 4 to 1 speed range with an adjustable dc field power supply. With this separate control scheme for both field and armature, a 100 to 1 speed range is easily achievable. Figure 9 summarizes the various

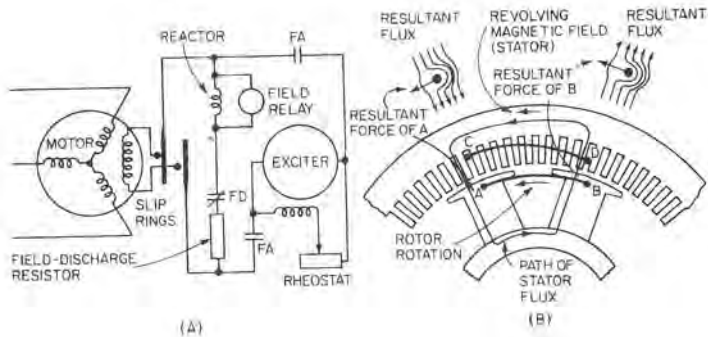


FIGURE 7 Method of starting synchronous motor. (A) Typical field control (brush type) energizes the dc field to pull the rotor into synchronism as it comes up to speed. Field relay senses change in induced frequency as motor speeds up. (B) Damper winding is similar to a squirrel-cage rotor winding. It produces most of the starting torque, and the synchronous motor starts with essentially induction-motor characteristics (from *Power* special report, "Motors," June 1969).

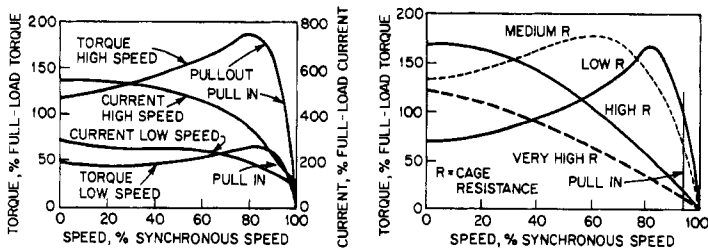


FIGURE 8 Characteristics of synchronous motors depend on rotor design. Torque and current relations are influenced by synchronous speed. High-resistance cage produces high starting torque but low pull-in torque (from *Power* special report, "Motors," June 1969).

field connection schemes for dual wound dc machines with mechanical commutation systems. Figure 10 illustrates typical characteristics of the dc motors discussed. Figure 11 shows the equivalent circuit of a permanent magnet type dc motor.

Permanent Magnet (PM) Brushless Motors The permanent magnet brushless dc motor has been in existence for about three decades and is finally being widely used for many applications^{1,5,6}. Its being brushless has frequently been mentioned as justification for its cost, higher than most other motor types. However, the PM brushless motor has two other features that are arguably more important than the fact that it contains no mechanical brushes and commutator for commutation of the phase windings to the power source. First, it produces the highest continuous output power per unit volume of any motor yet invented. The other important virtue of a PM brushless motor is that it produces its output power with the least input power. For a given size and output performance envelope, it has the most efficient motor of any electric motor yet invented. With the emphasis on reducing power consumption, this type of motor will be used in many pump applications. In addition, and perhaps most importantly for the application of pumps, the PM brushless motor possesses one other very important feature or use of the permanent magnets contained within the rotor assembly. Although the magnets are there to supply a magnetic field from the rotor to pass through the air gap of the motor into the stator, these same magnets can serve as a magnetic coupling. If a sleeve-shaped liquid barrier made of a suitable material to be impervious to the liquid is fitted between the rotor and stator, the PM brushless can be used as the best hermetically sealed wet motor known. There are, of course, many possible configurations of this PM brushless hermetic motor concept. These PM brushless hermetic motors can be configured as axial gap²,

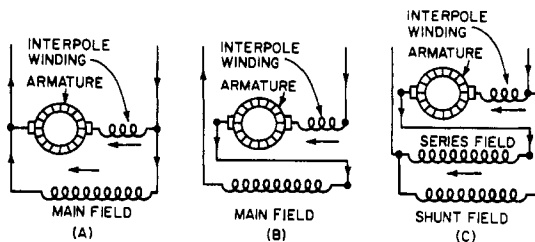


FIGURE 9A through C Types of dc motors. (A) Shunt motor has field winding of many turns of fine wire connected in parallel with the armature circuit. The interpole winding aids commutation. (B) Series motor has field in series with the armature. Field has a few turns of heavy wire carrying full-motor current flowing in the armature. (C) Compound motor has both a shunt and a series field to combine characteristics of both shunt- and series-type motors in the same machine (from *Power* special report, "Motors," June 1969).

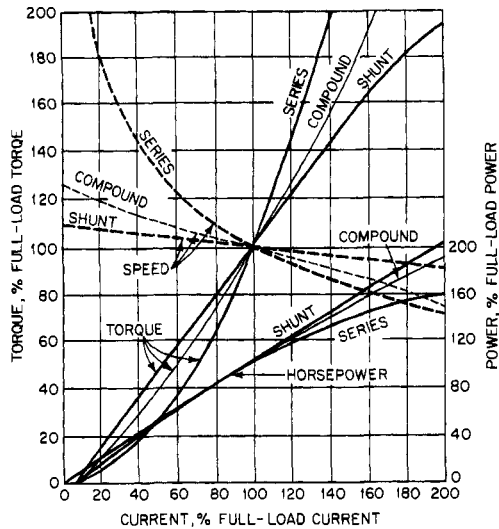
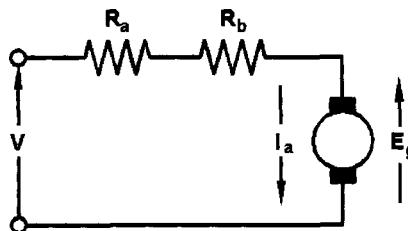


FIGURE 10 Speed, torque, and power characteristics of dc motors (from *Power* special report, "Motors," June 1969)

radial gap inside rotor, or radial gap outside rotor motors (Figure 12). The choice depends upon the best way to design the pump integration.

All PM brushless motors must be powered with an electronic inverter and controller, which can be used for controlling pump performance in controlled loop systems of all types. The electronic drive can be a simple square wave, a six-step trapezoid type, or a sinusoidal drive. The PM motor is a synchronous machine and the controller needs to know the shaft position; in some applications a resolver or encoder is used. However, in many pumping applications a 'sensorless' controller can be used. The choice of controller topology is application specific, but not difficult.

Three permanent magnet choices can be used for these motors. The lowest priced magnet material is known as ferrite or ceramic with a magnetic flux output of about one-third



V = Terminal voltage, volts
 R_a = Armature resistance, ohms
 R_b = Resistance of the brushes, ohms
 I_a = Armature current, amperes
 E_g = Back-EMF, volts

FIGURE 11 The equivalent circuit of a dc motor with a permanent magnet stator

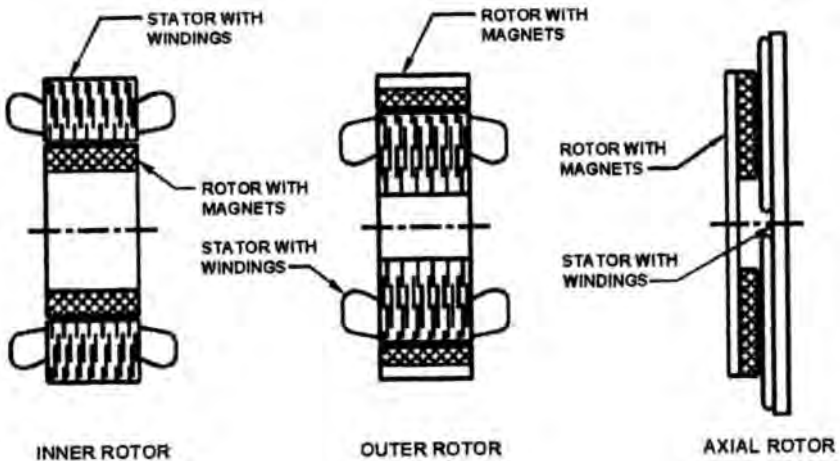


FIGURE 12 Inner, outer, and axial rotor PM brushless motor choices for pump drives

to one-half of the other two magnet choices. By careful design, this magnet material can be used for 50 hp (37 kW) and up in PM brushless machines to keep costs competitive with legacy motors. The two high energy or rare earth magnet materials, Samarium Cobalt and Neodymium Iron Boron, can be used from 1 to 50,000 hp (36 MW). The more expensive of the two, Samarium Cobalt, is used for high temperature and other hostile environment pump motor applications. The newer rare earth magnet material, Neodymium Iron Boron, is much lower in cost, with the highest useful magnetic flux at moderate operating temperatures. Permanent magnets can be assembled onto the surface of the rotor or they can be imbedded within the rotor structure. The imbedded magnet design can be used to direct drive high hp (kW) pumps to several thousand rpm. In either design, the rotor can be made to be quite robust and yield a very long useful operating life. In fact, the only failure modes for PM brushless motors have to do with bearings or winding insulation. Figure 13 shows examples of large PM brushless motors.

The speed can be controlled to greater than 100 to 1 if required with constant power over the highest speed range of any other motor type. The efficiency remains very high over the entire speed range. There are many design possibilities including axial gap and outside rotor configurations. For example, because of the high magnetic strength of the permanent magnets, the rotor of a PM brushless motor can also serve as the impeller of a pump that is integrated with the motor in a hermetically sealed package with no shaft seals. Utilizing the axial-gap rotor configuration of Figure 12, the concept is illustrated in Figure 14. Commercially significant impeller torque levels can be generated by the PM brushless motor in this configuration⁹. This eliminates the need for a separate magnetic coupling or larger-size canned induction motor (see the discussion at the end of this section and in Section 2.2.3).

The PM brushless motor can be driven with either square wave currents over a 120 degrees electrical commutation angle or with sinusoidal currents over 180° electrical commutation angle. The latter scheme generally yields lower torque fluctuations, which is sometimes important for pump applications. Either drive requires shaft angle feedback data to tell the phases when to be powered. This feedback data are also a requirement for vector driven ac induction motors. The sensor used for this feedback can be as simple as *Hall* switches mounted in the motor that send out a pulse to the controller each time a rotor magnetic pole changes polarity as the motor rotates. An optical encoder or a *resolver* can also be used for this purpose. Several sensorless or remote sensors have been developed that capture the rotor angle location from the stator phase windings so a shaft-mounted sensor is not required.

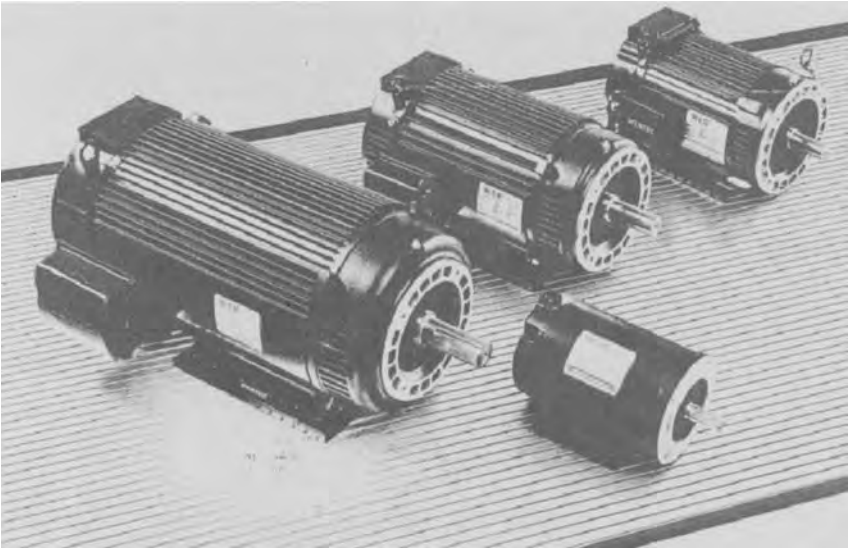


FIGURE 13 Large horsepower (kW) high-performance permanent magnet inside rotor brushless motors (courtesy of Pacific Scientific)

Switched Reluctance Brushless Motor The SR brushless dc motor is one of the oldest motors known, but it has not been used much until recently. Its main feature is that it is a true brushless motor with most of the virtues of its PM cousin, but it does not require permanent magnets⁴. Proponents of switched reluctance motors point out its inherent robustness. The rotor of the SR motor is so simple and rugged it can survive very at high speeds as well as in a variety of other unusual environmental conditions.

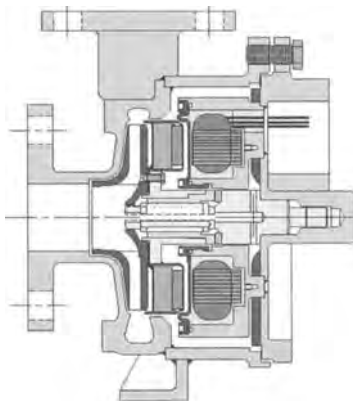


FIGURE 14 Permanent magnet brushless motor integrated into a sealless pump. Permanent magnets are mounted on the impeller, which serves as the rotor of the motor. (Courtesy of Flowserve Corporation)⁵

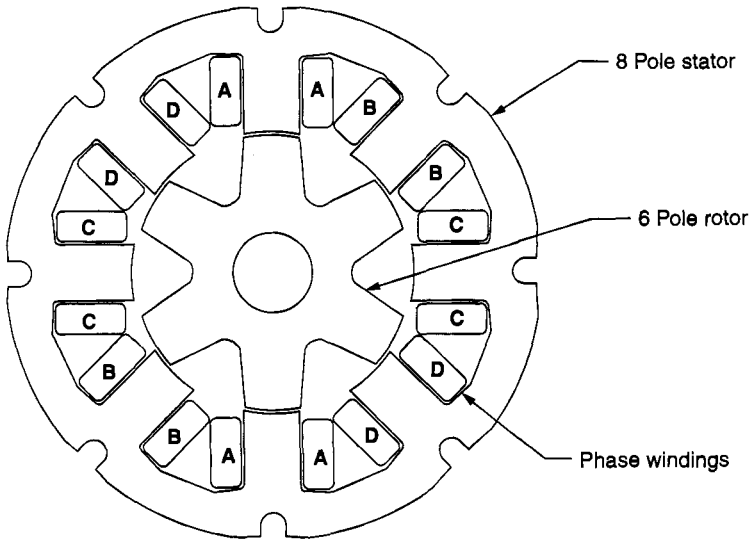


FIGURE 15 Switched reluctance brushless dc motor, (6) rotor poles, (2) stator poles per phase

Figure 15 shows the typical cross section of a switched reluctance motor, illustrating how simple this motor is with phase coils placed around the stator poles. The rotor consists of a set of magnetic steel gear-shaped laminations taken from the bore of the stator laminations. They are stacked and retained on the motor shaft. No magnets or windings of any kind are required to produce torque. The SR motor is said to be a doubly salient pole machine. The torque is produced by the magnetic attraction of the closest rotor poles to those stator poles which are magnetized by the phase coils. There is an abundance of technical information regarding the performance details of this technology. The important point is that the SR motor is an excellent choice for high-speed high-powered centrifugal pumps.

There is a considerable difference in the inverter topology for the SR motor as compared to either the PM brushless or the ac induction motor. The SR machine must be driven as a unipolar machine rather than a bipolar motor like the other two. This means the standard ac inverter cannot be used for the SR motor. The phases are each connected to the dc power in parallel rather than in the standard bridge fashion with all of the phases connected together at a center tap. Figure 16 shows the normal power circuit for the (4) phase SR motor shown in Figure 15. The same circuit is applicable for both two phase or three phase. It is essentially an independent half bridge circuit for each phase. Certain two-phase designs are very cost effective for high speed pump drives that are required to rotate in only one direction without the need for reversing.

MOTOR ENCLOSURES

Electric motors are manufactured with a variety of mechanical enclosure features to provide protection to the working parts for specific environmental conditions. Although these special enclosures are classified, they are not, for reasons of economy, available for all size motors. The electric motor industry has incorporated a number of specific enclosure classifications on standard designs of small- and medium-size motors.

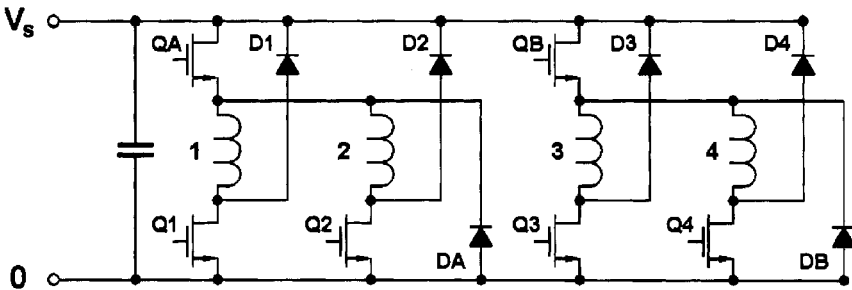


FIGURE 16 Power control circuit for (4) phase SR motor, (6) transistors, (6) diodes, (3) current sensors, (1) capacitor, and (6) motor lead connections

The following list describes the motor enclosure specifications available for polyphase ac induction motors used with pumps. This availability depends on demand. Furthermore, the PM brushless and SR brushless motors might not be available in some of the enclosures listed. In general, totally enclosed motors are required for most integral horsepower (kilowatt) pump applications regardless of the motor type. In some instances, such as high-speed centrifugal pumps with integral motors, the motor is furnished to the pump manufacturer as a rotor and stator parts kit. The motor is then included in the pump enclosure.

OPEN This enclosure permits passage of external cooling air over and around the windings and rotor of a motor and normally includes no special restrictions to ventilation other than those that are inherent in the mechanical parts of the motor. Most designs include openings at each end of the motor frame either in the bearing end frame or openings around each end of the frame.

OPEN DRIPPROOF This is an open machine with ventilating openings designed to permit satisfactory operation when liquids or solids fall on the machine at any angle up to 15° from the vertical. Openings are normally in the bottom portions of the frame assembly. This construction includes mechanical baffling to prevent materials from entering the machine within these limits.

SPLASHPROOF This is an open machine with ventilating openings designed to permit satisfactory operation when liquids or solids fall *directly* on the machine or come toward the machine in a straight line at any angle up to 100° from the vertical.

GUARDED ENCLOSURE This provision limits the size of the ventilating opening to prevent accidental contact with the operating parts of the motor other than the shaft.

SEMIGUARDED This is an open construction where some of the ventilating openings are guarded and the remaining openings are left open.

OPEN, EXTERNALLY VENTILATED This motor is ventilated by a separate motor-driven blower mounted on the motor enclosure (piggyback construction).

OPEN, PIPE-VENTILATED This motor is equipped to accommodate an air-inlet duct or pipe for accepting cooling air from a location remote from the motor. Air is circulated in the motor either by its own internal blower parts or by an external blower, in which case the motor is said to be *forced-ventilated*.

WEATHER-PROTECTED TYPE I This is an open-construction motor with ventilating openings to minimize the entrance of rain, snow, and airborne particles to the motor's electrical parts, and with the openings arranged to prevent the passage of a $\frac{3}{4}$ -in (19-mm) round rod.

WEATHER-PROTECTED TYPE II This construction has the same features as the type I machine, but in addition the intake and discharge ventilating passages are designed so high-velocity air and airborne particles blown into the machine by storms can be discharged without entering the internal ventilating passages of the motor leading to the electrical parts. The ventilating passages leading to the electrical parts of the motor are provided with baffles or other features to allow at least three abrupt changes of at least 90° for the ventilating air. The intake air path and openings are proportioned to maintain a maximum of 600 ft/min (3 m/s) velocity of the entering air.

TOTALLY ENCLOSED This motor is designed without air openings, so there is no free exchange of air between the inside and outside of the motor frame; the construction is not liquid or airtight. Normally the rotor has die-cast air circulating fins at each end to circulate the trapped inside air and improve convection cooling to the frame.

TOTALLY ENCLOSED, FAN-COOLED This totally enclosed motor is equipped with an external fan operating on the motor shaft to circulate external air over the outside of the motor. Most motors have cast fins the length of the frame to increase surface area and take advantage of the fan cooling.

EXPLOSIONPROOF This totally enclosed motor is designed to withstand an internal explosion of gas or vapor and constructed to prevent ignition by the internal explosion of gases or vapors outside the motor.

TOTALLY ENCLOSED, PIPE-VENTILATED This motor enclosure is similar to the open, pipe-ventilated motor except that it is equipped to accept outlet ducts or pipes in addition to inlet ducts or pipes.

TOTALLY ENCLOSED, WATER-COOLED This is a totally enclosed motor cooled by water passages or conductors internal to the motor frame.

TOTALLY ENCLOSED, WATER-TO-AIR HEAT EXCHANGE This is a totally enclosed motor equipped with a water-to-air heat exchanger in a closed, recirculating air loop through the motor. Air is circulated through the heat exchanger and motor by integral fans or fans separate from the rotor shaft and powered by a separate motor.

TOTALLY ENCLOSED, AIR-TO-AIR HEAT EXCHANGE This motor is similar to the water-to-air heat exchanger motor, except external air is used to remove the heat from the heat exchanger instead of water.

SUBMERSIBLE This totally enclosed motor is equipped with sealing features to permit operation while submerged in a specified medium at a specified depth.

Environmental Factors The environmental conditions of the pump application dictate the type of motor enclosures to be used. A brief set of rules for selecting motor enclosures follows.

DRIPPROOF For installation in nonhazardous, reasonably clean surroundings free of any abrasive or conducting dust and chemical fumes. Moderate amounts of moisture or dust and falling particles or liquids can be tolerated.

MILL AND CHEMICAL MOTOR (TO NEMA FRAME 449T) For installation in nonhazardous, high-humidity, or chemical applications free of clogging materials, metal dust, or chips, or where hosing down or severe splashing is encountered.

TOTALLY ENCLOSED, NONVENTILATED OR FAN-COOLED For installation in nonhazardous atmospheres containing abrasive or conducting dusts, high concentrations of chemical or oil vapors, where hosing down or severe splashing is encountered.

TOTALLY ENCLOSED, EXPLOSIONPROOF For installation in hazardous atmospheres containing:

- Class I, Group D* Acetone, acrylonitrile, alcohol, ammonia, benzene, benzol, butane, dichloride, ethylene, gasoline, hexane, lacquer-solvent vapors, naphtha, natural gas, propane, propylene, styrene, vinyl acetate, vinyl chloride, or xylenes
- Class II Group G* Flour, starch, or grain dust
- Class II, Group E* Metal dust including magnesium and aluminum or their commercial alloys
- Class II, Group F* Carbon black, coal, or coke dust

NOTE: Under Class 1 only, there are two divisions that allow some latitude on motor selection. Generally, Class 1, Division 1 locations are those in which the atmosphere is or may be hazardous under normal operating conditions, including locations which can become hazardous during normal maintenance. An explosionproof motor is mandatory for Division 1 locations. Class 1, Division 2 refers to locations where the atmosphere may become hazardous only under abnormal or unusual conditions (breaking of a pipe, for example). In general, a motor in a standard enclosure can be installed in Division 2 locations if the motor has no normally sparking parts. Thus, open or standard totally enclosed squirrel-cage motors are acceptable, but motors with open slip rings or commutators (wound rotor, synchronous or dc) are not allowed unless the commutators or slip rings are in an explosionproof enclosure.

BEARINGS AND LUBRICATION

Very large horsepower (kilowatt) motors are generally supplied with oil-lubricated sleeve bearings with oil supplied from a reservoir. In some cases, pressurized oil lubrication systems are installed by the pump manufacturer along with hydrodynamic thrust bearings. All NEMA frame induction motors are available with ball bearings. These standard ball bearings are normally permanently grease lubricated. The bearings used in a motor must be sealed to keep the lubricant inside the bearings and keep contaminants from getting into the bearings. Double-sealed bearings are common for many pump applications.

Ball bearings are subject to early failure when used in electric motors driven by PWM inverters. This very common problem must be addressed. It is caused by the high carrier frequency used in the inverter to generate the sinusoidal currents for each phase. This results in generation of high common-mode voltages inside the phase windings of the stator. Because there is an excellent electrostatic coupling between the stator/frame and the rotor from the windings, a voltage is induced in the shaft. The ball bearings represent the least-resistant path for a short circuit to the stator. However, the balls seldom actually contact the races because of the film of grease or oil in between. When the voltage builds up in the shaft until it is greater than the insulating capability of the film of lubricant, the voltages arc across the lubrication gap and a flashover current goes through the bearing. In a relatively short amount of time, the bearing races will become grooved, causing the bearings to become noisy. Metal particulate will then egress from the bearing surfaces as the process continues, causing catastrophic bearing failures after a few months.

Therefore, all electric motors that are driven by PWM drives must have a shaft grounding system to provide a low resistance path between the shaft and the motor

frame. There are other solutions to this potential problem, which can be discussed with the motor supplier.

Sleeve Bearings Motors that use oil-impregnated porous sleeve bearings are lubricated with an oil-soaked wick. These bearings are available in motors up to approximately 1 hp (0.75 kW). Sleeve-bearing motors larger than 1 hp (0.75 kW) are ring-lubricated. Lubricating oil is drawn up from the bearing sump to the bearing by a ring that rolls over the top of the motor shaft as the shaft rotates. Larger motors, having bearing heat losses that cannot be dissipated directly, may require the use of a pressurized lubrication system wherein oil is pumped into the bearings and allowed to recirculate through a heat exchanger. The oil delivered to each bearing is metered to provide only the required amount. A lubrication system composed of heat exchanger, sump, and pump is normally common to a number of bearings, rather than having a single lubricating pump for each bearing. Other types of bearings must be used in place of, or in addition to, sleeve bearings when thrust loads are present. Smaller sleeve bearings are in the form of a cylindrical shell and are usually made of bronze or steel-backed babbitt metal. Larger sleeve bearings are usually split on a horizontal centerline, allowing easy assembly and disassembly for inspection and replacement. The bearing housing is also split on the horizontal centerline and held together with bolts between the top and bottom halves.

Rolling Element Bearings All new electric motors manufactured to NEMA standards use grease-lubricated ball bearings with high radial and thrust load capacities. They are axially pre-loaded to eliminate any radial or axial play for quiet operation and long life. Most motor end frames include an outer race locking plate on the shaft end bearing to prevent race rotation due to output shaft loads. These bearing mounting features are required for high performance, long life, and high efficiency operation. Most motor manufacturers provide their larger frame sizes with grease fittings for relubrication during the lifetime of the motor. The smaller frames use bearings that are grease-packed and sealed for life at the factory and cannot be relubricated.

NEMA motors subject to very high loads and operating temperature in the larger frames may require oil lubrication to the rolling element bearings. This can be by either oil circulation within the bearing frame or from a pressurized lubrication system similar to that used with sleeve bearings. In addition, for motors required to carry very high thrust loads, quite often in only one direction, taper or spherical roller thrust bearings may be used.

Hydrodynamic Thrust Bearings Certain types of pump applications, such as very large vertical pumps, exhibit very high thrust loads that cannot be accommodated by rolling element bearings. Hydrodynamic bearings, usually with tilting, self-leveling thrust pads, are used for very high thrust loads. These bearings are sometimes referred to as "Kingsbury-type" in recognition of the original manufacturer of this bearing type. This type of bearing is oil-lubricated from a self-contained oil sump or an external pressurized pumping system, depending on size and rating.

MOTOR INSULATION

Insulation systems must be used in a motor to electrically insulate the windings from the mechanical parts of the motor as well as to insulate the phase winding conductors from one another. Extra insulation is also required between any adjacent phase windings for 230 or higher voltage motors. In general, the higher the operating voltage, the better the insulation system must be.

All NEMA-designed electric motors are manufactured with cord laced end turns to assure that they are positioned so they cannot touch any mechanical parts. In addition, the phase windings are nearly always impregnated with a varnish by either dipping or trickling into the heated end turns. The purpose of this is to secure the conductors to prevent them from vibrating between one another, which would wear through the wire insulation

and cause turn-to-turn shorts. This stator varnish also provides some additional insulation protection to the system. For example, this coating makes the windings resistant to ambient conditions such as moisture. Consequently, a motor insulation system is very complex, utilizing several different materials, parts, and processes to effectively insulate the windings. The parts considered in an insulation system include slot cells, phase barriers, conductor insulation, slot wedges, end turn supports, tie material, and winding impregnation material.

Since the introduction of PWM inverters for use with all types of variable speed drives, such as ac induction, PM brushless and switched reluctance brushless, a new insulation system failure mode has emerged. This is the most significant problem in installations where the inverter is located 50 ft (15 m) or more away from the motor. The long motor lead cables used in these applications cause very high voltage spikes not present at the inverter end of the lead cables. These voltage spikes across the first turns of each phase winding result in the degradation of the wire insulation due to the corona insulation failure. Special magnet wire insulation and motor manufacturing methods are available to overcome this problem. Therefore, if an inverter is to be used to drive a pump motor with long connection cables, an "inverter duty" motor should be specified.

There are four basic classes of insulating materials currently recognized by the motor industry. Each differs according to its physical properties and can withstand a certain maximum operating temperature (frequently termed total *temperature* or *hot-spot temperature*) and provide a practical and useful insulation life. The insulation classes and their maximum operating temperatures are

Class A	90°C
Class B	130°C
Class F	155°C
Class H	180°C

Those factors that contribute to the maximum operating temperature of a motor insulation system are the ambient temperature, the temperature rise in the motor winding caused by motor losses, and any overload allowance designed into the motor (service factor).

The current standard for motors within the range of NEMA ratings (frames 140T to 449T) requires nameplate marking for the maximum allowable ambient temperature, the power rating, the associated line current needed to develop this power, the class of insulation used, and the service factor provided. Motors larger and smaller than NEMA frame sizes have nameplate marking for maximum allowable ambient temperature, temperature rise in degrees Celsius either by thermometer or resistance measurement, power rating, line current needed to develop this rating, and any service factor provided.

In addition to ambient temperature, there are several additional environmental conditions that must be considered when applying an electric motor. In applications where chemical fume or moisture levels are abnormal and can cause decomposition of an insulation system, standard insulation will be inadequate. These applications require a motor with a premium insulation system that will incorporate highly resistant components and may include special impregnation techniques. Chemical fumes and moisture can also be destructive to the mechanical parts of a motor, and special protective treatment should be provided to these parts. NEMA frame-size motors have a special motor for chemical industry applications that has standard features to resist these environmental factors. For example, these motors usually contain shafts made from stainless steel.

Applications with excessive vibration can destroy a winding and damage the mechanical parts of a motor. In such cases, it is advisable to provide (1) extra treatment for the winding to ensure that it is rigid and will not vibrate and chafe the insulating materials and (2) a mechanical construction that will have the strength to withstand the above.

If abrasive dust is present, the motor insulation should be protected with a resilient surface coating to withstand the impact of the abrasive particles.

Because all insulation systems employ components that can in some degree support fungus growth in tropical locations, motors applied in such areas should incorporate fungus-proofing treatment on the insulation.

Obviously, applications exhibiting a combination of any or all of the environmental conditions discussed should have special protection for each condition. Applications in unusual environmental conditions may require special protection and should be discussed with the motor manufacturer or distributor.

COUPLING METHODS FOR PUMP APPLICATIONS

Direct Coupling Pumps are frequently directly coupled to motors, and where the pump is not close-coupled, it is usually coupled by means of a flexible coupling. The use of a flexible coupling permits minor misalignment (angular and parallel) between motor and pump shafts. The use of some older style flexible or solid couplings could cause severe radial and axial loads on the motor bearings. Since the development of the flexible disk couplings, the earlier coupling designs have largely disappeared, as have the severe loads resulting from misalignment. The flexible disk coupling is capable of transmitting very high torque for its size, with minimum radial and axial forces on the shafts resulting from misalignment. [See Section 9.3]

Close-Coupling Close-coupled pumps have become very popular for certain applications. In this arrangement, no coupling is provided between the pump and motor shafts and the pump housing is flange-mounted between close-tolerance fits on both the motor and the pump flanges. The pump impeller is mounted directly on the motor shaft. Care must be taken in this arrangement to ensure that the motor shaft runout or axial movement plus machine tolerances do not cause interference between the pump housing and its rotor. This is usually not a problem if properly fitted ball bearings are used in the motor. The motor shaft material must be compatible with the fluid being pumped, and if the pump impeller is held in place by a nut, the threat must respect the rotation of the motor. High-pressure close-coupled pumps of a nonbalanced design can cause excessive shaft thrust, which may be incorporated in the motor bearing capacity. It is always good practice with close-coupled pumps to provide some form of flinger on the motor shaft to prevent liquids that leak past the pump seal from entering the motor bearing.

Flanged Motors Flanged motors allow an easy means of aligning pump housings with motors. This construction is usually in the form of a vertical mounting in which the motor is set on top of the pump and the pump supports the motor weight. The pump and motor shafts are normally coupled, and those comments made under the subject of direct-coupling methods are applicable. Also, as in the case of coupled pumps, this construction permits thrust forces that must be considered when selecting a motor if the pump does not have a thrust bearing.

A further extension of flange motors includes the vertical hollow-shaft motor. With this design, a variable length of shafting connects the pump and motor. The pump shaft passes through the center of the motor bore, and the motor torque is imparted to the pump shaft by a suitable coupling at the top of the motor. The weight of the shaft and the pump impeller and the force of the hydraulic thrust are assumed by the motor bearings.

The coupling on the motor can be made a *self-release coupling* (Figure 17) to prevent the motor from delivering torque to the shaft in the event the motor is started in the wrong direction and to prevent reversed motor rotation from unscrewing the threaded joints between lengths of pump shafting.

Another modification to a coupling is a nonreverse *ratchet* (Figure 18), which prevents the remaining head of liquid in a pump from rotating the pump in the reverse direction when the pump is stopped. This prevents possible overspeeding of the pump and motor when the pump is connected to a large reservoir and most of the total pump head is static. This also prevents a pump with a long discharge column pipe from running in reverse with no liquid in the upper portion of the pipe to lubricate the line shaft bearings. Starting a pump capable of back spinning is also prevented.

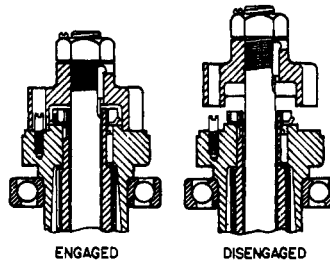


FIGURE 17 Self-release coupling connecting pump head shaft to hollow shaft of vertical motor disengages as a result of pump shaft couplings unscrewing (U.S. Electrical Motors)

PERFORMANCE

Ac induction motors are designed to produce their rated power at a certain speed at a specific line voltage, line frequency, and ambient temperature. These motors will also operate at a specific efficiency and power factor when all of these conditions are met. The normal operating conditions of a motor are stipulated on its nameplate with values for power,

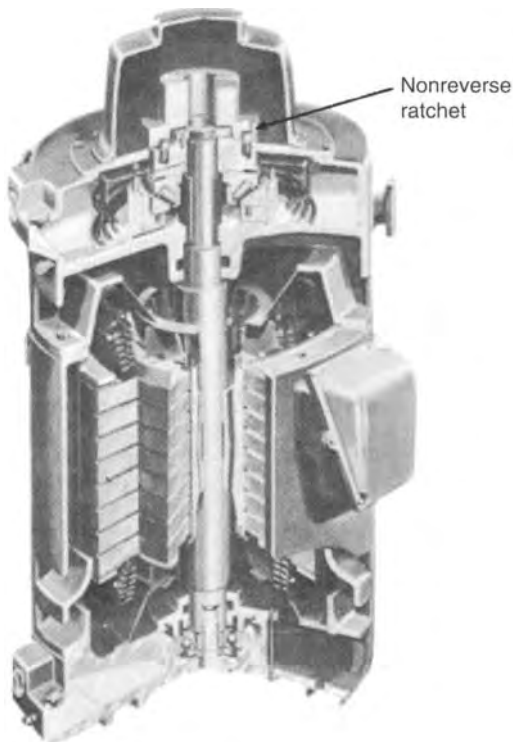


FIGURE 18 Section of vertical hollow shaft motor showing nonreverse ratchet. Spring-loaded pins ride on ratchet plate in one direction only (General Electric)

speed, ambient temperature, and frequency. If the operating conditions are different from the nameplate ratings, the motor performance will be altered. Many of the newer PM and SR brushless motors are designed to produce their rated output power over a wide speed range. A high efficiency ac induction motor can also perform similarly if a Flux Vector Inverter is used for control.

Voltage Ac motors are designed to operate satisfactorily at their nameplate voltage rating with a $\pm 10\%$ variation from nameplate voltage when operating at rated nameplate frequency. This dictates that the motor will develop rated power and speed to a pump and will operate at a safe insulation temperature over the range of voltage. The motor torque will vary directly with the square of the applied voltage divided by the nameplate voltage. This affects the peak torque of the motor and will cause the motor speed-torque curve as shown in Figure 2 to be altered. Within the $\pm 10\%$ voltage band, a motor can be expected to accelerate and operate a pump safely and continuously. A motor should never be expected to operate continuously beyond the $\pm 10\%$ band. If the voltage varies more than $\pm 10\%$, the pump and motor may not operate satisfactorily.

For example, assume a pump is operated by a NEMA design B motor (refer to Figure 2) that will produce 200% pull-out torque at rated voltage. If the line voltage were to fall to 70% of the rated name-plate voltage, the motor would produce only 49% of its peak torque value. The pull-out torque of the motor would then become $0.49 \times 200 = 98\%$ of rated torque. It then becomes very doubtful whether the motor will be able to sustain the pump load, and the motor can be expected to lose speed, stall, or become overloaded. Certainly, it will also heat up, which will shorten the expected life of the bearing lubrication and winding insulation system.

In a similar sense, a motor may be unable to accelerate a pump if low line voltage exists. In the example previously discussed, this same motor develops 150% of rated torque when started at zero speed and rated voltage. If the line voltage is again 70% of nameplate voltage, the motor will develop $0.49 \times 150 = 73\%$ of rated torque. This may be a problem with certain types of pumps, such as a constant displacement pump. It is conceivable that this would not be a problem in starting a centrifugal pump because of its square-law speed-torque characteristics. If the motor voltage never increased beyond 70%, the centrifugal pump would not reach normal operating speed. An exception to this rule is the commutating ac motor, for which a $\pm 6\%$ voltage variation is allowable.

Varying motor voltage from nameplate rated voltage will also affect the motor operating speed, power factor, and efficiency established for rated voltage and load. Most polyphase ac induction motors will operate at several rpm faster than nameplate speed at 10% over voltage and several rpm below nameplate speed at 10% under voltage. The speed of a synchronous motor is determined by the line frequency of the alternating voltage not the voltage level. Therefore, voltage variation has no effect on the speed of a synchronous motor. However, voltage variation does effect maximum torque output of a synchronous motor.

Dc motors can also be operated over a $\pm 10\%$ voltage range from nameplate rated voltage. However, it should be recognized that different types of dc motors will have different speed and torque characteristics over the voltage range. This should be taken into account when meeting pump performance requirements.

One of the most common applications for dc motors has been for metering pumps because of their ease of setting the speed to precisely meter the fluid being pumped. Although a voltage variation would change the pump speed, it is customary to include a speed control loop with the dc supply to maintain constant speed in spite of line voltage variations.

Frequency Ac induction motors will operate satisfactorily at rated load and voltage with a frequency variation up to $\pm 5\%$ from rated nameplate frequency. However, the speed of the motor will vary almost directly with the line frequency. The speed of a synchronous motor will vary directly with applied frequency. The combined variation of voltage and frequency must not be more than $\pm 10\%$ from rated nameplate voltage and frequency, provided the frequency variation does not exceed $\pm 5\%$ from rated nameplate frequency. The reason is that frequency variation from the nameplate frequency will cause motors to

operate at a power factor and efficiency other than those established for rated frequency. Any electric motor such as PM or SR brushless and even ac induction, which is powered from an inverter, is unaffected by frequency variations.

Speed and Speed Range Synchronous and induction motors were originally expected to be operated at one specific speed when powered from line voltage and frequency. However, with the availability of inverters, this is no longer the case. In fact, the use of inverters for driving induction motors is expanding at a fast pace as the merits and benefits of adjustable speed are being realized. The use of speed controlled PM and SR brushless motors along with their inverters and controllers allow the pump application to take advantage of the usefulness of adjustable speed. Using the inverter with any of these types of motors eliminates any difficulties from speed variations from the past. This is because most inverters have a speed control loop that will automatically maintain the set speed.

Acceleration A motor must be capable of accelerating as well as driving a pump at rated speed and power. The acceleration can be analyzed by examining a typical pump-motor combination involving a centrifugal pump driven by a six-pole squirrel-cage induction motor rated 10 hp (7.46 kW) with NEMA design B torque characteristics. The curve in Figure 19 demonstrates this combination when the pump is loaded and the motor is operating at nameplate frequency and voltage. It will be noted that the torque produced by the motor at any speed must be greater than the torque required by the pump when up to speed. The excess torque at any speed is available to accelerate the motor, coupling, and pump rotating parts.

NEMA has provided a handbook for motion control that outlines the fundamental equations for the required torque calculations to accelerate any inertial loads to a predetermined speed. For the use of ac induction motors that are line-fed with constant voltage and frequency, the formulas are used to calculate the time to the motor slip from synchronous speed for a given total rotating inertia. For the use of inverter-fed ac induction, PM or SR brushless motors, the formulas are used to determine the torque required to accelerate the total rotating inertial load to the desired speed in the desired time.

The fundamental equations for motion, which include torque, inertia, time and acceleration relationships, are given below. They can be used for motor selection analysis for pumps. For example, if the torque available from an induction motor is known, the equation can be solved for the acceleration time of the system inertia (from NEMA "Programmable Motion Control Handbook").

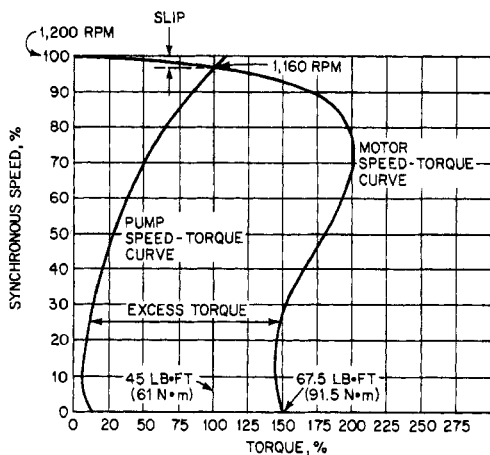


FIGURE 19 Typical speed-torque characteristic curves for a centrifugal pump and a squirrel-cage induction, NEMA design B motor. Excess torque accelerates pump.

$T_{TOTAL} = T_A + T_C$: Total torque = Acceleration torque + constant torque

$T_A = J_{TOTAL} \times \alpha$: Acceleration torque = Inertia $\times$ Angular acceleration

$\alpha = \frac{\omega_{max}}{t_a} \times 2\pi$: Angular acceleration = Max RPM/acceleration time

T_C = Torque from all other constant forces, friction, windage, preload, and so on.

NOTE: Use consistent units depending on whether U.S. Customary or SI.
The time to accelerate the pump is equal to

$$\text{in USCS units} \quad t = \frac{WK^2 \times \Delta \text{rpm}}{308T}$$

$$\text{in SI units} \quad t = \frac{MK^2 \times \Delta \text{rpm}}{9.55T}$$

where t = time, s

WK^2 = total weight (force) moment of inertia, lb $\cdot$ ft²

Δrpm = change in speed

T = torque, lb $\cdot$ ft ($N \cdot m$)

MK^2 = total mass moment of inertia, kg $\cdot$ m

Because the difference in torque is not uniform over the speed range, the curve (Figure 19) can be analyzed by assuming discrete changes in speed and an average torque over this change in speed. A time period can be calculated for each discrete speed change, and all time values can be totaled to obtain the complete acceleration time.

Increasing the inertia of the pump, the operating speed of the pump and motor, or the torque required by the pump at any speed will result in a longer acceleration time, which may not be possible for the motor. Each motor can operate at a reduced speed and at a torque in excess of rated torque for a given time. Beyond this time, the windings and/or rotor can be damaged. The use of an inverter-driven ac induction SR or PM brushless motor eliminates all of these problems associated with the use of line-fed ac induction motors. This is due to the torque control capability of the inverter/motor drive system using current control. The limit on the ability of the adjustable speed drive to control acceleration is a function of the current capability of the inverter (as long as the thermal limit of the motor has not been exceeded).

When an application is analyzed and found to present an acceleration-time problem, consideration should be given to

1. Unloading the pump during acceleration
2. Reducing the inertia of the rotating parts
3. Selecting a larger motor
4. Considering an adjustable speed motor and drive

When very large pumps are started, the line voltage will sometimes drop because of the high starting current, which causes the motor torque to be reduced by the square of the voltage ratio. Naturally, the acceleration time is greater because of the reduced torque produced by the motor. This will not hamper the ability of the motor to accelerate the pump as long as the motor develops more torque than is required to drive the pump at any speed over the accelerating range. The use of SR or PM brushless induction motors eliminates this problem because these motors do not draw high inrush currents. In fact, the starting current is controlled very precisely.

A similar analysis can be made for synchronous motors after the accelerating speed-torque curve for the motor is known. It should be recognized that a synchronous motor operates as a squirrel-cage induction motor up to the moment of synchronization. At that time, the synchronous motor must have an additional capability of synchronizing torque,

frequently called *pull-in torque*, to accelerate the motor from subsynchronous to synchronous speed. If the motor cannot develop enough torque at its synchronous speed, it will pull out or stall. When this type of motor is used for a pump, it is advisable to unload the pump during acceleration operation to reduce the torque required from the motor for acceleration.

When using a synchronous motor, attention should always be given to the breakaway torque that is required to start the pump from zero speed. This is particularly important with constant displacement pumps, where the pump will operate at a constant torque over the entire accelerating speed range. Again, a better selection might be an inverter-driven ac induction motor, SR or a PM brushless motor.

Service Factor Motors are available with service factor ratings that range from 1.0 to as high as 1.5. A service factor implies that a motor has a built-in thermal capacity to operate at the nameplate power times the service factor stamped on the nameplate. It should be noted, however, that when the motor is operated at the service factor power, the motor will operate at what is termed a *safe temperature*. This means that the motor will operate at a total temperature that is greater than the temperature for a motor designed for the same power with a 1.0 service factor. Consequently, it is not advisable to apply a motor with a service factor larger than 1.0 where the continuous power requirements will be greater than the normal power.

A service factor rating on a motor is to provide an increased power capacity beyond nominal nameplate capacity for occasional overload conditions. Also, the speed-torque characteristics are related to the nominal power rating and not the service factor power. When adjustable speed motor/drive systems are selected, the old service factor ratings are not useful. The supplier of the drive system will assist in selecting a thermally rated system that will satisfy the pumping operation. All inverters are equipped with thermal safety protection for both the motor and the drive so the system cannot be overloaded to minimize overheating. This is a very useful feature for many pump applications.

Efficiency Motors are designed to operate with an efficiency expressed in percent at rated voltage, frequency, and power. Efficiency is defined as

$$\text{Efficiency \%} = \frac{\text{shaft output power} \times 100}{\text{electrical input power}}$$

The efficiency of a given motor design will vary slightly from unit to unit because of manufacturing tolerances and variations in materials. For this reason, guaranteed efficiencies are usually lower than actual efficiencies. When motors are operated at reduced powers, the tendency is for the efficiency to decrease because the losses tend to be fixed.

Several other factors have an effect on motor efficiency. Increasing the applied voltage and operating a motor at its rated power will increase efficiency very slightly, whereas decreasing applied voltage will decrease the efficiency noticeably. Also, increasing the frequency will cause a very slight increase in efficiency, and decreasing the applied frequency will cause a slight decrease in efficiency.

Inverter-fed ac induction and PM or SR brushless motors are not subject to changes in their efficiencies due to changes in line voltage or frequency because the ac line voltage is rectified into a dc voltage before the regeneration of the power to the motor. The dc voltage is always reasonably constant and unaffected by changes in the line power. This of course assumes that there is sufficient voltage and power headroom in the design.

Dynamometers are used to determine efficiency for small motor ratings—up to approximately 500 hp (373 kW)—and standard methods and formulas are used for calculating efficiency for large motors. For the latter, efficiency values can vary, depending upon which “standard” procedure is used. Several suppliers market torque and power analyzers for induction motors that base their output on these formulas using current data measured from the motor. These methods are accurate enough to be acceptable. They have been reasonably verified by applying the method to smaller motors that have been tested on a

dynamometer. NEMA has established one method, but some suppliers use different methods, depending upon their national standards or established practices.

Higher efficiencies have been designed into most new motors by selecting materials and proportions that reduce losses, but such design choices usually make the motor more costly. This also includes filling the stator slots with more copper of a larger wire gage to reduce ohmic losses. These new motors designed for higher efficiencies offer improvements from 2% to as high as 7% depending upon the size and manufacturer.

Motor operating efficiency in a typical application often may not be that described by the manufacturer. Quoted efficiencies are always at rated power output, rated frequency, and rated voltage. However, it is almost universal in application that a motor is oversized for its applied load and frequently operates at other-than-normal voltage. Both of these variables can greatly reduce the efficiency of a motor. However, when a motor is inverter-driven, the efficiency of the system can be much better controlled.

Power Factor The power factor of a motor is expressed as

$$PF = 100 \cos \theta$$

where θ is the angle between voltage and current at motor terminals (leading or lagging).

The power factor at which a motor operates is dependent on the design of the motor and is established at rated voltage, frequency, and power output.

For induction motors, the power factor can never be 100% leading. A number of factors will influence the power factor of an induction motor:

Condition	Effect on power factor
Increase applied voltage	Decrease
Decrease applied voltage	Increase
Increase load	Increase
Decrease load	Decrease
Increase applied frequency	Slight increase
Decrease applied frequency	Slight increase

In synchronous motors, it is usual to use two varieties of motors, the 100 (unity) and 80% leading motors. The power factor of a synchronous motor operated at rated voltage and frequency is fixed by its field excitation and its power output. At a given power output, the power factor can be adjusted over a range by adjusting the field excitation. Increasing the field excitation will cause the motor to operate at a more leading power factor, and, conversely, reducing the field excitation will make the power factor lag.

Varying the power output of a synchronous motor with a constant field excitation will vary the operating power factor. A decrease in power output will cause a more leading power factor; conversely, increasing the power output will induce operation at a lagging power factor. Consequently, to operate the motor at rated power factor with a varying output power, it is necessary to adjust the field excitation. However, this is not normally done because a synchronous motor is frequently used for improving the power factor and the more leading power factor is used to accommodate power factor improvement. When a synchronous motor is overloaded and operates at a more lagging power factor, it is not usual to increase the excitation beyond its rated excitation because of the extra heating this will develop in the motor. In this case, the more lagging power factor is simply accepted.

The power factor is not of much concern for SR and PM brushless motors driven by inverters because of the control capabilities for these systems. The PM machine exhibits the best power factor of any known motor under nearly all operating speeds and power levels except if excessive field weakening or phase advance is used.

TYPES OF CONTROLS

In order to use an electric motor to drive a pump, a means of starting and stopping the motor is required. The devices used for this purpose can be called controls. Only small—perhaps up to two or three hp (kW)—motors can be controlled with a simple on-off switch (starting from full rated line voltage). This is because when a motor is first started it produces no back EMF and will draw very high starting currents. If the load inertia is very high, the motor can overheat during the long acceleration time. In addition, the high starting current peaks can cause heavy dips in the line voltage. Therefore, for larger motors, these line voltage dips must be minimized by the use of starters. There are four basic ways to start ac induction motors:

1. Direct online starting
2. Low voltage starting
3. Rotor resistance starting
4. Low frequency starting

Other devices used to control motors such as *soft starters*, *adjustable frequency controls*, and *vector drives* are discussed along with the inverters and controls for other motor types such as dc motors, SR and PM brushless in Section 9.2.2.

Alternating-Current Motor Starters

MANUAL STARTERS Manual motor starters are designed to provide positive overload protection and start and stop control of single-phase and polyphase motors up to about 3 HP (2.5 kW). A single manual toggle switch or a push button switch is used for “on,” “off,” and “tripped” states. These types of starters use full line voltage applied to the motor.

MAGNETIC STARTERS Magnetic motor starters cost more and are more reliable than manual starters. They are designed to control a motor by incorporating a magnetically operated contactor to apply power to the motor terminals rather than allowing full current to pass through the contacts.

An overload relay is incorporated to protect the motor from overloading. Magnetic starters are available for reversing and nonreversing service and are also made as non-combination or combination types. The non-combination starter combines only the motor contactor and overload relay. The combination starter combines these parts along with either a circuit breaker or a fused switch to provide shortcircuit protection.

REDUCED-VOLTAGE STARTERS Reduced-voltage starters are available in several types and are basically magnetic starters with additional features to provide reduced voltage, which in turn provides for reduced motor-starting current or torque. These starters include the types discussed in the following paragraphs.

PRIMARY-RESISTOR STARTERS Primary-resistor starters, sometimes known as *cushion-type* starters, will reduce the motor torque and starting inrush current to produce a smooth, cushioned acceleration with closed transition. Although not as efficient as other methods of reduced-voltage starting, primary-resistor starters are ideally suited to applications where reduction of starting torque is of prime consideration. A typical diagram for this type of starter is shown in Figure 20.

AUTOTRANSFORMER STARTERS Autotransformer starters are widely used reduced-voltage starters because of their efficiency and flexibility. All power taken from the line, except

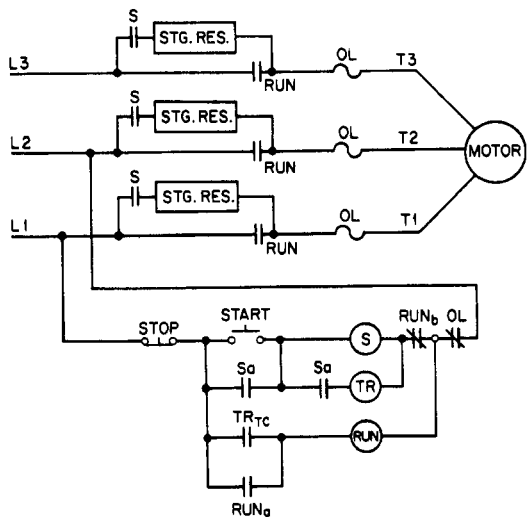


FIGURE 20 Typical primary-resistor, reduced-voltage starter wiring diagram (Westinghouse Electric)

transformer losses, is transmitted to the motor to accelerate the load. Taps on the transformer allow adjustment of the starting torque and inrush to meet the requirements of most applications. The following characteristics are produced by the three voltage taps:

Tap, %	Starting torque, % locked torque	Line inrush, % locked current
50	25	28
65	42	45
80	64	67

A typical diagram for this type of starter is shown in Figure 21.

Part-winding starting provides convenient, economical, one-step acceleration at reduced current where the power company specifies a maximum or limits the increments of current drawn from the line. These starters can be used with standard dual-voltage motors on the lower voltage and with special part-winding motors designed for any voltage. When used with standard dual-voltage motors, it should be established that the torque produced by the first half-winding will accelerate the load sufficiently so as not to produce a second undesirable inrush when the second half-winding is connected to the line. Most motors will produce a starting torque equal to between one-half and two-thirds of NEMA standard values with half the winding energized and draw about two-thirds of normal line-current inrush. A typical diagram is shown in Figure 22.

Star-delta starters have been applied extensively to starting motors driving high-inertia loads with resulting long acceleration times. They are not, however, limited to this application. When 6 or 12 lead delta-connected motors are started star-connected, approximately 58% of full-line voltage is applied to each winding and the motor develops 33% of full-voltage starting torque and draws 33% of normal locked rotor current from the line.

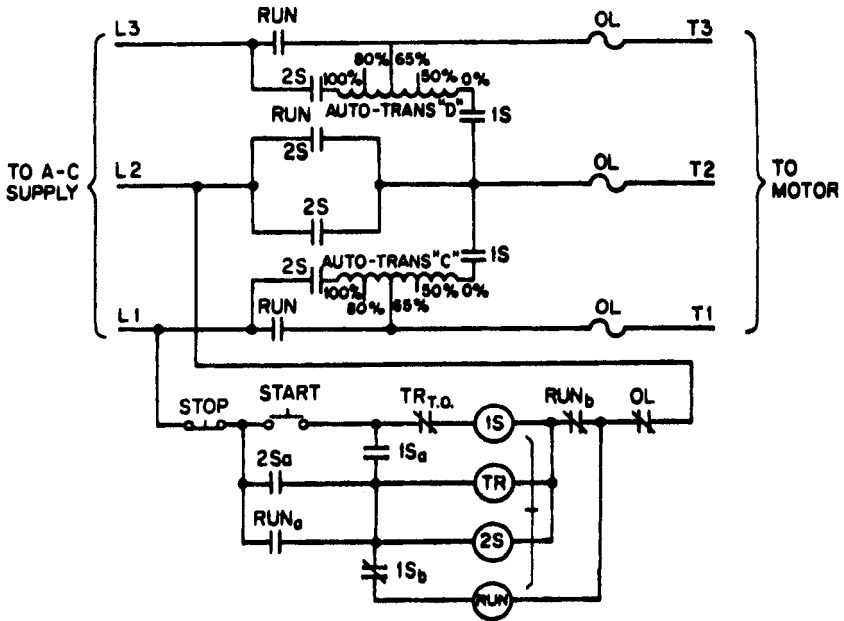


FIGURE 21 Typical autotransformer, reduced voltage starter wiring diagram (Westinghouse Electric)

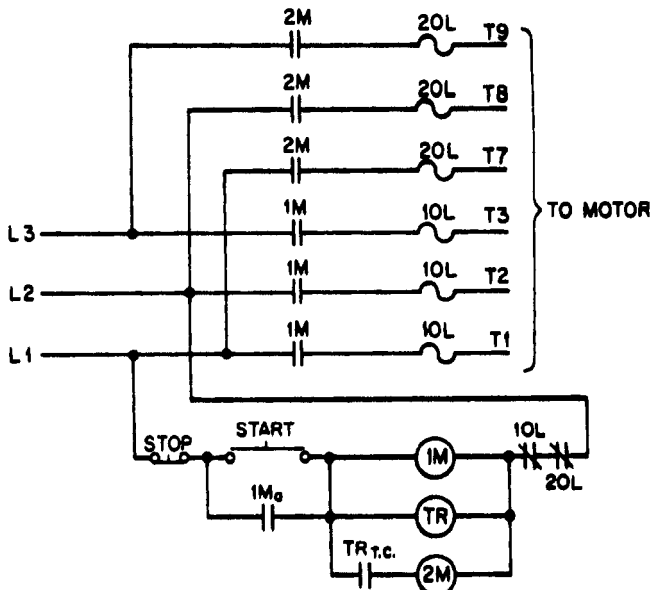


FIGURE 22 Typical part-winding starter wiring diagram (Westinghouse Electric)

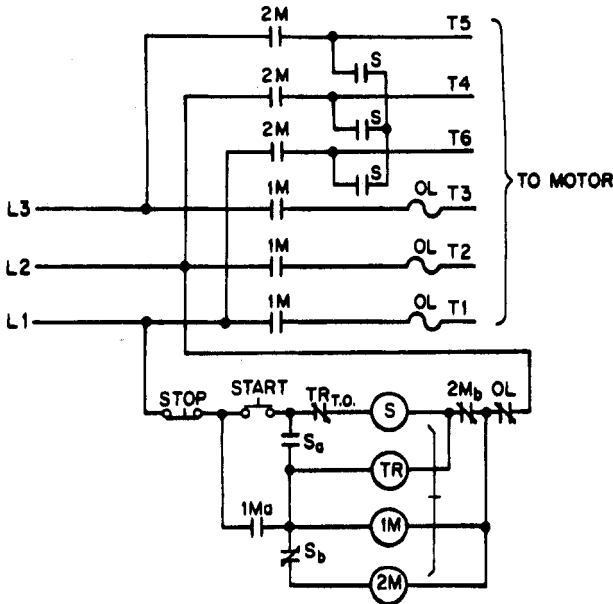


FIGURE 23 Typical star-delta, reduced-voltage starter wiring diagram (Westinghouse Electric)

When the motor has accelerated, it is reconnected for normal delta operation. A typical diagram is shown in Figure 23.

WOUND ROTOR MOTOR STARTERS These magnetic motor starters are used for starting, accelerating, and controlling the speed of wound-rotor motors. The primary control includes overload protection and low-voltage protection or low-voltage release, depending on the type of pilot device. Disconnect switches, circuit breakers, and reversing can be added to the primary circuit when required. Reversing starters are not designed for plugging.

The secondary circuit contains the NEMA recommended number of accelerating or running contactors and resistors to allow approximately 150% of motor full-load torque on first point of acceleration. Additional accelerating points can be added for high-inertia loads or exceptionally smooth starts. Adjustable timing relays permit field adjustment. Standard starting duty NEMA 135 resistors allow 10 s starting out of every 80 s. A typical diagram is shown in Figure 24.

SYNCHRONOUS MOTOR STARTER Synchronous, magnetic, full-voltage starters provide reliable automatic starting of synchronous motors. They can be used whenever full-voltage starting is permissible. Automatic synchronization is provided by field relay, which assures application of the field at the proper motor speed and at a favorable angular position of stator and rotor poles. As a result, line disturbance resulting from synchronization is reduced and effective motor pull-in torque is increased. A typical diagram is shown in Figure 25.

Brushless synchronous-motor starters require special consideration inasmuch as all brushless synchronous motors are not constructed in the same way. The usual starter incorporates a low-power adjustable dc excitation source to energize and control the output of an integral shaft-connected exciter.

In addition, a pull-out relay and a timing relay are incorporated to initiate synchronization and stop the motor in event of pull-out.

Direct-Current Motor Starters Direct-current motor starters are designed to apply normal voltage to the motor field and, by means of a resistor, reduce voltage to the armature.

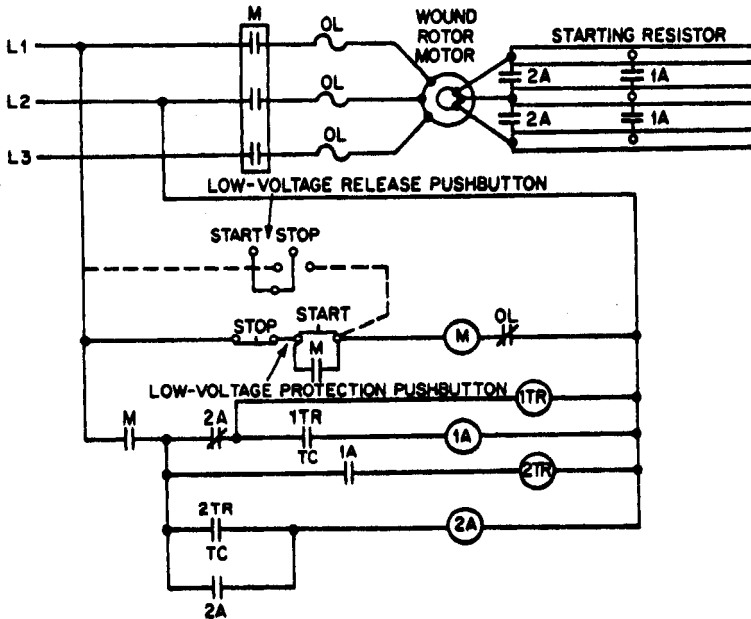
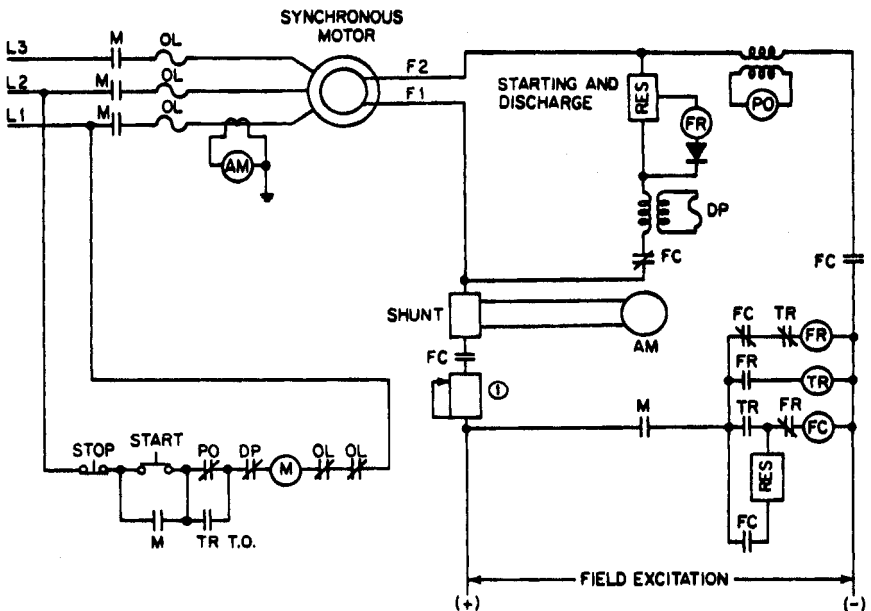


FIGURE 24 Typical wound-rotor motor starter wiring diagram (Westinghouse Electric)



① FIELD RHEOSTAT OR PERMANENT RESISTOR WHEN USED

FIGURE 25 Typical synchronous-motor starter wiring diagram (Westinghouse Electric)

Timing relays and contractors progressively short out the resistor until full voltage is on the armature.

If the motor is equipped with a field rheostat for field-range speed adjustment, the starter will apply the preset field voltage as adjusted by the field rheostat after full voltage is applied to the armature.

These starters incorporate a field failure relay to deenergize in the event of field failure and overload relays to protect the motor against overspeed.

The use of dc drives or SCR voltage controllers includes the facilities for all required types of starting conditions for using dc motors for pump applications. Some of these issues are covered in Section 9.2.2.

SEALLESS PUMP MOTORS

Various centrifugal pump designs are available that require no shaft sealing; that is, no packing or mechanical seal. These pumps are completely leakproof, and some are submersible. Such pumps are used when leakage cannot be tolerated or when pumping conditions such as pressure and/or temperature make conventional sealing difficult, if not impossible.

The shaft seal is eliminated by joining the pump and motor housings together to create a single leakproof unit. The impeller and motor rotor are mounted on a single shaft. Most designs permit the pumped liquid to circulate through the motor rotor and motor bearings. However, there are designs where either circulation is also through the stator or the stator is sealed and filled with dielectric oil. See Section 2.2.3.2.

Another popular method used for smaller pumps is to separate the pump shaft completely from the motor shaft. A permanent magnet coupling is provided to transmit the motor torque to the pump shaft through a nonmagnetic shell or section of the pump housing. The pump and motor can be integrated in these designs. See Section 2.2.3.1.

Figure 26 classifies various motor designs and references other places in the handbook where the special features and applications of these units are described in more detail.

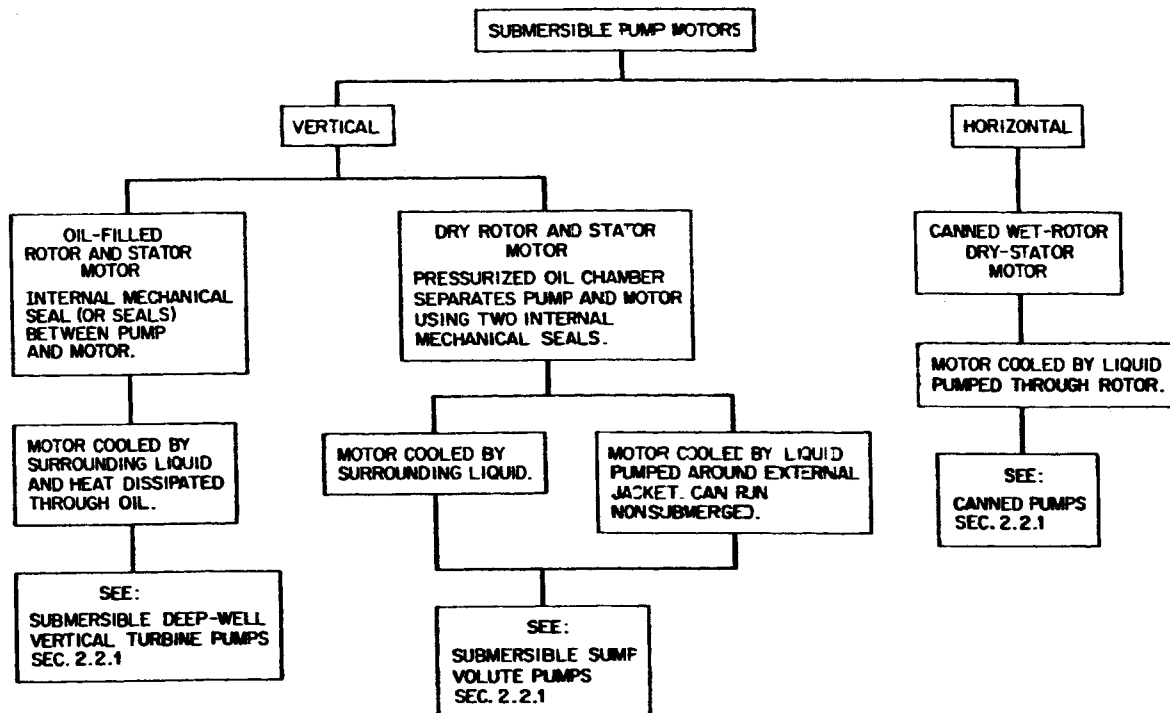


FIGURE 26A Pump motors for sealless centrifugal pumps: Submersible pumps

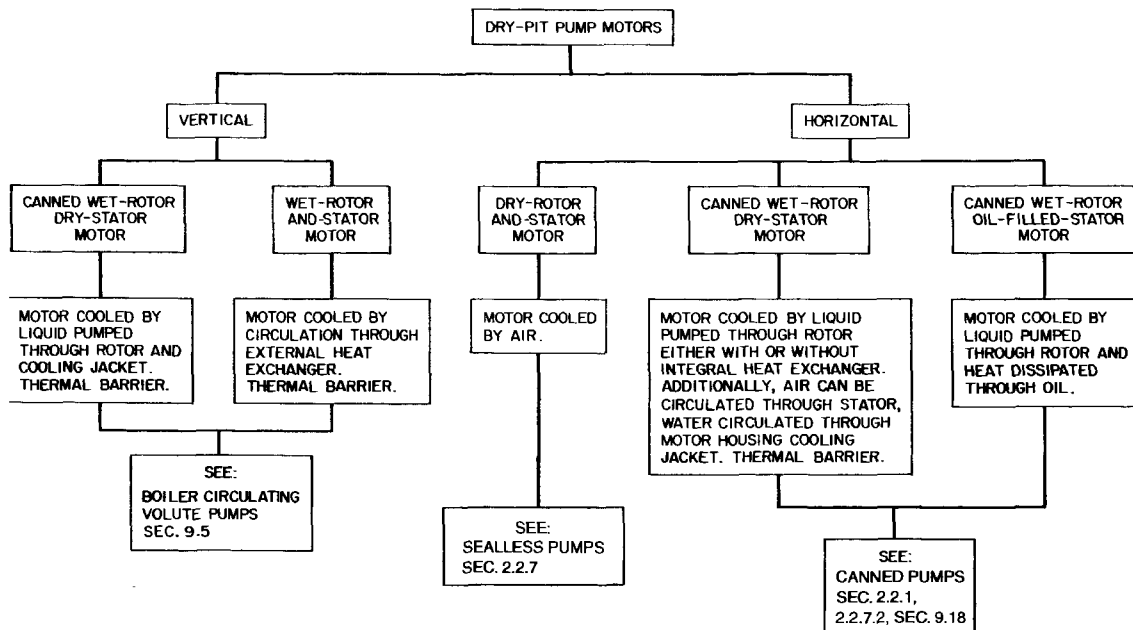


FIGURE 26B Pump motors for sealless centrifugal pumps: Dry pit pumps

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9.1.2 STEAM TURBINES

WALLACE L. BERGERON

DEFINITIONS

"A steam turbine may be defined as a form of heat engine in which the energy of the steam is transformed into kinetic energy by means of expansion through nozzles, and the kinetic energy of the resulting jet is in turn converted into force doing work on rings of blading mounted on a rotating part."¹

This definition may be restated:

"A steam turbine is a prime mover which converts the thermal energy of steam directly into mechanical energy of rotation."²

REASONS FOR USING STEAM TURBINES

Steam turbines are used to drive pumps for a variety of reasons:

1. The economical generation of steam often requires boiler steam pressures and temperatures that are considerably in excess of those at which the steam is utilized. Steam may also be used at two or more pressure levels in the same plant. Pressure reduction can be accomplished through valves, pressure-reducing stations, or use of a steam turbine.

Pressure reduction by using a steam turbine and thereby developing power to drive a pump permits lower utility costs. The incremental increase in steam flow and consequently in fuel costs for the same lower pressure steam heating load is in most instances less than the cost of purchased power for a motor-driven pump.

2. A pump driven by a steam turbine may be operated over a wide speed range, utilizing the turbine governor system or a separately controlled valve in the turbine or in the steam line to the turbine. Operation at variable speeds is an inherent characteristic

of steam turbines and does not require the use of special speed-changing devices, as is the case with other prime movers.

The overall efficiency of the turbine and pump unit can be optimized by operating at reduced speeds and at the resultant reduced power ratings. Pump performance can be controlled by reducing the speed of the pump rather than throttling it. Although the turbine efficiency normally declines when operating at a reduced speed, the steam flow will still be less than when the pump is throttled.

Operation at reduced power but at constant speed is also permitted by the speed governor, which throttles the steam to the nozzles as the power is reduced. Efficiency may be improved by equipping the turbine with auxiliary steam valves that are closed for reduced power operation. Closing these valves reduces the available nozzle area and reduces the pressure drop across the governor valve.

When the turbine is operated with the auxiliary steam valve closed, the steam flow will approximate that for the same turbine designed for the reduced rating.

3. The use of a steam turbine driver permits the driven pump to operate essentially independent of the electric power or distribution system. The steam turbine is not affected by electric power stoppages or interruptions and is therefore ideal for critical pumping operations.
4. A turbine may be used as a secondary driver for a pump; it may also drive an independent standby or emergency pump. The particular plant design may not afford sufficient steam for the pump to be normally driven by the steam turbine. However, in the event of an electric power failure or power system disturbance, a steam turbine may be employed as a dual drive or to drive a separate pump to assure continued operation of the plant until the electric power system is again operable.
5. The steam turbine controls—governor system and overspeed trip system—are inherently sparkproof. Consequently, steam turbines can be readily applied to drive centrifugal pumps in a wide variety of hazardous atmospheres without entailing additional cost for explosionproof or sparkproof construction.
6. Steam turbines can normally be readily altered to accommodate an increase in rating for increased pump output or for new pump applications. This inherent flexibility of a steam turbine also permits it to be readily altered to accommodate changes in the initial steam pressure and temperature and in the exhaust steam pressure at which the turbine operates.
7. Steam turbines have a starting, or breakaway, torque of approximately 150 to 180% of the rated torque. Additional starting torque can be readily furnished by designing the turbine for the additional required steam flow—and without reducing the efficiency at the normal operating rating by using an auxiliary steam valve. The additional starting torque can often be obtained without increasing the turbine frame size.
8. Steam turbines can be used to drive all types of pumps.
9. Steam turbines are inherently self-limiting with respect to the power developed. Special protective devices do not have to be furnished to prevent damage to the turbine because of overload conditions. The maximum power that can be developed by a turbine is a function of the flow areas provided in the design of the nozzle ring and governor valve. Application of a load greater than that which can be developed by the turbine causes the turbine to slow down to a speed at which the torque generated by the turbine matches that required by the pump.
10. When the pump application requires the driver to be designed with excess power or to permit operation of the pump “at the end of the curve,” the steam turbine can be designed for the corresponding rating without reducing the turbine efficiency when operating at the normal rating. Closing an auxiliary steam valve furnished for operation at the normal rating preserves efficiency because the turbine governor valve is not throttling to obtain the power rating.
11. With respect to the operation of the various types of pump drivers and their supporting systems, steam turbines afford minimum maintenance, low vibration, and a quiet installation.

TYPES OF STEAM TURBINES

Single-Stage Turbines A single-stage steam turbine is one in which the conversion of the kinetic energy to mechanical work occurs with a single expansion of the steam in the turbine—from inlet steam pressure to exhaust steam pressure.

A single-stage turbine may have one or more rows of rotating buckets that absorb the velocity energy of the steam resulting from the single expansion of the steam.

Single-stage turbines are available in wheel diameters of 9 to 28 in (22 to 71 cm). The overall efficiency of a turbine for a particular operating speed and steam conditions is normally dependent on the wheel diameter. The efficiency will generally increase with an increase in wheel size, and therefore the steam rate will be less (for the more usual speeds and steam conditions).

The larger-wheel-diameter steam turbines can be furnished with more nozzles to provide increased steam flow capacity and consequently greater power capabilities. The larger-wheel-diameter turbines are therefore furnished with larger steam connections, valves, shafts, bearings, and so on. Consequently the size of the turbine will generally increase with increases in power rating.

Multiple-Stage Turbines A multiple-stage turbine is one in which the conversion of the energy occurs with two or more expansions of the steam in the turbine. The number of stages (steam expansions) is a function of three basic parameters: thermodynamics, mechanical design, and cost. The thermodynamic considerations include the available energy and speed. The mechanical considerations include speed, steam pressure, steam temperature, and so on, most of which are material limits. Cost considerations include the number, type, and size of the stages; the number of governor-controlled valves; the cost of steam; and the number of years used as a basis for the cost evaluation.

The two factors generally used in selecting multistage turbines are initial cost and steam rate. Because these two factors are a function of the total number of stages, the application becomes a factor of stage selection. The initial cost increases with the number of stages, but the steam rate generally improves.

Multiple-stage turbines are normally used to drive pumps when the cost of steam or the available supply of steam requires turbine efficiencies greater than these available with a single-stage turbine, or when the steam flow required to develop the desired rating exceeds the capability of single-stage turbines.

Multiple-stage turbines can be furnished with a single or multiple governor valves. A single governor valve is often of the same design whether used in a single-stage or multiple-stage turbine and generally has the same maximum steam flow, pressure, and temperature parameters. Multiple valves are used when the parameters for a single valve are exceeded or to obtain improved efficiency, particularly at reduced power outputs.

Shaft Orientation Some steam turbines, particularly single-stage turbines, can be furnished with vertical downward shaft extensions. The application of such turbines can require considerable coordination between the pump and the turbine manufacturer to assure an adequate thrust bearing in the turbine, shaft length and details, mounting flange dimensions, and even shaft runout.

Vertical shaft pumps are frequently driven by horizontal turbines through a right-angle speed-reduction gear unit.

Direct-Connected and Geared Turbines Steam turbines can be directly connected to the pump shaft so the turbine operates at the pump speed or can drive the pump through a speed-reduction (and even speed-increase) gear unit, in order to permit the turbine to operate at a more efficient speed.

Turbine Stages The two types of turbine stages are impulse and reaction. The turbines discussed in this section employ impulse stages because steam turbines driving pumps normally have impulse-type stages.

In the ideal impulse stage, the steam expands only in the fixed nozzles and the kinetic energy is transferred to the rotating buckets as the steam impinges on the buckets while

flowing through the passages between them. The steam pressure is constant, and the steam velocity relative to the bucket decreases in the bucket passages.

In a reaction stage, the steam expands in both the fixed nozzles and the rotating buckets. The kinetic energy is transferred to the rotating buckets by the expansion of the steam in the passages between the buckets. The steam pressure decreases as the steam velocity relative to the buckets increases in the bucket passages.

In an impulse stage, the steam can exert an axial force on the buckets as it flows through the blade passages. Although this force is usually referred to as a reaction, the use of the term does not imply a reaction-type stage.

The larger buckets used in the last stages of an impulse-type multistage turbine can be of a free-vortex design—twisted and tapered. Such a bucket is ideally subjected to a nearly pure impulse force at its root and a nearly pure reaction force at its tip, but, in reality, this bucket is a high reaction-design bucket compared with a normal impulse-stage bucket. A steam turbine stage with such a bucket design is still referred to as an impulse stage because the primary conversion of kinetic energy is by a reduction rather than an increase in relative steam velocity.

A reaction turbine has more stages than an impulse turbine for the same application because of the small amount of kinetic energy absorbed per stage, and requires a larger thrust bearing or a balancing piston because of the pressure drop across the moving blades. The small pressure drop per stage and the pressure drop across the moving blades require that the steam-leakage losses be minimized by elaborate sealing between the tips of the nozzle blades and the rotor, and the tips of the moving blades and the casing.

The small pressure differential across the rotating blades of an impulse stage results in smaller thrust bearings and no close blade-tip clearances. Consequently, impulse turbines can be started more quickly without thermal-expansion damage, and their stage efficiencies remain relatively constant over the life of the turbine.

CONSTRUCTION DETAILS

Component Parts The main components of a single-stage steam turbine are shown in Figure 1.

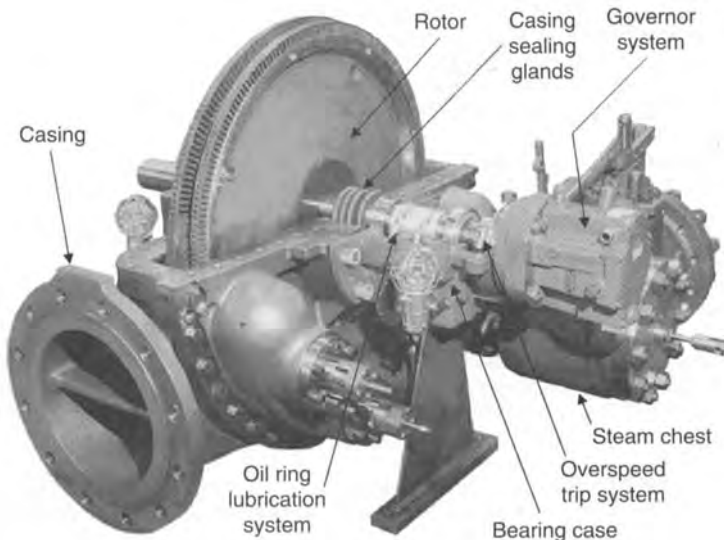


FIGURE 1 Component parts of steam turbines (Elliott)

Function and Operation—Single-Stage Turbine The *steam chest* and the *casing* contain the steam furnished to the turbine, being connected to the higher-pressure steam supply line and the lower-pressure steam exhaust line, respectively. The steam chest, which is connected to the casing, houses the governor valve and the overspeed trip valve. The casing contains the rotor and the nozzles through which the steam is expanded and directed against the rotating buckets.

The *rotor* consists of the shaft and disk assemblies with buckets. The shaft extends beyond the casing and through the bearing cases. One end of the shaft is used for coupling to the driven pump. The other end serves the speed governor and the overspeed trip systems.

The *bearing cases* support the rotor and the assembled casing and steam chest. The bearing cases contain the journal bearings and the rotating oil seals, which prevent outward oil leakage and the entrance of water, dust, and steam. The steam end bearing case also contains the rotor positioning bearing and the rotating components of the overspeed trip system. An extension of the steam end bearing housing encloses the rotating components of the speed governor system.

The *casing sealing glands* seal the casing and the shaft with spring-backed segmented carbon rings (supplemented by a spring-backed labyrinth section for the higher exhaust steam pressures).

The *governor system* commonly consists of spring-opposed rotating weights, a steam valve, and an interconnecting linkage or servomotor system. Changes in the turbine inlet and exhaust steam conditions, and the power required by the pump will cause the turbine speed to change. The change in speed results in a repositioning of the rotating governor weights and subsequently of the governor valve.

The *overspeed trip system* usually consists of a spring-loaded pin or weight mounted in the turbine shaft or on a collar, a quick-closing valve that is separate from the governor valve, and interconnecting linkage. The centrifugal force created by rotation of the pin in the turbine shaft exceeds the spring loading at a preset speed. The resultant movement of the trip pin causes knife edges in the linkage to separate and permit the spring-loaded trip valve to close.

The trip valve may be closed by disengaging the knife edges manually, by an electric or pneumatic signal, by low oil pressure, or by high turbine exhaust steam pressure.

The two usual types of lubrication systems are oil-ring and pressure. The *oil-ring* lubrication system employs an oil ring(s) that rotates on the shaft with the lower portion submerged in the oil contained in the bearing case. The rotating ring(s) transfers oil from the oil reservoir to the turbine shaft journal bearing and rotor-locating bearing. The oil in the bearing case reservoirs is cooled by water flowing in cooling water chambers or tubular heat exchangers.

A *pressure* lubrication system consists of an oil pump driven from the turbine shaft, an oil reservoir, a tubular oil cooler, an oil filter, and interconnecting piping. Oil is supplied to the bearing cases under pressure. The oil rings may be retained in this system to provide oil to the bearings during startup and shutdown when the operating speed and bearing design permit.

Typical sectional drawings are shown in Figures 2, 3, and 4.

GOVERNORS AND CONTROLS

Governor systems are typically speed-sensitive control systems. The turbine speed is controlled by varying the steam flow through the turbine by positioning the governor valve. Variations in the power required by the pump and changes in steam inlet or exhaust conditions alter the speed of the turbine, causing the governor system to respond to correct the operating speed.

Control systems, unlike governor systems, are not directly speed-sensitive but respond to changes in pump or pump-system pressures and then reposition the turbine "governor" valve to maintain the preset pressure. Consequently, changes in turbine steam conditions or in the power required by the pump result in a repositioning of the turbine governor valve or of a separate steam valve only after the pressure being sensed by the controller has changed.

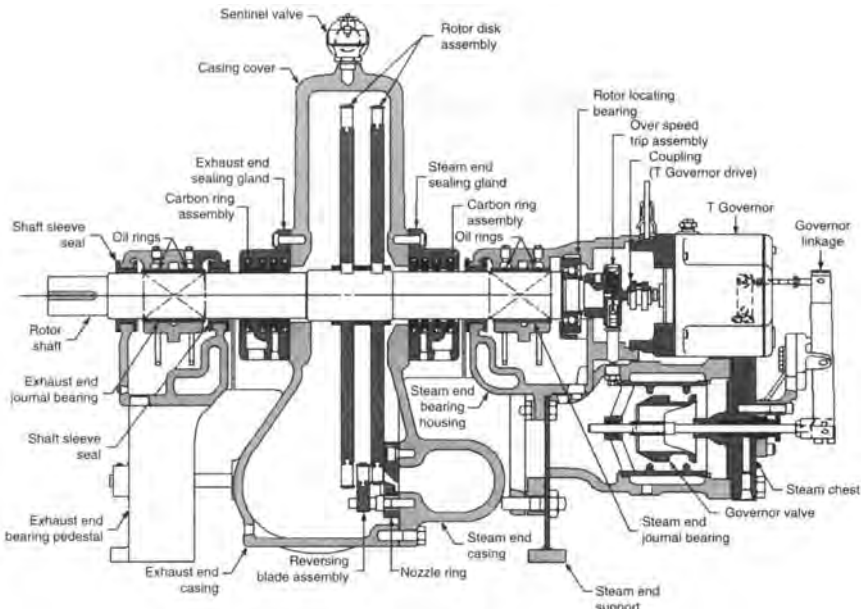


FIGURE 2 Section of single-stage turbine and governor system (Elliott)

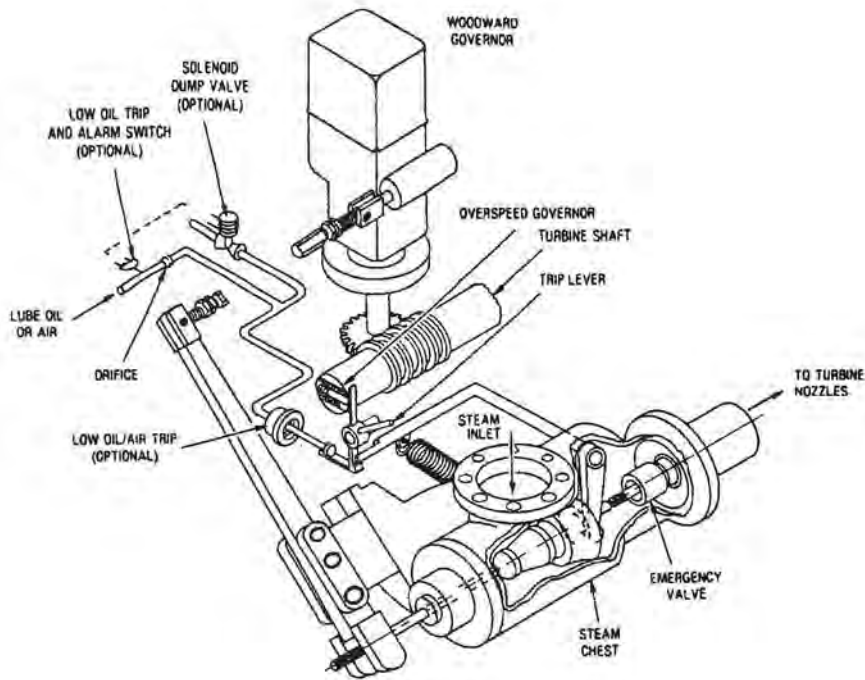


FIGURE 3 Typical turbine overspeed trip arrangement (Dresser-Rand)

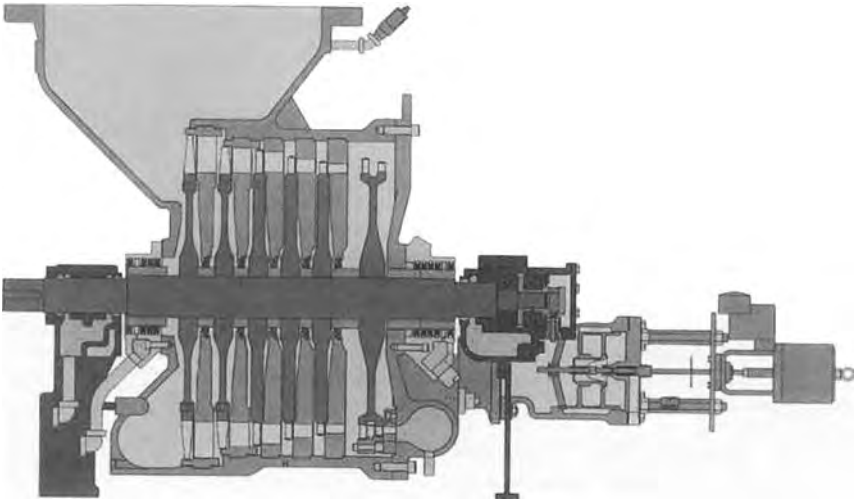


FIGURE 4 Section of multistage turbine (Elliot)

Even when a control system is furnished, a speed governor is also normally furnished. The speed governor is set for a speed slightly higher than the desired operating speed in order to function as a pre-emergency governor; that is, prevent the turbine from reaching the trip speed when the controller causes the turbine to operate at a speed above rated speed.

Governor systems are defined by their performance as follows²:

Class of governor system	Speed range, % (as specified)	Maximum speed regulation, %	Maximum speed variation, %, ±	Maximum speed rise, %
A	10-65	10	0.75	13
B	10-80	6	0.50	7
C	10-80	4	0.25	7
D	10-90	0.50	0.25	7

Speed range is the percentage below rated speed for which the governor speed setting may be adjusted. For example, a turbine with 4000 rpm rated speed and a governor system having a 30% range can be operated at a minimum speed of 2800 rpm:

$$4000 - \frac{30 \times 4000}{100} = 2800$$

If the speed range had been specified as plus 5% and minus 25%, for example, the maximum and minimum speeds would be 4200 and 3000 rpm.

Steady-state speed regulation is the change in speed required for the governor system to close the governor valve when the load is gradually reduced from rated load to no load with turbine steam conditions constant. Regulation is always expressed as a percentage of rated speed and calculated as follows:

$$\frac{\text{No-load speed} - \text{rated speed}}{\text{Rated speed}} \times 100 = \% \text{ regulation}$$

Therefore

$$\text{No-load speed} = \text{rated speed} \left(1 + \frac{\% \text{ regulation}}{100} \right)$$

A turbine with 4000 rpm rated speed and equipped with a NEMA A speed governor system would have 4400 rpm maximum no-load speed:

$$4000 \times \left(1 + \frac{10}{100} \right) = 4400 \text{ rpm}$$

Consequently, whenever the turbine is developing less than rated power, the operating speed will be greater than rated speed, as required for the governor system to reposition the governor valve.

Speed variation, expressed as a percentage, is the total magnitude of the fluctuations from the set speed permitted by the governor system when the turbine is normally operating at rated speed, power, and steam conditions. This is the “insensitivity” of the governor system. The speed variation equation is

$$\frac{\text{Maximum speed} - \text{minimum speed}}{\text{Rated speed} \times 2} \times 100 = \pm \% \text{ speed variation}$$

A turbine with 4000 rpm rated speed and NEMA A governor system would have ± 30 rpm maximum speed variation:

$$4000 \times \frac{\pm 0.75}{100} = \pm 30 \text{ rpm}$$

Maximum speed rise represents the momentary increase in speed when the load is suddenly reduced from rated power to no-load power with the pump still coupled to the turbine shaft. Shortly after the sudden loss of load, the governor system will cause the turbine speed to be reduced to the no-load speed.

Maximum speed rise is also expressed as a percentage of rated speed and calculated as follows:

$$\frac{\text{Maximum speed} - \text{rated speed}}{\text{Rated speed}} \times 100 = \% \text{ speed rise}$$

Therefore

$$\text{Maximum speed} = \text{rated speed} + \frac{\text{rated speed} \times \% \text{ speed rise}}{100}$$

A turbine with 4000 rpm rated speed and equipped with a NEMA A governor will have a maximum speed rise to

$$4000 + \frac{4000 \times 13}{100} = 4520 \text{ rpm}$$

The setting of the *overspeed trip* is a function of the maximum speed rise of the governor system—it must be higher than the maximum speed rise. The recommended settings are

Class of governor system	Overspeed trip setting, (% of rated speed)
A	115
B	110
C	110
D	110

The overspeed trip system for a turbine with 4000 rpm rated speed and equipped with a NEMA A governor is set for operation at

$$4000 \times \frac{115}{100} = 4600 \text{ rpm}$$

The speed-sensitive portion of the speed governor system is usually a set of spring-loaded rotating weights. Movement of the weights caused by a change in turbine speed positions the governor valve through a suitable linkage.

The speed-sensitive element can also be other devices that are speed-responsive, such as a positive displacement oil pump, electric generator, or a magnetic impulse signal generator.

The rotating-weight governor system is a direct-acting type and is classified as a NEMA A governor. *Direct-acting* designates a governor system in which the speed-sensitive element also provides the power for positioning the governor valve.

The NEMA B, C, and D governor systems have speed-sensitive elements that position the governor valve through a relay or servomotor system instead of actuating the valve directly. The speed-sensitive element can therefore be more precise and sensitive, as required for the improved governor system performance.

Electronic governors with pneumatic/hydraulic actuators are also available that offer more control, flexibility, and speed adjustment as well as other operating modes.

THEORY

A steam turbine develops mechanical work by converting to work the available heat energy in the steam expansion. Heat and mechanical work, being two forms of energy, can be converted from one to the other.

The heat energy is converted in two steps. The steam expands in nozzles and discharges at a high velocity, converting the available heat energy to velocity (kinetic) energy. The high-velocity steam strikes moving blades, converting the velocity energy to work. Because the total heat energy available in the steam is converted to velocity (kinetic) energy, the magnitude of the steam velocity is dependent upon the available energy.

The mechanical work that is developed in the turbine by the high-velocity steam striking the buckets is a function of the speed of the buckets. Maximum work occurs when the bucket velocity is approximately one-half the steam jet velocity for an impulse stage and one-fourth the steam jet velocity for a velocity-compounded impulse stage. Although the steam jet velocity is fixed by the available heat energy, the bucket velocity is fixed by the speed of the turbine and the diameter of the turbine wheel on which the buckets are mounted. The work developed, or the efficiency of the turbine, ignoring losses in the turbine, is therefore determined by the size of the turbine and the turbine (pump) speed for a fixed amount of available heat energy.

The most common single-stage turbine is the velocity-compounded (Curtis) type. The complete expansion from inlet to exhaust pressure occurs in one step. The Curtis stage, with two rows of rotating buckets, and two re-entry-type velocity-compounded stages are illustrated in Figure 5.

Single-stage turbines are available with a wide range of efficiencies because they are manufactured with a variety of wheel diameters: 9 to 28 in (22 to 71 cm).

Multistage turbines are manufactured with a more limited variety of wheel sizes. The efficiency of multistage turbines is varied primarily by varying the number of stages. When the total available energy of the steam results in a steam velocity greater than twice the bucket velocity (using convenient wheel sizes), a multistage turbine will be more efficient. In a multistage turbine, the total steam expansion is divided among the various impulse stages to produce the desired steam velocity for each row of buckets.

A steam turbine is normally evaluated using *steam rate*—the amount of steam required by the turbine to produce the specified power per hour at the specified speed—rather than *efficiency*. The steam rate is a direct function of the turbine efficiency.

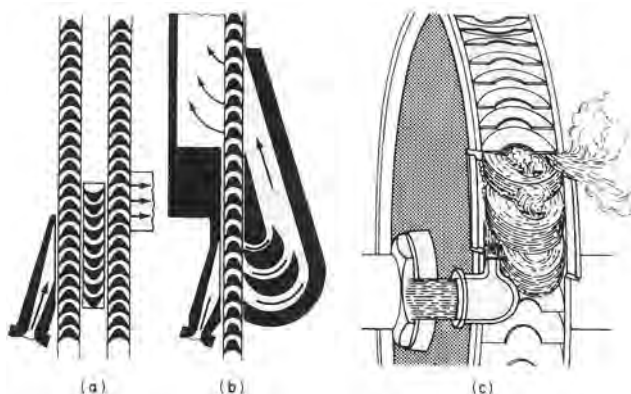


FIGURE 5 Velocity-compounded stages: (a) Curtis stage; steam flows once through moving buckets; (b) re-entry stage; steam flows twice through moving buckets; (c) re-entry stage; steam flows three times through moving buckets (reprinted with permission from *Power*, June 1962)

The steam consumption can be expressed either as steam rate, pounds of steam per horsepower-hour (kilograms per kilowatt-hour) or as steam flow, pounds of steam per hour (kilograms per hour). The higher the efficiency, the lower the steam rate or steam flow, and vice versa.

The total available energy of the steam is that available from an isentropic expansion. For given initial steam pressure and temperature and exhaust pressure, the available energy in British thermal units per pound (kilojoules per kilogram) of steam can be obtained from the tables or the Mollier chart in Reference 3.

The available energy can be converted to power units and expressed as the theoretical steam rate—pounds per horsepower-hour or pounds per kilowatt-hour (kilograms per kilowatt-hour). The theoretical steam rate is the steam rate for a 100% efficient turbine and therefore can be used more conveniently than energy in British thermal units per pound (kilojoules per kilogram) for the calculation of turbine steam rates. Theoretical steam rates can be obtained directly from theoretical steam rate tables (ASME) or from the polar Mollier chart (Elliott Company).

The actual steam rate for a turbine is greater than the theoretical steam rate because of the losses that occur in the turbine when the available energy is converted to mechanical work and because of the ratio of the steam velocity to bucket velocity. The energy remaining in the steam exhausting from the turbine is greater than that after an isentropic expansion, as illustrated in Mollier diagram shown in Figure 6, where

- 1 = energy in steam at initial steam pressure and temperature
- 2 = energy in steam at exhaust pressure for an isentropic expansion
- 3 = actual energy in steam at exhaust pressure

The efficiency of the turbine is

$$\frac{h_1 - h_3}{h_1 - h_2} \quad \text{or} \quad \frac{\text{theoretical steam rate}}{\text{actual steam rate}}$$

The governor and trip valve pressure drop losses are a function of the sizes of these two valves and the steam flow. The governor valve pressure drop will vary more than the trip valve pressure drop because of changes in valve position with power required by the driven pump, speed variations, and changes in inlet and exhaust steam conditions.

The nozzle loss is due to friction in the nozzles as the steam expands. The efficiency of the nozzles is a function of the ratio of the actual and ideal exit steam velocities squared. The efficiency is usually between 95 and 99%.

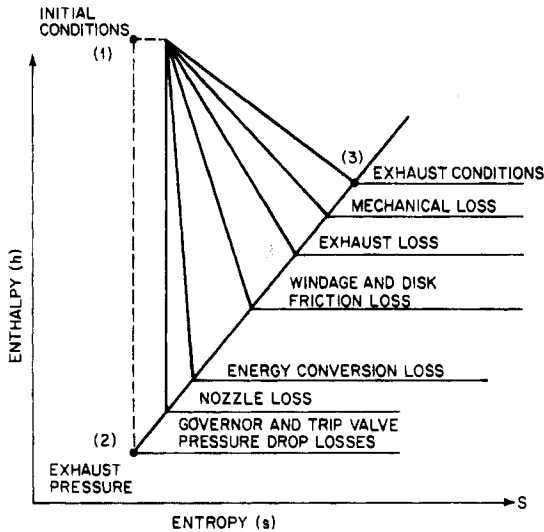


FIGURE 6 Mollier diagram with energy losses

The windage and disk friction losses are due to the friction between the steam and the disks and the blades fanning the steam. This loss varies inversely with the specific volume of the steam, increases with exhaust pressure, and increases with the diameter of the wheel and the length of the blades.

The use of a larger-diameter wheel may increase the efficiency, but the windage and disk friction losses will reduce the improvement and may even cause a net loss in overall efficiency.

The exhaust losses represent the kinetic energy remaining in the steam as a result of the velocity of the steam leaving the bucket and the pressure drop in the steam as it passes out the exhaust connection.

The energy conversion loss is due to the nonideal conversion of the steam velocity energy to mechanical work in the buckets as a function of the steam velocity and bucket velocity, plus nonideal nozzle and bucket angles, friction in the system, and so on.

The performance that can be expected from a single-stage Curtis-type turbine may be obtained from Figures 7, 8, and 9, and Table 1 after determining the theoretical steam rate.

$$\text{Steam rate} = \frac{\text{base steam rate}}{\text{superheat correction factor}} \times \frac{\text{power} + \text{power loss}}{\text{power}}$$

To obtain superheat, subtract temperature given in Table 1 from total initial temperature.

SAMPLE CALCULATION

STEAM CONDITIONS 250 lb/in² (1724 kPa) gage inlet, 575°F (302°C) inlet, and 50 lb/in² (345 kPa) gage exhaust.

DESIGN CONDITIONS turbine to develop 500 hp (373 kW) at 4000 rpm.

1. Theoretical steam rate = 26.07 lb/kWh (11.82 kg/kWh)
2. Base steam rate for 28-in (71.12-cm) wheel (Figure 7) = 36 lb/hp · h (22 kg/kWh)

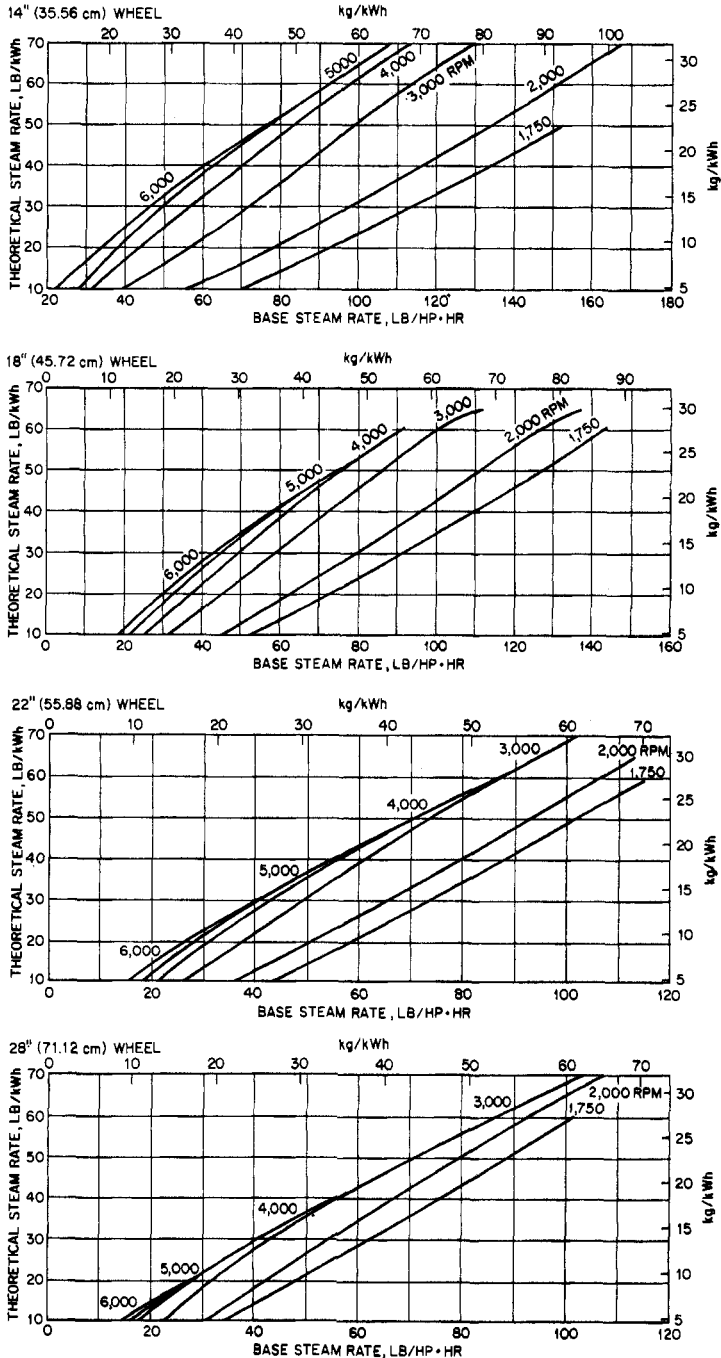


FIGURE 7 Base steam rates (Elliott)

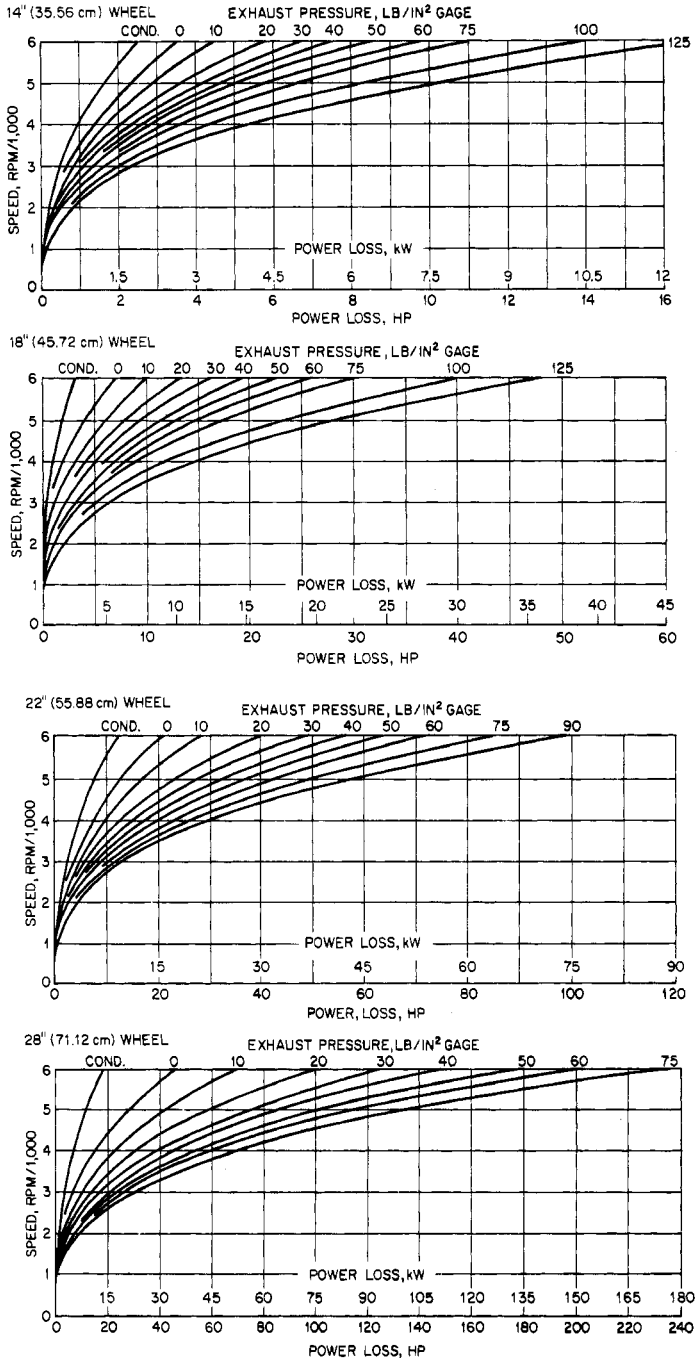


FIGURE 8 Power loss ($\text{lb/in}^2 \times 6.895 = \text{kPa}$) (Elliott)

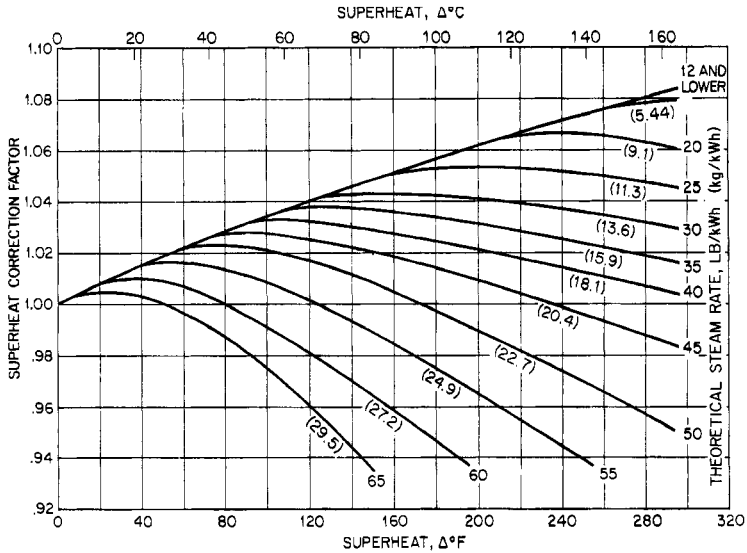


FIGURE 9 Superheat correction factor (Elliott)

3. Power loss for 28-in (71.12-cm) wheel (Figure 8) = 55 hp (41 kW)
4. Temperature of dry and saturated inlet steam (Table 1) = 406°F (208°C)
5. Superheat (Table 1) = 575 - 406 = 169Δ°F (302 - 208 = 94Δ°C)
6. Superheat corrections factor (Figure 9) = 1.052
7. Steam rate =

In USCS units
$$\frac{36}{1.052} \times \frac{500 + 55}{500} = 38.0 \text{ lb/hp} \cdot \text{h}$$

In SI units
$$\frac{21.89}{1.052} \times \frac{373 + 41}{373} = 23.1 \text{ kg/kWh}$$

The ability of a particular size turbine to develop the required power is determined primarily by

1. The flow capacity of the inlet and exhaust connection from Figures 10 and 11, where steam flow = power × steam rate
2. The flow capacity of the nozzles available in a particular turbine (the number and size of the nozzles vary considerably with each design of turbine manufactured, and thus a meaningful plot cannot be included here).

EVALUATION OF COSTS

The economical selection of a turbine considers the initial cost of the turbine and the operating costs. A lower-steam-rate (more efficient) turbine generally has a higher initial cost

TABLE 1 Temperature of dry and saturated steam

Lb/in ² gage	Saturation temp., °F	Lb/in ² gage	Saturation temp., °F	Lb/in ² gage	Saturation temp., °F	Lb/in ² gage	Saturation temp., °F
0	213	150	366	300	422	450	460
5	228	155	368	305	423	455	461
10	240	160	371	310	425	460	462
15	250	165	373	315	426	465	463
20	259	170	375	320	428	470	464
25	267	175	378	325	429	475	465
30	274	180	380	330	431	480	466
35	281	185	382	335	432	485	467
40	287	190	384	340	433	490	468
45	293	195	386	345	434	495	469
50	298	200	388	350	436	500	470
55	303	205	390	355	437	510	472
60	308	210	392	360	438	520	474
65	312	215	394	365	440	530	476
70	316	220	396	370	441	540	478
75	320	225	397	375	442	550	480
80	328	230	399	380	444	560	482
85	328	235	401	385	445	570	483
90	331	240	403	390	446	580	485
95	335	245	404	395	447	590	487
100	338	250	406	400	448	600	489
105	341	255	408	405	449	610	491
110	344	260	410	410	451	620	492
115	347	265	411	415	452	630	494
120	350	270	413	420	453	640	496
125	353	275	414	425	454	650	497
130	356	280	416	430	455	660	499
135	358	285	417	435	456	670	501
140	361	290	419	440	457	680	502
145	364	295	420	445	458	690	504

To obtain superheat, subtract temperature given above from total initial temperature. Pressure in kPa = $6.895 \times \text{lb/in}^2$. Temperature in °C = $(°F - 32) \div 1.8$.

than a turbine with a higher steam rate. The operating costs are the cost of the steam for the number of years upon which the evaluation is to be based:

$$\text{Total cost} = \text{initial cost} + \left(\text{power} \times \frac{\text{steam}}{\text{rate}} \times \frac{\text{steam}}{\text{cost}} \times \frac{\text{operating hours}}{\text{per year}} \times \frac{\text{number}}{\text{of years}} \right)$$

The cost of installation—foundation, steam piping, and cooling water service (also electric service if required)—is not normally considered unless there are significant differences between the turbines: single-stage versus multistage, direct-connected versus turbine and gear, vertical turbine versus horizontal turbine with a right-angle gear, and so on.

The economical selection of a steam turbine versus an electric motor, diesel engine, gas engine, gas turbine, and so on, must include an evaluation of all installation costs, costs of supporting equipment (starters, switch gear, fuel-supply system, and so on), operating costs, and the initial costs of the various drivers being considered.

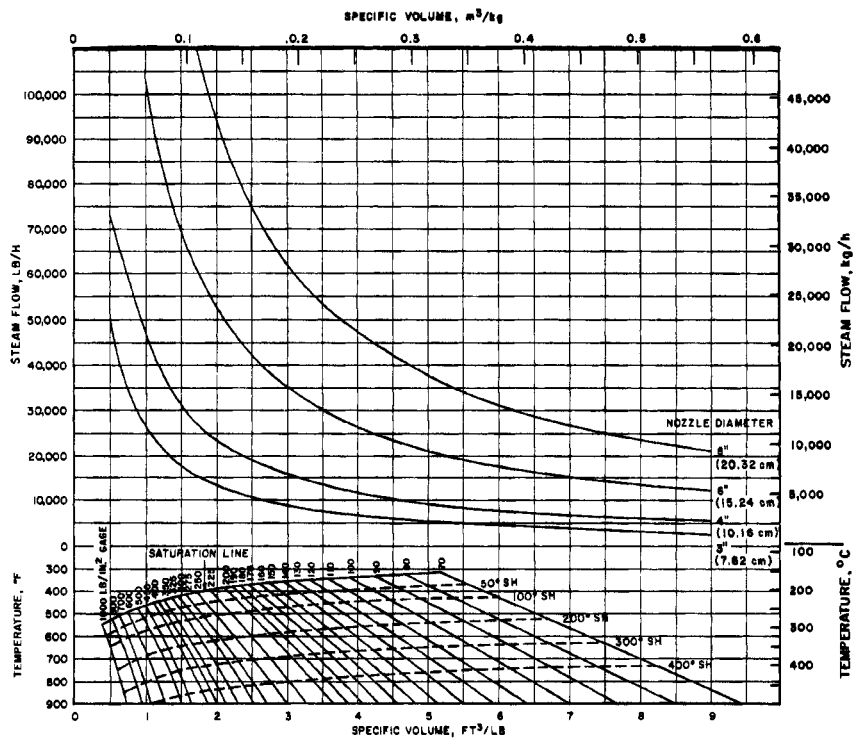


FIGURE 10 Nominal inlet flow capacity. Read inlet nozzle size required to pass maximum flow, based on 150-ft/s (45.7-m/s) steam velocity ($lb/in^2 \times 6.895 = kPa$; $^{\circ}F_{SH} \times 0.555 = ^{\circ}C_{SH}$)

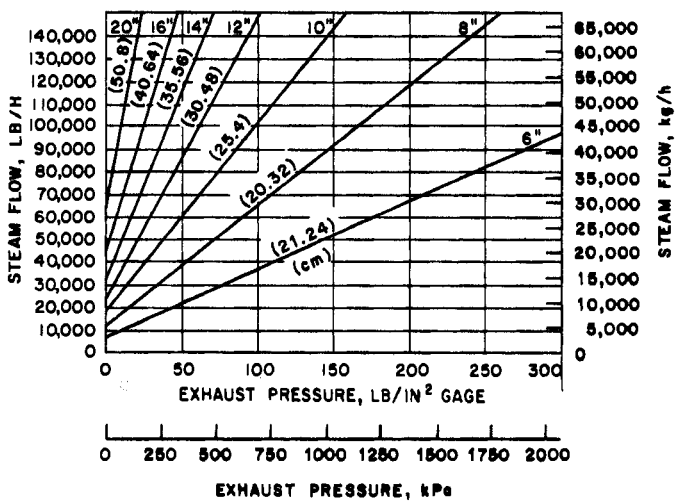


FIGURE 11 Nominal exhaust flow capacity. Noncondensing exhaust nozzles, based on 200-ft/s (61 m/s) steam velocity

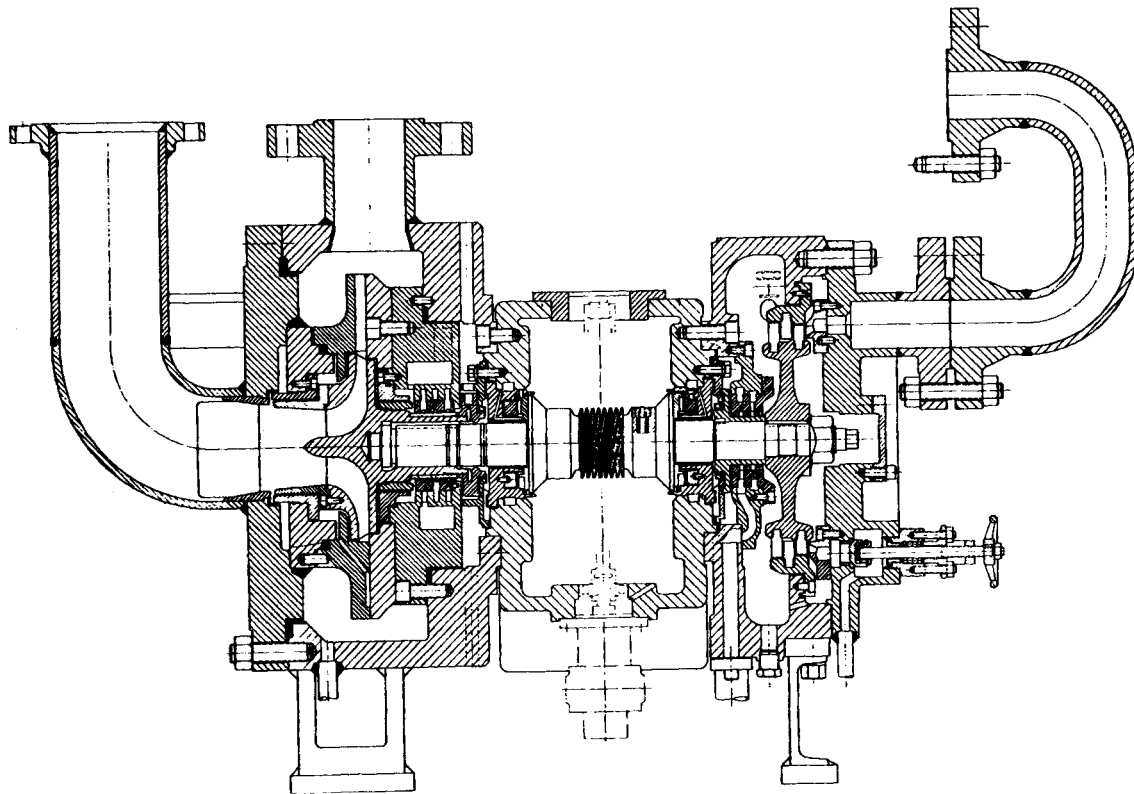


FIGURE 12 High-speed centrifugal pump and single-stage steam turbine mounted on a common shaft. Package is designed for boiler feed in marine, industrial, and process steam plants (Flowserve Corporation).

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9.1.3 ENGINES

F. J. GUNTHER

ENGINE SELECTION AND APPLICATION

The internal combustion engine is used extensively as a driver for centrifugal and displacement pumps. Depending on the application, the engine fuel may be gasoline, natural gas, liquid petroleum gas (LPG), sewage gas, or diesel fuel. It may be either liquid- or air-cooled.

Basic Design Variations The cylinder block construction is vertical or horizontal, in-line or V-type. The number of cylinders ranges from 1 to 20. The engine cycle is either four (one power stroke in two revolutions of the crankshaft) or two (one power stroke in one revolution of the crankshaft). The combustion chamber and cylinder head design are classed as L-head (valves in the cylinder block) or valve-in head. In the diesel engine, the combustion chamber may be of the precombustion chamber design, where an ante chamber is used to initiate combustion, or of the direct injection design, where the fuel is injected directly into the cylinder. The piston design action may be vertical, horizontal, at an angle as in the V-type engine, or opposed piston (two pistons operating in the same cylinder).

Power Ratings The power range of engines in current production, depending on displacement, number of cylinders, and speed, is as follows:

1. Air-cooled gasoline, natural gas, and diesel: 1.0 to 75 hp (0.75 to 56 kW)
2. Liquid-cooled gasoline: 10 to 300 hp (75 to 224 kW)
3. Liquid-cooled natural gas, LPG, and sewage gas: 10 to 15,000 hp (75 to 11,200 kW)
4. Liquid-cooled diesel: 10 to 50,000 hp (75 to 37,300 kW)
5. Dual fuel, natural gas, LPG, and diesel: 150 to 25,000 hp (112 to 18,700 kW)

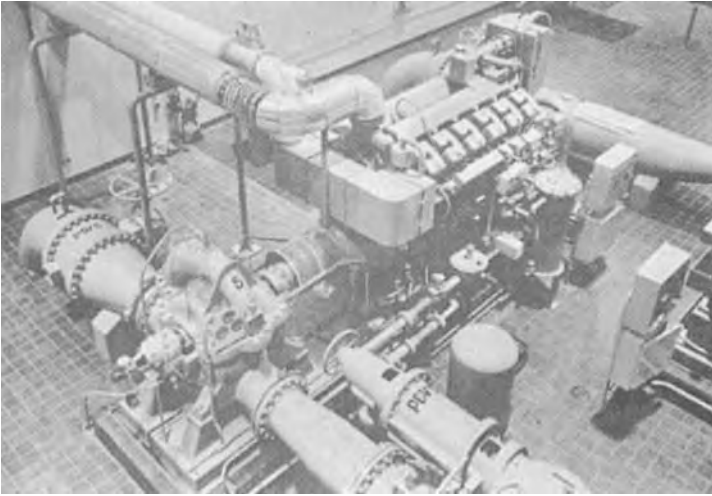


FIGURE 1 Natural gas engine driving a horizontal water pump for Winnipeg, Canada, water utility (Waukesha Motor)

Typical engine driver applications are represented in Figures 1 and 2.

The rating of the internal combustion engine is the most important consideration in making the proper selection. The general practice is to rate engines according to the severity of the duty to be performed. The most common rating classifications are maximum, standby or intermittent, and continuous.

The maximum output is based on dynamometer tests that are corrected to standard atmospheric conditions for temperature and barometric pressure. In applications, this power rating is reduced by accessories such as cooling fans, air cleaners, and starting systems.

Standby, or intermittent, and continuous ratings are arrived at by applying a percentage factor to the net maximum power rating. For example, 75 to 80% is used for continuous and 90% for intermittent.

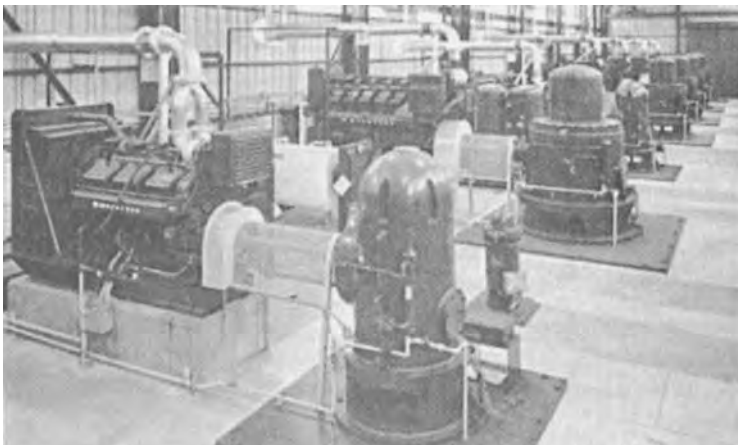


FIGURE 2 Diesel engine driving pumps for flood-control station in Seattle (Waukesha Motor)

Duty cycle is a term used to describe the load pattern imposed on the engine. If the load factor (ratio of average load to maximum capabilities) is low, we call the duty cycle "light," but if it is high, we classify the cycle "heavy." Continuous, or heavy-duty, service is generally considered to be 24 h/day, with little variation in load or speed. Intermittent service is classified as duty where an engine is called upon to operate in emergencies or at reduced loads at frequent intervals.

In analyzing power problems when selecting a proper engine, certain terms are used in the industry:

DISPLACEMENT The displacement in cubic inches (cubic centimeters) of an engine is

In USCS units, $D = \text{bore (in)}^2 \times 0.7854 \times \text{stroke (in)} \times \text{no. of cylinders}$

In SI units $D = \text{bore (cm)}^2 \times 0.7854 \times \text{stroke (cm)} \times \text{no. of cylinders}$

TORQUE The twisting effort of the engine in pound-feet (Newton-meters) is

In USCS units $T = 5252 \times \frac{\text{bhp}}{\text{rpm}}$

In SI units $T = 9545 \times \frac{\text{bkW}}{\text{rpm}}$

ENGINE POWER Brake horsepower (bkW) is the useful power at the flywheel:

In USCS units $\text{bhp} = T \times \frac{\text{rpm}}{5252}$

In SI units $\text{bkW} = T \times \frac{\text{rpm}}{9545}$

SPECIFIC FUEL CONSUMPTION The mass flow rate of fuel consumed per unit time divided by the power output, usually at the engine rating, measures how efficiently an engine is using the fuel supplied to produce that power. It is measured in pound-mass per brake horsepower-hour (gram per kilowatt-hour).

In USCS units $\text{bsfc} = \frac{\text{lb mass of fuel per hour}}{\text{bhp}}$

In SI units $\text{bsfc} = \frac{\text{grams of fuel per hour}}{\text{bkW}}$

BRAKE MEAN EFFECTIVE PRESSURE The average cylinder pressure to give a resultant torque at the flywheel in pounds per square inch (kilopascals)

In USCS units $\text{bmep} = \frac{792,000 \times \text{bhp}}{\text{rpm} \times D} \text{ (four-cycle)}$

$\text{bmep} = \frac{396,000 \times \text{bhp}}{\text{rpm} \times D} \text{ (two cycle)}$

In SI units $\text{bmep} = \frac{120 \times 10^6 \times \text{bkW}}{\text{rpm} \times D} \text{ (four-cycle)}$

$\text{bmep} = \frac{60 \times 10^6 \times \text{bkW}}{\text{rpm} \times D} \text{ (two cycle)}$

PISTON SPEED At a given speed, the average velocity of piston in feet per minute (meters per second) is

$$\text{In USCS units} \quad \text{Piston speed} = \text{stroke (in)} \times 2 \times \frac{\text{rpm}}{12}$$

$$\text{In SI units} \quad \text{Piston speed} = \text{stroke (cm)} \times 2 \times \frac{\text{rpm}}{6000}$$

In selecting an engine for a particular application, the following variables should be considered:

- Altitude
- Ambient air temperature
- Rotation and speed
- Bmep and piston speed
- Maintenance
- Type of fuel
- Operating atmosphere (dust and dirt)
- Vibrations and torsionals
- Engine pollutants

The observed power is that produced by an engine at the existing altitude and temperature. All engine manufacturers publish power ratings corrected to certain conditions; for conditions other than these, it is necessary to correct by applying a percentage factor for altitude and temperature. Generally this is $3\frac{1}{2}\%$ per thousand feet (305 m) above sea level and 1% for every 10°F (5.6°C) above 60°F (50.4°C). In a turbocharged engine, there is no established standard and the engine manufacturer should be consulted.

The basic rotation of engines in current production is counterclockwise when viewed from the flywheel end of the engine, although many of the larger engines are available in both counterclockwise and clockwise rotation. The speed of the engine is generally fixed by the equipment being driven. Through the use of speed-increasing or -reducing gear boxes, the proper engine for a given application may be selected. A gear box may also be used to correct a rotation problem.

Speed ranges for engines generally fall into three categories:

- High—above 1500 rpm
- Medium—700 to 1500 rpm
- Low—below 700 rpm

High-speed engines generally offer weight and size advantages as well as cost savings and thus are used for standby applications. On the other hand, medium- or low-speed engines, although heavier and larger, offer a gain in service life and lower maintenance costs.

The speed flexibility of an engine drive is important when the engine is to be used to drive a pump that must move variable quantities of liquid. The engine speed may be changed very simply either manually or through the use of liquid or pressure controls.

Bmep is generally a measure of load, and piston speed a measure of potential wear and maintenance. Although the introduction of the turbocharged and intercooled engine has somewhat changed the consideration given these factors, it is still important to consider them in selecting engines where long life is a factor.

Maintenance of engines has been considered by some as objectionable and more costly than electric power. A recent innovation of engine manufacturers, in the form of a service contract for installations where trained personnel are not available or desirable for economic reasons, can eliminate these objections and costs. The complete maintenance of the engine is done on a fixed-fee basis for a designated period of time.

The exhaust gases of spark ignition and compression ignition (diesel) engines contain pollutants that for many engine applications are increasingly the subject of legislation restricting the quantity of pollutants the engine can emit. Examples of pollutants are carbon monoxide (CO), oxides of nitrogen (NO_x), unburned hydrocarbons (HC), and, for diesels, particulates in the form of carbon soot. Pollutants can be measured on a specific basis, such as g/bhp-hr and ppm (parts per millions by volume), or on a site basis, such as lb/hr or tons/year. Engine manufacturers are designing "clean" engines that incorporate features to minimize the formation of pollutants during combustion and may include catalytic converters and particulate filters in the exhaust system to further reduce the pollutants emitted in the exhaust gases.

The remaining conditions listed previously will be discussed in detail later.

FUEL SYSTEMS

Gasoline Gasoline is used primarily with standby pumping units. Inasmuch as the spark ignition system first introduced the internal combustion engine to power applications, gasoline was used as the primary fuel. Commercial gasoline has an average heating value of 19,000 Btu/lb (44.2 MJ/kg). It is easy to transport and handle and, unlike gaseous fuels, does not require pressure storage and regulating equipment. The starting capabilities of a gasoline engine are satisfactory, provided the engine is in good operating condition. With the high-power engine of today, refinery control can produce a fuel matched to the octane requirement of the engine.

Gasoline does have some disadvantages that are reducing its use as an engine fuel. In small-volume usage, it is safe and easily handled. In larger volumes, it becomes expensive and hazardous. Gasoline will form gums and resins when exposed to air over a period of time. There is also the possibility of condensation of water in the fuel, which is detrimental to good operation. The danger of fire is always present because of leaks in the system. Finally, the increased production and distribution of gasoline have made it a target of increasing taxation, making it economically prohibitive in many installations.

Gas A gaseous fuel system using natural gas, LPG, or sewage gas may be a simple manually controlled system, such as a gasoline engine, or a carefully engineered automatic system. The basic gas carburetion system consists of a carburetor and pressure regulator mounted on the engine. A gas distribution system, like a water supply system, must be at some designated pressure and flow, and so a field pressure regulator is required. The characteristics of this regulator will depend upon the gas analysis, the displacement of the engine, the speed range, and local regulations. A typical schematic of a gas fuel system is shown in Figure 3. The location of the regulator is generally under the jurisdiction of the gas utility that supplies it. In most cases, a single field regulator is all that is required, but at times this can cause problems. For example, subsequent installation of gas-burning equipment used intermittently may cause gas pressure regulation not compatible with the

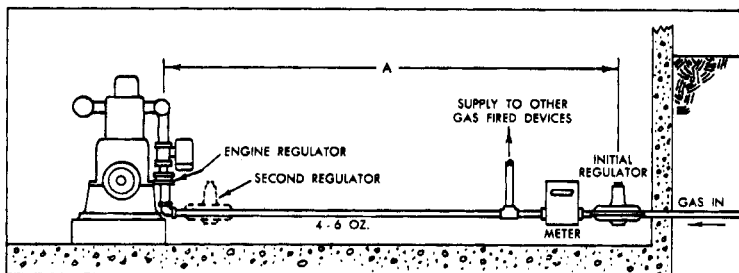


FIGURE 3 Typical natural gas fuel system (Waukesha Motor)

small amount required for pilot lighting. A single regulator installed some distance from the engine could result in hard starting because the engine vacuum is not sufficient for a full gas flow. To eliminate this problem, the initial regulator is set at a higher pressure in order to give a readily available supply of fuel for all devices.

Natural gas has an average heating value of 800 to 1000 Btu/ft³ (29.8 to 37.3 MJ/m³). Commercial butane has a value of 2950 Btu/ft³ (110 MJ/m³), and propane a value of 3370 Btu/ft³ (126 MJ/m³). Commercial LPG fuel, which is a mixture of butane and propane, varies in both amount and corresponding heating value.

LPG fuel is produced by mechanical and compression processes, and the methods of distribution and handling must meet regulations. Natural gas is usually supplied under moderate pressures, seldom exceeding 50 lb/in (345 kPa), whereas LPG fuel is supplied as a liquid under pressures as high as 200 lb/in² (1380 kPa) in warm weather. As a result, natural gas will readily mix with air and burn, whereas LPG fuel must be changed from a liquid to a vapor with the addition of heat, as shown in Figure 4.

As a natural process in the modern waste-treatment plant, sewage gas may be produced in the sewage digester. This gas has the same basic qualities as natural gas and is composed of about 65 to 70% methane. It has a heating value of 550 to 700 Btu/ft³ (20.5 to 26.1 MJ/m³). The same basic carburetion system used for natural gas is used. Sewage gas contains inert substances, particularly hydrogen sulfide or free sulfur, which in the presence of free moisture or moisture resulting from combustion will form sulfurous acid, which is corrosive and thus damaging to the valves, pistons, and cylinder walls of an engine. An engine can tolerate from 10 to 30 g of sulfur per 100 ft³ (350 to 1100 g per 100 m³). Beyond this, a filtering system to remove the sulfur and moisture is advisable.

Diesel The diesel engine over the years has been used for larger power systems. The initial cost of the system is justified to an extent by lower cost of the fuel. The better fuel economy of the diesel engine and its torque characteristics also are important factors in the selection of this fuel for many applications. Diesel fuel has one distinct advantage: it does not form the dangerous fuel vapors that other fuels do. It does require, however, a good fuel-filtering system because of contaminants in the fuel that can create problems for the precision design of the fuel-injection system.

Commercial diesel fuels are the residue that remains after the more volatile fractions of crude oil have been removed. The heating value is generally about 19,000 Btu/lb (44 MJ/kg). In the diesel engine, the fuel is injected into the cylinder at the end of the compression stroke in an atomized form. The compression stroke results in a temperature sufficient to ignite the fuel without the use of any ignition device. Although fuel systems from different engine manufacturers vary, the basic components are the same.

The larger diesel engines may be designed to operate on a dual fuel system. The engine operates on gaseous fuels with a pilot injection of diesel fuel for ignition. In case of a loss of the gaseous fuel supply, the engine will convert to 100% diesel fuel.

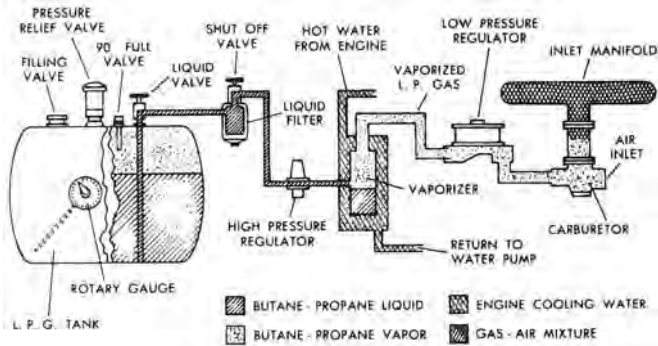


FIGURE 4 Typical LPG fuel system (Waukesha Motor)

Relative Performance Curves Typical performance curves comparing power, torque, and part-load fuel economy are shown in Figure 5.

COOLING SYSTEMS

Cooling is essential in all internal combustion engines because only a small portion of the total heat energy of any fuel is converted to useful energy. The remainder is dissipated into the coolant, exhaust, and lubricating oil and by radiation. A hypothetical heat balance is shown in Figure 6.

Specific data and recommendations on cooling requirements vary from one manufacturer to another. In general, the heat rejection to an engine cooling system will range between 30 and 60 Btu/hp · min (25 and 51 MJ/kWh) for diesel engines and up to 70 Btu/hp · min (59 MJ/kWh) for natural gas and gasoline engines. This heat must be transferred to some form of heat-exchange medium.

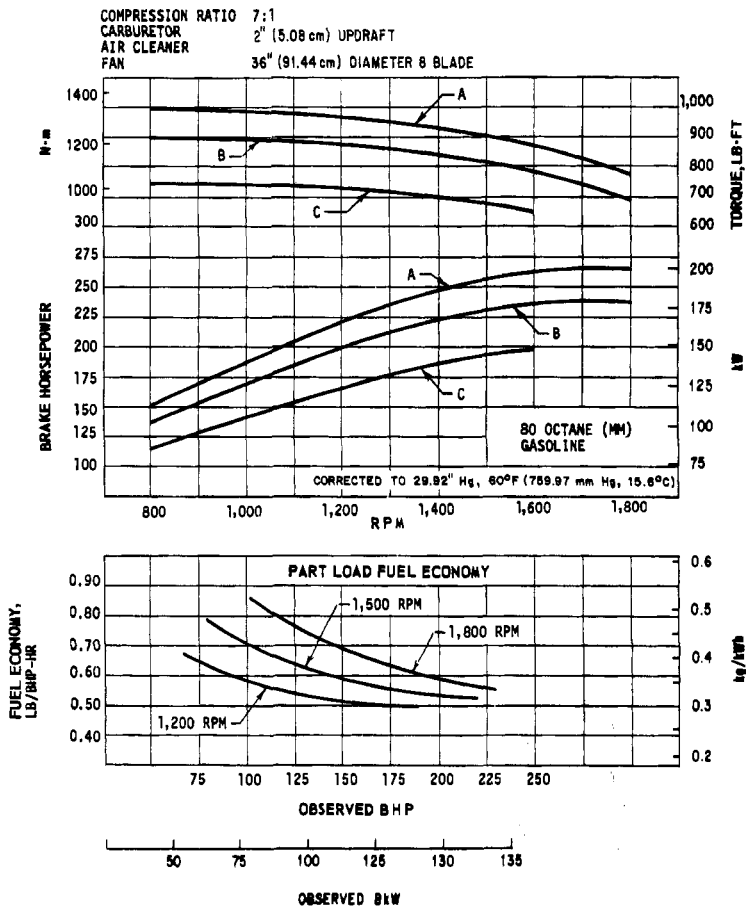


FIGURE 5A Gasoline engine performance curves: (A) maximum, (B) intermittent, (C) continuous ratings of engines with accessories (Waukesha Motor)

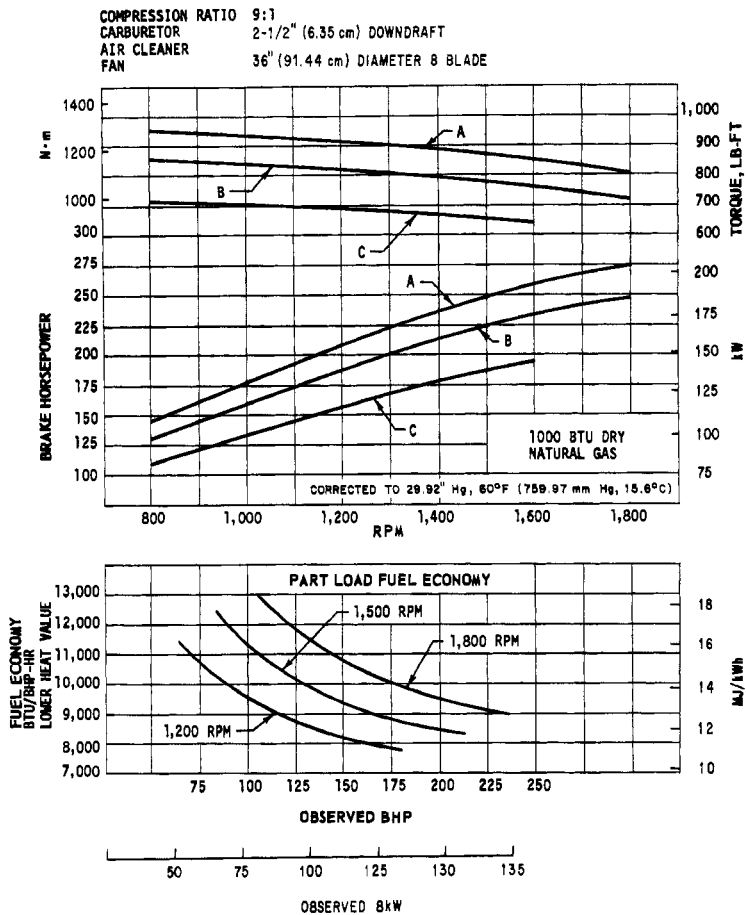


FIGURE 5B Gas engine performance curves: (A) maximum, (B) intermittent, (C) continuous ratings of engines with accessories (Waukesha Motor)

In designing any cooling system, certain factors must be considered:

- Additional heat from driven equipment, such as the cooling of speed-reducing or -increasing gears where the engine coolant is the medium
- Water-cooled exhaust manifolds on the engines or water-cooled exhaust turbochargers or after-coolers
- High ambient temperatures or heat from nearby equipment
- Variation in the heat-exchanger coolant temperature
- Entrapment of substantial quantities of air in the coolant water
- Inability to maintain a clean cooling system

With any cooling system, one of the most important factors of design is the temperature drop across the engine. Most engine manufacturers desire a temperature differential of no more than 10 to 12°F (5.6 to 6.7°C), and closer values are desirable. A jacket-water

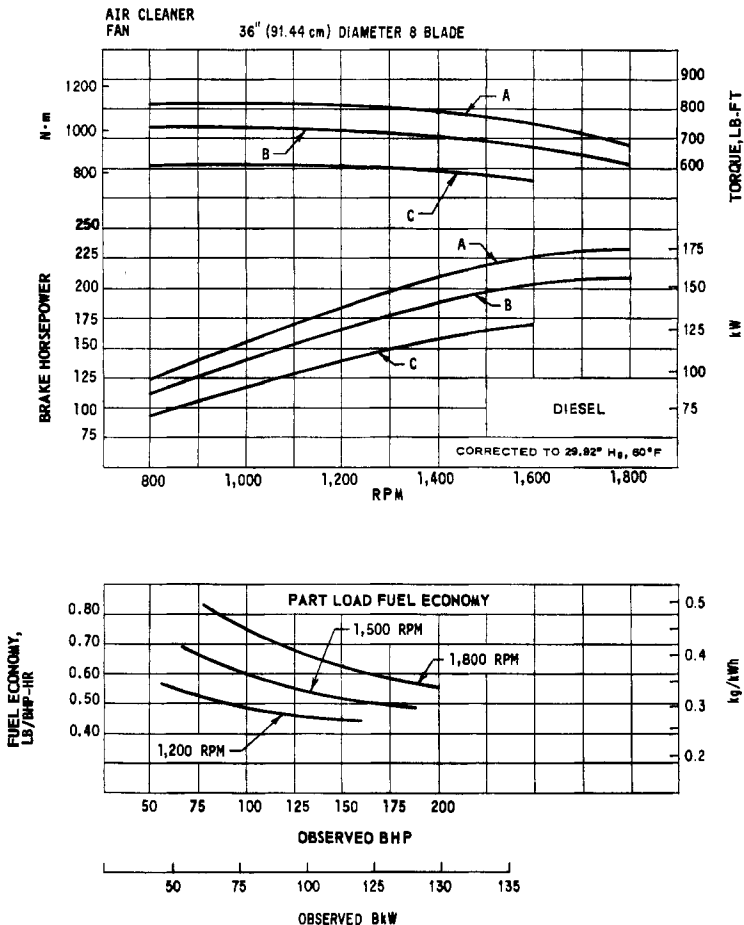


FIGURE 5C Diesel engine performance curves: (A) maximum, (B) intermittent, (C) continuous ratings of engines with accessories (Waukesha Motor)

temperature across the engine of 170°F (77°C) is preferred, and in high-temperature or waste-heat-recovery systems, temperatures of 200°F (93°C) or more are common and not harmful.

Radiator The radiator cooling system is perhaps the most common and best understood method of cooling. It is based on a closed system of tubes through which the jacket water passes. The heat is dissipated by a fan, creating a stream of moving air passing through the tubes. The fan is driven either by the engine or by an auxiliary source of power (Figure 7).

An engine in a fixed outside installation can be cooled without much difficulty. Certain factors must be considered, such as ambient temperature, direction of the prevailing wind, and presence of foreign airborne materials. In high temperatures (usually above 110°F [48°C]), a larger radiator is required. If the prevailing winds are extremely high, the unit can be located to offset normal fan flow. Screening can be used to prevent the clogging of the air passes in the radiator where the atmosphere tends to contain foreign airborne material, such as dust.

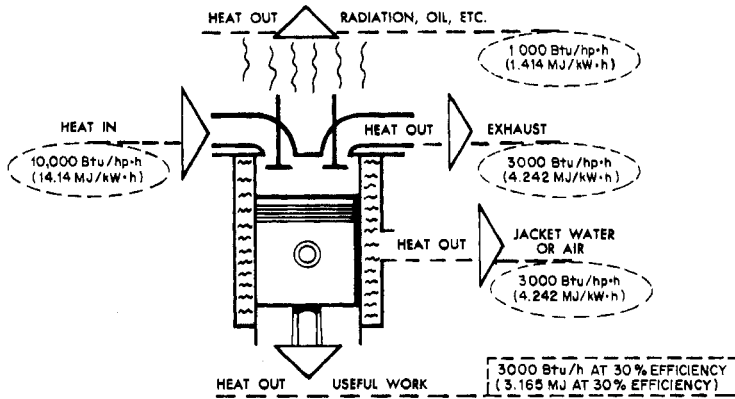


FIGURE 6 Hypothetical heat balance (Waukesha Motor)

Radiator cooling may be used in an inside installation, but there are certain problems which, unless properly anticipated, limit this system. The recirculation of cooling air and the radiation of exhaust heat from the engine create a problem. As was previously pointed out, every 10°F (5.6°C) rise above 60°F (15.6°C) results in a 1% loss in power. When 5 to 10% of the total heat put into an engine is radiated, some means of power ventilation must be provided to remove this heat through ducts, louvers, or forced ventilation with a separate fan. An installation of this type is illustrated in Figure 8.

Heat Exchanger The heat exchanger cooling system (Figure 9) is the best system for a stationary engine installation. Using a tube bundle in a closed shell, the cooling exchange medium is water (often called raw water). This water may be plant or process water; it may be recirculated or, in standby installations, allowed to pass to waste. The system has the advantage of the radiator cooling system in that it is self-contained: the quantity and quality of the water in the engine can be controlled. It has the further advantage of not being affected by the flow of heat to air movement if the heat of radiation is taken into account in the design of the system. On the other hand, the cooling medium, unless used in a plant system, is a disadvantage because it is costly. A separate pump is required to provide the necessary water for cooling unless city water or process water is under sufficient pressure.

City Water and Standpipe City water cooling is designed to take water directly from the city main or from the pump the engine is driving. It is used on some emergency or

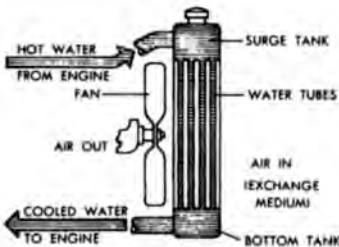


FIGURE 7 Radiator cooling system (Waukesha Motor)

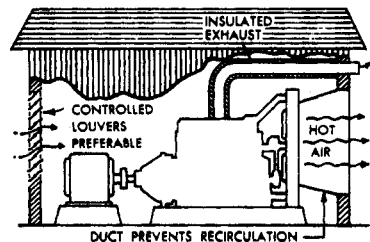


FIGURE 8 Radiator cooling system for an inside installation (Waukesha Motor)

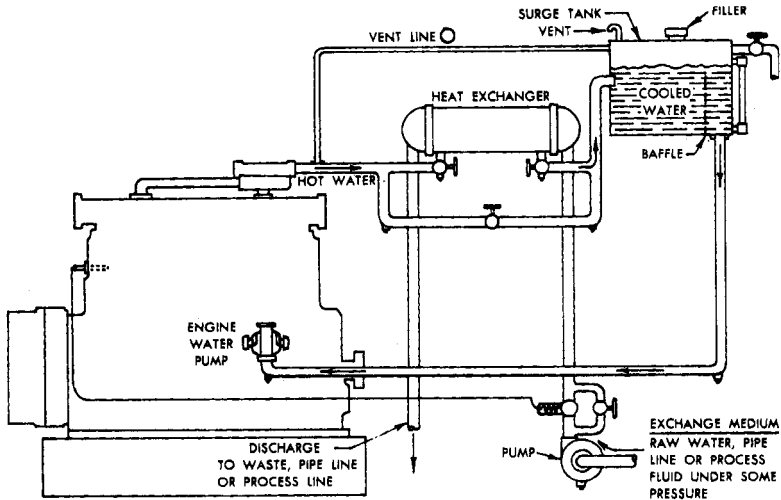


FIGURE 9 Heat exchanger cooling system (Waukesha Motor)

standby installation. It is simple and inexpensive, gives unlimited cooling for moderate-size engines, is easily understood, and will operate instantly in an emergency. On the negative side, the cooling water is wasted, corrosive elements may be introduced into the engine jacket water system, and it may create excessive temperature changes across the engine jacket.

Standpipe cooling (Figure 10) is basically the same as city water cooling except that a thermostatic valve is employed to admit makeup water as required. The vertical pipe is a

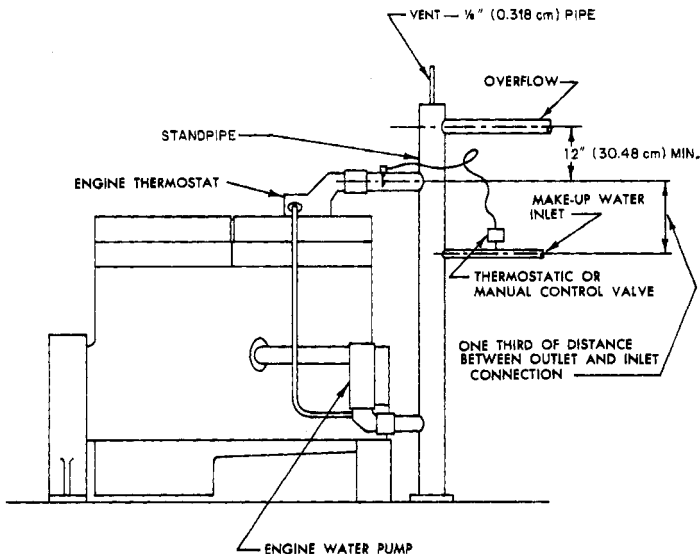


FIGURE 10 Standpipe cooling system (Waukesha Motor)

blending tank into which city water is introduced only in the amount necessary for makeup. The standpipe system is inexpensive and simple to operate.

Ebullition In installations where heat is required for process equipment, a method of high-temperature, or ebullition, cooling is being used as a very economical method, particularly with larger installations. This system has been termed *steam cooling*, *high-temperature*, or *Vapor-phase* (a registered trademark). In this system, the coolant leaves the engine at a temperature equal to or above its atmospheric boiling point and under sufficient pressure to remain liquid until discharged from the engine into a flash chamber, where a drop in pressure causes the formation of steam, which is condensed and returned to the engine at very near the discharge temperature. A schematic of this system is shown in Figure 11. This system has the advantages of a very small temperature differential across the engine, which minimizes distortion of all working parts, and a constant working temperature regardless of load. Because of the higher working temperatures, the combustion area and crankcase of the engine have fewer liquid by-products of combustion and corrosive materials. Of prime importance is the waste heat that can be recovered for plant process with a very small amount of makeup water for cooling.

Cooling Tower Cooling towers are used in some large or multiple-engine installations. Through the use of a current of air, produced either by a natural draft or by mechanical means, a tower causes a sensible heat flow from the cooling water to the air. Atmospheric, or natural, draft towers depend upon natural wind velocities and thus can vary widely. Mechanical draft towers, where the air supply can be controlled, can be put in any area, but the limit to their cooling capacity is the power required to operate them. As the water volume increases, the volume of air required and the pressure the fan has to operate against increase. A point is reached where the cost of installation and operation becomes prohibitive. A diagram of a system that combines components of the cooling systems mentioned plus the waste-heat recovery system silencer, to be mentioned later, is shown in Figure 12.

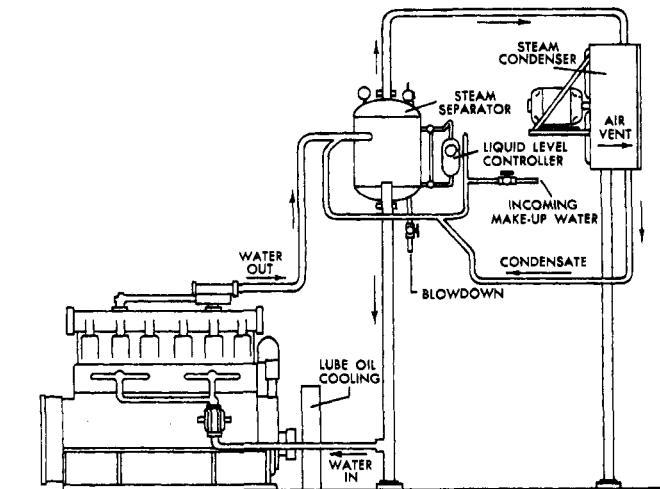


FIGURE 11 High-temperature cooling system (Waukesha Motor)

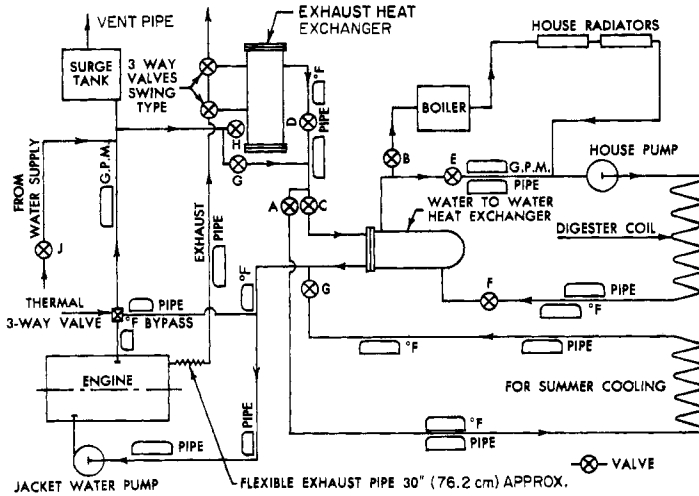


FIGURE 12 Complete cooling system, including waste-heat recovery (Waukesha Motor)

AIR-INTAKE SYSTEMS

A most important consideration in the application of an engine to any pump drive is the engine's ability to "breathe." As air is required for combustion, the design engineers of any project must take into account the necessary provisions for this air. The environment, the service, the speed range of the engine, the duty cycle, and the location from which the combustion air is to be taken are of vital importance.

The first step in the selection of any intake system is to determine the volume of air, in cubic feet per minute (cubic meters per second) required for combustion, which may be calculated by the formula:

$$V_{\text{ideal}} = \frac{B^2 \times S \times \text{rpm} \times N}{C \times K}$$

where B = cylinder bore, in (cm)

S = piston stroke, in (cm)

rpm = engine speed, revs/minute

N = number of cylinders

C = 2,200 for V_{ideal} in CFM or 76.39×10^6 for V_{ideal} in m^3/s

K = 1 for two-stroke cycle and 2 for four-stroke cycle

Volumetric efficiency, defined as the ratio of actual air flow to ideal air flow, will vary with engine design, but an average of 80% may be used to determine the air cleaner size. Two-stroke cycle engines will require approximately 140% air flow calculated by the above equation. For supercharged and turbocharged engines, the air flow requirements should be obtained from the manufacturer.

The two basic cleaners available are wet and dry. The wet, or oil-bath, cleaner consists of either an oil wire mesh or an oil bath through which the air must pass. A dry cleaner uses a paper or cloth filter that traps dust, lint, and so on but allows the air to pass through.

The installation of a suitable air cleaner is important. In cases where adequate air can be supplied through proper ventilation of the area surrounding the engine, it is best to

mount the cleaner on the engine. If it is necessary to bring air to the engine from outside the area of the building, certain design factors must be considered. The pipe connections from an outside cleaner should be tight and mechanically strong, and fabric hose should not be used unless the length is relatively short. The air to the engine should not be heated by close proximity to the engine or any other heating device because, as previously mentioned, power loss occurs with air temperatures above 60°F (15.6°C). To avoid restrictions in the system, there should be no sharp bends in the piping. Finally, any outside air cleaner must be designed so moisture such as rain or snow cannot enter the system.

The air cleaner system on a turbocharged engine must be able to remove any impurities in the air that would be detrimental to the efficiency of the turbocharger. Because of the increased air requirements, a larger air-cleaner system must be used on such units.

EXHAUST SYSTEMS

An engine consumes a large volume of air for combustion, and that volume must be removed after combustion. It is necessary that the exhaust back pressure be kept at a minimum while this volume is being removed. The exhaust piping should be properly sized, and long sweep elbows should be employed if necessary. Unless the back pressure is kept low, the following conditions can result:

- Loss of power
- Poor fuel economy
- High combustion temperatures with increased maintenance
- High jacket water temperatures
- Crankcase sludging with resulting corrosion and bearing wear

All internal combustion engines create noise. Depending upon the location of the engine, this noise can be a problem. Normally, when measured from a distance of about 10 ft (3 m), an unmuffled engine creates a decibel noise level ranging from 100 for the small- and medium-size engine to 125 for the larger engine. Thus most engine installations incorporate some form of exhaust silencer, or muffler. Depending upon the degree of silencing, a muffler will reduce the unmuffled decibel reading by 30 to 35 dB. Various types are manufactured to meet the required conditions and are classified as follows:

- Standard or industrial
- Semiresidential (high-degree)
- Residential or hospital (supercritical)

The basic exhaust silencer is designed as either a dry type or wet type. The latter is used in installations such as sewage or water treatment plants, where the recovery of heat for plant processes is important. Basically this type may be classified as a low-pressure boiler. Water is admitted to the silencer through tubes or coils to pick up the heat from the exhaust. As shown in Figure 6, approximately 30% of the total heat input into the engine is exhausted. The wet-type silencer is designed to regain about 60 to 70% of this heat. This silencer has an advantage in its ability to operate either wet or dry. Thus when heat is not required, it may be operated dry and vice versa.

STARTING SYSTEMS

Starting methods for engines fall into two broad categories: direct and auxiliary.

Direct The direct system, used primarily with large engines, employs some means of exerting a rotating force on the crankshaft, such as the introduction of high-pressure air

directly into the cylinders of the engine. A direct system on small air- or water-cooled engines using either a rope or a hand crank has to a large extent been discarded in favor of an auxiliary method.

Auxiliary The auxiliary system employs a small gear that meshes with a larger gear (ring gear) on the engine flywheel. The ratio of the number of gear teeth on the large gear to those on the small gear is called the *cranking ratio*. Generally, the larger the ratio, the better the cranking performance. The auxiliary system uses several means to drive the small gear:

- Electric motor
- Air motor
- Hydraulic motor
- Auxiliary engine

ELECTRIC MOTOR The electric motor may be either dc or ac. The dc motor most commonly used is available in 6, 12, 24, or 32 V. The voltage size will depend upon the size of the engine, the ambient temperature, and the desired cranking speed of the engine. The dc system requires a source of outside power, usually in the form of a storage battery. To assure prompt starting, a charging system for the battery is required in the form of either a charging generator driven by the engine or an ac-powered battery trickle charger. The latter is recommended for standby installations where an engine-driven charging generator functions only when the engine is operating. During idle periods, the battery will lose its charge unless maintained by a trickle charger. In recent years, there has been a trend toward the use of an ac generator, or alternator, which has the advantages of small size, higher voltage and amperage, competitive price, and good charging ability under idle speed conditions.

Another type of electric motor is a line voltage starter available in 110, 220, or 440V ac. It has the advantages of faster and more powerful cranking, the elimination of the battery and charging system, less maintenance, and sustained cranking through unlimited available electric power. Its disadvantages are a higher initial cost, the requirement of high line voltage at the site, the danger to personnel due to the high voltage, and the requirement to conform to existing wiring and installation codes.

AIR MOTOR The air motor, which is usually of the rotary-vane type, uses high-pressure air in the range of 50 to 150 lb/in² (340 to 1030 kPa) to turn it in starting the engine. It is mounted on the engine flywheel housing to mesh with the gear on the flywheel in the same manner as the electric motor. An outside source of air from an air compressor, usually with a 250-lb/in² (1720-kPa) capacity, is required. A pressure-reducing valve is installed in the line to the engine. The high-pressure air stored in an adequate receiver is sufficient for several starting cycles. This starting system has the advantages of faster cranking, sustained cranking as long as the air supply lasts, suitability in hazardous locations where an electric system might be dangerous, and ability to operate on either compressed air or high-pressure natural gas. Its disadvantages include a higher initial cost, the requirement of an air-compressor system, and, finally, a shutdown condition if the air supply is depleted before the engine starts.

HYDRAULIC MOTOR The hydraulic motor system consists of the motor, an oil reservoir, an accumulator, and some means of charging the accumulator. The accumulator, which is a simple cylinder with a piston, is charged on one side with nitrogen gas. As the hydraulic fluid, usually oil, is pumped into the other side, the gas is compressed to a very high pressure. When released, the fluid turns the motor, which in turn rotates the engine. The system can be charged by hand, with an engine-driven pump, or with an electric motor-driven pump. Generally, an engine-driven or electric-motor-driven pump is used in conjunction with the hand pump in case of an engine or electrical failure. This system has the same basic advantages of the air motor except that there is no prolonged starting. If the engine is in good operating condition, the cranking is fast and a start is instantaneous, but, if not, it is necessary to recharge the system before another start can be made.

AUXILIARY ENGINE A small auxiliary air-cooled or water-cooled engine is sometimes employed for starting. It may be mounted on the engine in the same manner as the other systems, or a belt drive may be employed. Some form of speed reduction is required to reduce the higher speed of the auxiliary engine to that required for proper cranking. The principal advantage of such a system is a complete independence from outside sources of power, such as batteries, air, or pumps, but this is offset by a higher initial cost and the required regular maintenance of the engine.

IGNITION SYSTEMS

The internal combustion engine requires some means of igniting the combustible charge in the cylinder at the proper time. Today's high-compression gasoline and gas engines demand a system that will produce a high-tension spark across a short gap in the combustion chamber for the ignition of the charge. It is obvious that the design of the combustion chamber must be such that this combustible mixture of fuel and air is present between the discharge gap when the spark occurs; otherwise, ignition will not take place.

Ignition systems for gasoline or gas engines are considered high or low tension. The energy system for the high-tension system is either an electric generator and battery or a magneto. The generator and battery produce a direct current at 6 to 12 V potential, and the magneto produces an alternating current with higher peak voltages. With either energy source, this system has a primary circuit for the low-voltage current and a secondary circuit for the high-voltage current.

The primary circuit consists of the battery, an ammeter, an ignition switch, a primary coil, and breaker points and a condenser in the distributor. When the ignition switch and the breaker points close, a current flows through the circuit and builds up a magnetic field in the primary coil. Opening the breaker points breaks this circuit, causing the magnetic field to start collapsing. As the field collapses, it produces a current that flows in the same direction in the primary circuit and charges the condenser plates. The condenser builds up a potential opposing flow, which discharges back through the current. This results in a sudden collapse of the remaining magnetic field and the induction of a high voltage into the secondary winding of the coil. The breaker points are opened and closed by a cam which is engine-driven, usually at half engine speed.

The secondary circuit consists of the secondary coil winding, the lead to the distributor rotor, the distributor, the spark plug leads or wires, and the spark plug. A magneto eliminates the battery, but includes in its construction the balance of both primary and secondary circuits. It may have either a rotating coil and stationary permanent magnets or a stationary coil and rotating magnets. The relative movement of the primary coil winding and the magnets induces an alternating current in the primary circuit, the breaking of which induces a high-voltage current in the secondary circuit.

The development of the modern engine has required many refinements in spark plug design, but basically a spark plug consists of two electrodes, one grounded through the shell of the plug and the other insulated with porcelain or mica. The insulated electrode is exposed to the combustion. The heat flow occurs from this electrode to the spark plug shell through the grounded electrode.

Recent developments in ignition systems have produced the low-tension and breakerless systems. The breakerless system has eliminated most of the moving parts in the distributor system. The breaker points in the distributor system are actually a switch that opens and closes the primary circuit of an ignition coil. In the breakerless system, the use of solid-state devices provides a switch with no moving parts to wear or require adjustment.

The low-tension magneto system has been developed to reduce electrical stresses in the ignition circuit. The secondary coil has been removed from the magneto proper and relocated near each spark plug. The low voltage generated by the magneto is transmitted through the wiring harness to secondary coils, which then step up to the voltage to be transmitted through short leads to the spark plug. These leads may be insulated to withstand the stresses imposed upon them. This results in a minimum of electrical stresses with a resulting longer life of all components of the system.

As was mentioned previously under fuel systems, the diesel engine used the heat of compression for ignition; thus no auxiliary systems, such as the systems mentioned above, are required. The fuel is injected into the combustion chamber under relatively high pressure through the use of a fuel pump and injection nozzle. This system may be either an individual pump and nozzle for each cylinder, commonly called a *unit injection*, or a multicylinder pump that maintains a high pressure in a common fuel line connected to each injection nozzle. The latter is normally called the *common rail system*.

ENGINE INSTALLATIONS

Foundation The correct foundation, mounting, vibration isolation, and alignment are most important to the success of any engine installation. All stationary engines require a foundation or mounting base. There are many variations, but all basically serve to isolate the engine from the surrounding structures and absorb or inhibit vibrations. Such a base also provides a permanent and accurate surface upon which the engine and usually the pump may be mounted.

To meet these requirements, the foundation must be suitable in size and mass, rest on an adequate bearing surface, provide an accurately finished mounting surface, and be equipped with the necessary anchor bolts.

The size and mass of the foundation will depend upon the dimensions and weight of the engine and the pump (if a common base is considered). The following minimum standards should be followed:

1. Width should exceed the equipment width and length by a minimum of 1 ft (0.3 m).
2. The depth should be sufficient to provide a weight equal to 1.3 to 1.5 times the weight of the equipment. This depth may be determined by the following formula:

$$\text{In USCS units} \quad H = \frac{(1.3 - 1.5)W}{L \times B \times 135}$$

$$\text{In SI units} \quad H = \frac{(1.3 - 1.5)W}{L \times B \times 2162}$$

where H = depth of foundation, ft (m)

W = weight (mass) of equipment, lb (kg)

L = length of foundation, ft (m)

B = width of foundation, ft (m)

135 = density of concrete, lb/ft³ (2162 kg/m³)

The soil-bearing load in pounds per square foot (kilograms per square meter) should not exceed the building standard codes. It may be calculated by the formula

$$\text{Bearing load} = \frac{(2.3 - 2.5)W}{B \times L}$$

Foundation or anchor bolts used to hold the equipment in place should be of SAE grade No. 5 bolt material or equivalent. The diameter, of course, is determined by the mounting holes of the equipment. The length should be equivalent to a minimum embedded length of 30 times the diameter plus the necessary length for either a J or an L hook. An additional 5 to 6 in (13 to 15 cm) should be provided above the top surface of the foundation for grout, sole plate, chocks, shims, equipment base washers, and nuts, plus small variations in the surface level. Around the bolts, it is a good practice to place a sleeve of iron pipe or plastic tubing to allow some bending of the bolts to conform with the mounting hole locations. This sleeve should be about two-thirds the length of the

bolt, with its top slightly above the top surface of the foundation to prevent concrete from spilling into the sleeve.

Sole plates running the length of the equipment are recommended for mounting directly to the foundation. Made of at least $\frac{3}{4}$ -in (19-mm) hot- or cold-rolled steel and a width equivalent to the base-foot mounting of the equipment, they will provide a level means of mounting and will avoid variations in the level of the concrete. These plates should be drilled for the mounting holes and drilled and tapped for leveling screws, which will permit the plates to be leveled and held during the pouring of grout.

Alignment Although the alignment will vary with the type of engine and the pumping equipment, the basic objective remains the same. The driven shaft should be concentric with the driver shaft, and the centerlines of the two shafts should be parallel to each other. Rough alignment should be made through the use of chocks and shims. A dial indicator should be used to check deflection by loosening or tightening the anchor bolt nuts until there is less than a 0.005-in (0.13 mm) reading at each bolt. Shims should be added or removed to arrive at this point. A final check should be made with all the conditions "hot," as the engine and its driven equipment expand at the rate of 0.000006 in/°F (0.27 $\mu\text{m}/^{\circ}\text{C}$) above ambient hot to cold. Although the coupling, or driving member between the engine and the pump is not discussed in this section, it must be considered in the final alignment.

Vibration Isolation It is desirable to isolate the engine, and at times the pumping equipment, from the building structure because of vibrations. Cork (Figure 13) is used in the larger and heavier installations. A combination of cork and rubber pads may be used at each mounting hole on small- or medium-size installations, and spring isolators may be used on a complete installation if flexible hoses are used for fuel, water, and air connections where required. The manufacturer of the engine and the pumping equipment should be consulted in the use of any isolating material or device.

Vibrations are closely associated with the driving and driven equipment, couplings, and other connections. These linear vibrations may be caused by improper supports of the unbalanced parts, which produce a *torsional* vibration. An understanding of this vibration is important because its elimination is the responsibility of the engine and the driven-equipment manufacturer. It is complex and cannot be detected without the use of calculations and special instruments.

The basic concept involves an elastic element, such as an engine crankshaft, which tends to twist when any firing impulses are applied. When these forces are removed, the

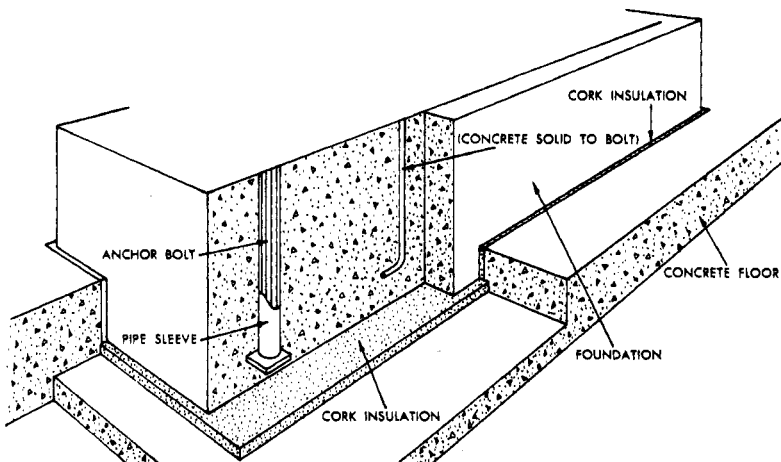


FIGURE 13 Engine foundation installation (Waukesha Motor)

elastic body will try to return to its original position. The driven mass and the connecting elements tend to resist these external impulses. The natural elasticity of the crankshaft and its connecting system allows a small amount of torsional deflection and tends to reduce the deflection as the external impulses reduce in force. Other reciprocating forces in the engine and external forces in the driven equipment may excite vibrations in the entire system. When all these forces come into resonance with the natural frequency of the entire system, torsional vibration will occur. This vibration may or may not be serious but, being complex, cannot be solved hastily.

The engine and pump manufacturer designs and constructs the product so critical harmonic vibrations will not be present under normal speeds and loads. However, there is no way to control the combination. An analysis of the complete system should be made. This requires a study of the mass elastic system of the combination, involving the mass weight and radius of gyration of all rotating parts. This study should be made either by the engine or pump manufacturer or by a torsional-analysis specialist.

FURTHER READING

Gunther, F. J. "Gas Engine Power for Water and Wastewater Facilities." *Water & Sewerage Works*, 112 and 113 (November 1965 to July 1966).

Heywood, J. B. "Internal Combustion Engine Fundamentals." McGraw-Hill Book Company New York, 1988.

9.1.4 HYDRAULIC TURBINES

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PUMPING APPLICATIONS

In many pumping applications, the liquid remains at a high pressure after it has completed its cycle. It is often economically desirable to recover some of the energy, which will otherwise have to be dissipated when the liquid is brought back to a lower pressure. Instead of running the liquid through a pressure-reducing valve to destroy the energy, a hydraulic turbine may be installed. This turbine can therefore assist in driving the process pumps. There are many such applications in use. Among these are the use of turbine-driven pumps in gas-cleaning operations. Here the solutions from scrubber towers are passed through turbines that in turn drive pumps. Some of the solutions involved in such a process are water saturated with carbon dioxide at a temperature of 40 to 70°F (4 to 21°C) and potassium carbonate containing dissolved carbon dioxide at a specific gravity of 1.31. A liquid at the much lower specific gravity of 0.84 has been used in power-recovery turbines in a glycol-ethylene hydration process.

One of the oldest applications of turbine-driven pumps was for fire-fighting equipment in mills having a natural head of water. The large volume of low head water was routed through a hydraulic turbine that drove the pump to pressurize a sprinkler system or provide high-pressure water to the fire-hose connections. Today, hydraulic turbines are used to drive pumps that generate fire-fighting foam.

At thermal power plants, cooling water returning from the cooling towers has been used to drive a hydraulic turbine directly connected to a pump that provides part of the cooling water. Naturally, in such an application, the power from the turbine alone is not enough to maintain the pumping system. Therefore an auxiliary power source is also required.

Turbines using oil pressure have been employed at thermal power plants. Large steam turbines obtain bearing oil and control oil pressure from a main feed pump that is directly connected to the steam turbine shaft. A small amount of the oil at this high pressure is used for the relay control, with the majority of the oil going to the bearings at a lower pressure. To reduce the oil pressure from that required by the relays to that needed by the

bearings, the oil is passed through a turbine driving a booster pump that pressurizes the oil at the intake of the main pump. At locations with high flood levels, it is preferable to locate pumps and electrical equipment above flood level. Some recent steam power plant installations have used the return from the condenser to the river (normally an appreciable drop in head) to drive a hydraulic turbine, which in turn may assist in driving the condensate pumps. Also, effluent from sewage treatment plants may have a significant discharge head.

It is often economical to use a small volume of water at a high head to move a large volume of water at a low head. Of course, the reverse application can also be accomplished. The former procedure is employed at hydroelectric projects where it is necessary to operate fishways to enable migrating fish to continue traveling upstream over the dam. The large volume of water is used both to attract the fish to the fish flume and to transport the fish to the fish ladders. An example of such an installation can be seen in Figure 1.

Turbines have also been used to start large pumping units when the hydraulic conditions are suitable. The impulse turbine, which develops maximum torque at zero speed, is especially useful for this application. The power required on such a starting turbine would be less than the power required on a starting motor to do the same job. Many electric controls can be eliminated when starting with a turbine. An illustration of this application is shown in Figure 2.

Desalinization plants have been built wherein salt is removed from seawater by pumping it through a membrane at high pressure. Only pure water goes through the membrane, with most of the sea water being used to carry away the salt. This excess high-pressure sea water is then put through a power-recovery turbine that helps to drive the high-pressure pumps.

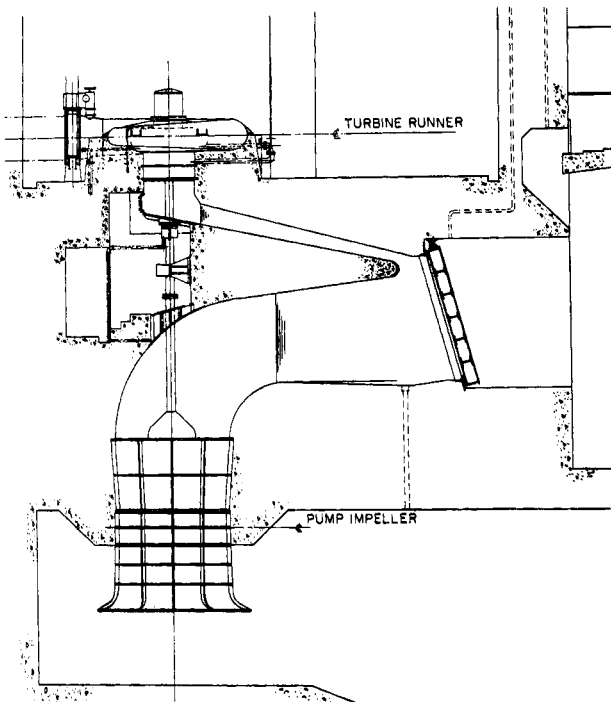


FIGURE 1 Turbine-driven fishway pumps. High head, low volume Francis-type turbine drives a low head, high-volume propeller pump at Rocky Reach, Washington.

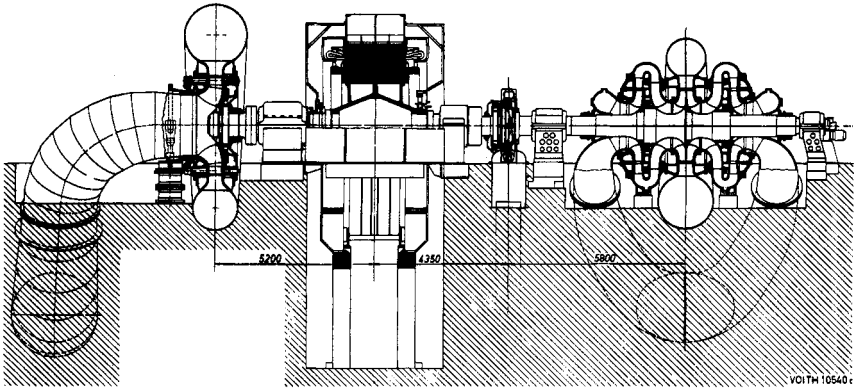


FIGURE 2 Impulse runner used to start a large pump. Left to right: Francis turbine, generator motor, starting impulse turbine, and two-stage double-suction storage pump. (Adapted from Voith Publication 1429, J. M. Voith GmbH, Heidenheim, Germany).

It is often desirable to have the turbine and pump running at different speeds, and this has been accomplished by means of a variable-speed coupling between turbine and pump. Reliable low-speed gear increasers or reducers of up to 35,000 hp (26,000 kW) are being built by many manufacturers. In applications such as starting a pump by means of a turbine, a mechanism may be needed to disengage the turbine after the pump has been started, in order to minimize windage and friction. Disengaging couplings may be hydraulic or mechanical and may be actuated when the unit is stationary or rotating, depending on the application. However, it is possible to allow the turbine to spin in air with the pump after the pump has been started. Figure 3 is an illustration of a turbine-drive pump arrangement.

TYPES OF TURBINES

Three types of turbines will be discussed: propeller (fixed and adjustable blades), Francis, and impulse. Figures 4 to 7 are illustrations of these turbine runners. Figure 8 shows a complete Francis turbine unit. The fixed- and adjustable-blade propellers and runners are essentially the same, with the exception that the adjustable is suited to a much wider range of loading conditions. This versatility is reflected in a higher manufacturing cost for the adjustable blade. The Francis runner is much like a centrifugal pump impeller running backward. The impulse runner (or Pelton wheel) is for high-head applications. The water is first channeled through a nozzle that directs a jet of water into the bowl-shaped runner buckets. This jet then discharges into the atmosphere. When a high back pressure is present in the housing, the impulse wheel cannot discharge properly. The performance drops off rapidly as the back pressure is increased. However, it is sometimes possible to admit low air pressure into the impulse runner housing to lower the level of the liquid surface to below the level of the runner. When the high turbine back pressure cannot be eliminated, a Francis turbine will perform much better than an impulse turbine. Figure 9 illustrates the relative efficiencies of the different types of runners. The fixed-blade propeller and Francis runners operate most efficiently in a range near full load, and operating time should be limited at low loads. The impulse and adjustable-blade runners, however, are designed for high efficiencies over a large load range. The adjustable-blade propeller accomplishes this by changing the angle of its blades by means of linkage in the hub of the runner. With the impulse runner, the size of the jet is controlled by the nozzle needle, which enables the runner to maintain high efficiencies at low loads.

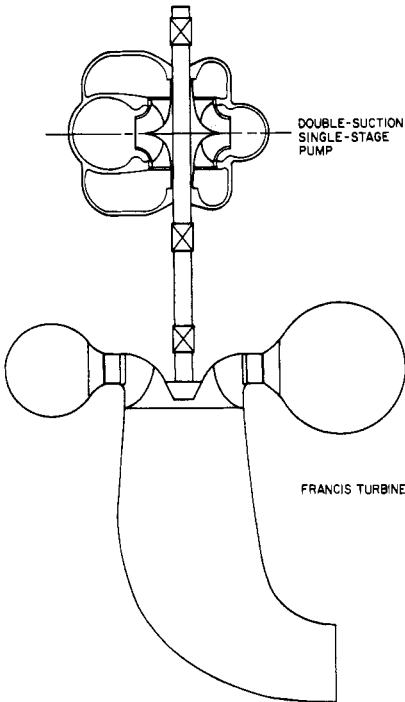


FIGURE 3 Double-suction single-stage pump driven by a Francis turbine (Voith Siemens Hydro)



FIGURE 4 Mixed flow propeller runner (Voith Siemens Hydro)

TURBINE ARRANGEMENTS

As is typical with pumps, hydraulic turbines are usually arranged with their power shafts either vertical or horizontal. In recent years, several low head installations have had inclined shafts to reduce excavation costs. The large (above 10,000 hp or 7,000 kW) vertical propeller turbines with concrete water passages have not been used with pumps. This is also true with very large, high head Francis turbines with steel-lined spiral shaped water passages. Mid-size and small hydraulic turbines of all three types may have vertical or horizontal shafts and usually steel water passages when used with pumps. Horizontal shaft arrangements have the advantage of better access to each piece of equipment. They usually require a greater floor area but lower power/pump house. For low heads (under 30' or 10 meters), propeller turbine installations have used inclined shafts with Bulb and Pit generator housings. TUBE turbines with a nearly straight draft tube have also been arranged with inclined shafts. For very low heads (under 15' or 5 meters), water wheels extending the width of a spillway are being considered where conventional hydraulic turbines are too costly.

Each project or site is likely to have special conditions, which will influence selection of the optimum turbine and pump arrangement.

Flow Units The size of hydraulic turbines has become so large that the earlier conventional flow quantity of cubic feet/minute (cfm) is today cubic feet/second (cfs) or cubic meters per second (cms). The following conversions are therefore useful:



FIGURE 5 Adjustable-blade propeller runner. "Kaplan" turbines have adjustable blades coordinated with adjustable wicket gates. (Voith Siemens Hydro)

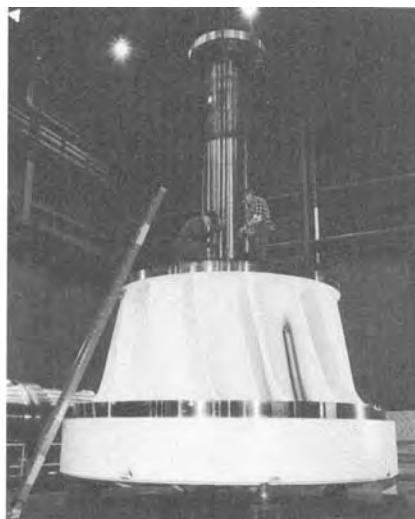


FIGURE 6 Francis runner has shorter buckets than a centrifugal pump impeller (Voith Siemens Hydro)

1 cfs = 449 gpm or 1.077 mgd

1 cubic foot = 7.48 US gallons

British imperial gallon = 1.2009 US gallon

1 cms = 35.3 cfs

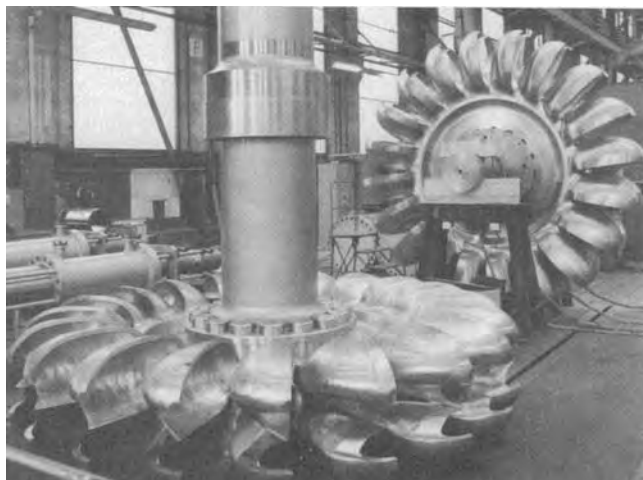


FIGURE 7 Impulse (or Pelton) runners (Voith Siemens Hydro)

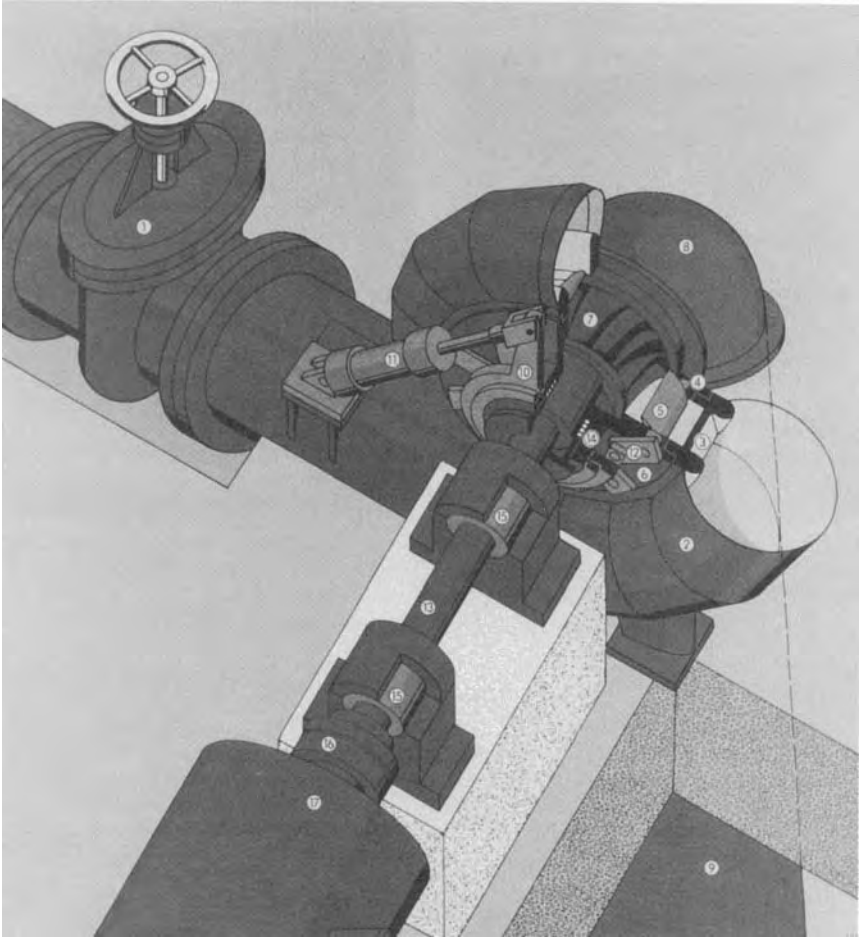


FIGURE 8 Complete standard Francis turbine unit with spiral case and elbow draft tube: 1) Shut-off valve, 2) Spiral case, 3) Stay ring, 4) Curb ring, 5) Guide vane, 6) Turbine cover, 7) Runner, 8) Draft tube elbow, 9) Draft tube, 10) Operating ring, 11) Servomotor, 12) Lever, 13) Turbine shaft, 14) Shaft seal, 15) Bearing, 16) Flexible coupling, 17) Generator (Voith Siemens Hydro)

Specific speed This is the speed at which a runner will rotate if the runner diameter is such that, under 1 ft (1 m) net head, it will develop 1 hp (1 kW):

In USCS units
$$N_s = \frac{\text{rpm} \times \text{hp}^{1/2}}{\text{ft}^{5/4}}$$

In SI units
$$N_s = \frac{\text{rpm} \times \text{kW}^{1/2}}{\text{m}^{5/4}}$$

The specific speed is an important factor governing the selection of the type of runner best suited for a given operating range. The impulse wheels have very low specific speeds

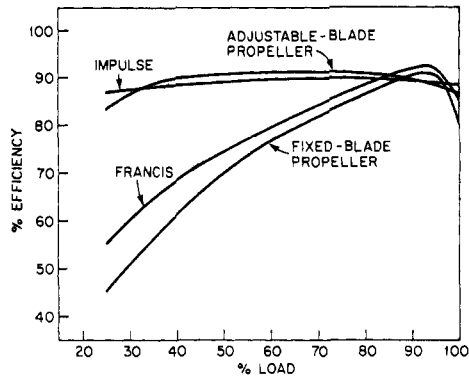


FIGURE 9 Efficiency versus load for different runner types

relative to propellers, and the specific speed of a Francis turbine lies between the impulse and propeller. Table 1 shows some statistics for existing units.*

SIZES AND RATINGS

Although power-recovery turbines are relatively small, there are much larger turbines in existence that could be used to drive pumps. Turbines of the following sizes do exist: propeller, 405.5-in (10.3-dm) diameter; Francis, 288-in (7.3-m) diameter; and impulse, 176-in (4.5-m) diameter. The extreme range of power, head, and speed is shown in Table 2.

GENERAL CHARACTERISTICS

Sigma As in pumps, the problem of cavitation also exists in turbines. Sigma is defined as follows:

$$\sigma = \frac{H_a - H_s}{H}$$

where H_a = feet (meters) of atmospheric pressure minus vapor pressure

H_s = distance in feet (meters) the centerline of the blades is above tailwater for vertical units or distance in feet (meters) the highest point of the blade is above tailwater for horizontal units

H = feet (meters) of elevation between inlet head water surface and tailwater surface elevation

Critical sigma is defined as that point at which cavitation begins to affect the performance of the turbine. Figure 10 shows the relationship between critical sigma and specific speed and also shows which type of runner is best for a given specific speed. The impulse

*Universal specific speed Ω_s (defined in Chap. 1 and Section 2.1.1) is found from these turbine N_s -values as follows:

$$\Omega_s = \frac{N_s(USCS)}{4.344\sqrt{\eta_t \times \text{sp. gr.}}} = \frac{N_s(SI)}{16.564\sqrt{\eta_t \times \text{sp. gr.}}}$$

where η_t is the turbine efficiency in percent.

TABLE 1 Turbine statistics of existing units

Speed, rpm	Power, hp ^a	Head, ft ^b	N _s , USCS ^c	Type ^d	Use ^e	Supplier ^f
450	2,000	925	3.94	I	SUP	BLH
450	450	485	4.19	I	SUP	BLH
750	170	575	3.47	I	SUP	BLH
340	570	485	3.57	I	SUP	BLH
1,775	264	1,085	4.63	I	SUP	BLH
437.5	425	405	4.97	I	SUP	BLH
1,770	640	960	8.38	I	SUP	BLH
1,775	230	1,085	4.32	I	SUP	BLH
720	885	460	10.05	I	SUP	BLH
900	640	866	4.85	I	SUP	BLH
3,550	85	1,920	2.58	I	SUP	BLH
690	535	86.5	60.50	F	FW	BLH
126	670	80	13.63	F	FW	BLH
450	2,600	118	58.90	F	SUP	LEF
525	1,050	83.5	67.39	F	SUP	LEF
900	750	123	60.18	F	SUP	LEF
1,000	375	96	64.43	F	SUP	LEF
700	53	150	9.71	I	SUP	LEF
750	160	30	135.13	P	SUP	LEF
882	128	70	49.26	F	SUP	LEF
1,750	145	135	45.78	F	SUP	LEF
1,750	220	135	56.39	F	SUP	LEF
280	200	14	146.20	P	SUP	A-C
550	400	35	129.21	P	IR	A-C
3,450	7.5	196	12.89	I	PP	A-C
1,775	353	1,085	5.36	I	ST	A-C
1,185	280	231	22.01	I	ST	A-C
1,775	300	1,510	3.27	I	CH	A-C
1,300	1.6	150	3.13	I	PP	A-C
700	1,760	798	6.92	I	ST	A-C
600	392	460	5.58	I	SUP	A-C
122	973	65	20.61	F	FW	A-C
108	1,200	75	16.95	F	FW	A-C

^akW = 0.746 hp^bm = 0.3048 ft^cN_s (SI) = 3.814 N_s (USCS)^dI = impulse, P = propeller, F = Francis^eSUP = supplement electric motor to drive pump, FW = drives fishway pumps, ST = scrubber-tower application, PP = turbine drives petroleum pumps, CH = power recovery in chemical plant, IR = irrigation project.^fBLH = Baldwin-Lima-Hamilton Corp., LEF = The James Leffel Co., A-C = Allis-Chalmers Corporation, since 1986 Voith Hydro, Inc., also American Hydro Corp.

wheel is not affected by sigma because it is a free jet action and therefore not subject to low-pressure areas.

Affinity Laws The relationships between head, discharge, speed, power, and diameter can be seen in the following equations, where Q = rate of discharge, H = head, N = speed, P = power, D = diameter, and subscripts denote two geometrically similar units with the same specific speed:

TABLE 2 Range of power, head, discharge, and speed of existing units of one manufacturer

	Propeller		Francis		Impulse	
	Low	High	Low	High	Low	High
Power hp ^a	82.5	268,000	1.2	820,000	1.6	330,000
Head ft ^b	6.0	180	4.0	2,204	75.0	5,790
Speed, rpm	50	750	56.4	3,500	180	3,600

^akW = 0.746 hp

^bm = 0.3048 ft

$$\frac{Q_1}{N_1 D_1^3} = \frac{Q_2}{N_2 D_2^3}$$
$$\frac{Q_1^2}{H_1 D_1^4} = \frac{Q_2^2}{H_2 D_2^4}$$
$$\frac{N_1^2 D_1^2}{H_1} = \frac{N_2^2 D_2^2}{H_2}$$
$$\frac{P_1}{N_1^3 D_1^5} = \frac{P_2}{N_2^3 D_2^5}$$

Most designs used are tested as exact homologous models, and performance is stepped up from the model by the normal affinity laws given above. Because of difficulties in measuring large flows at the field installation, only approximate or relative flow metering is normally done.

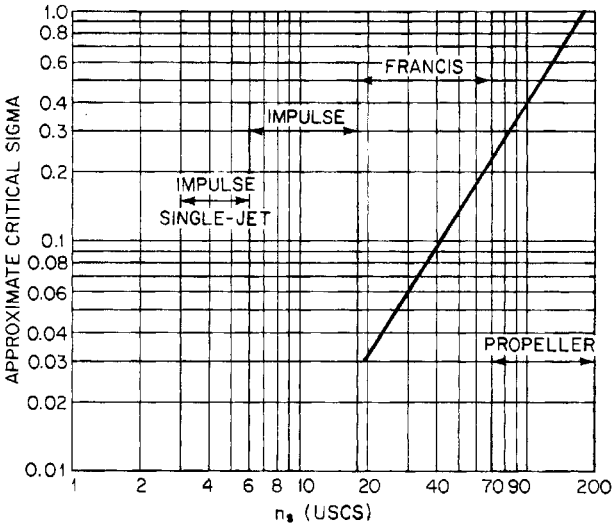


FIGURE 10 Critical sigma versus N_s . Specific speed N_s (SI) = $3.814 N_s$ (USCS)

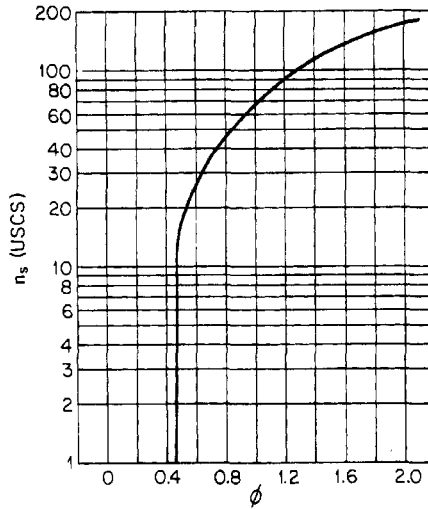


FIGURE 11 Specific speed versus ϕ . Specific speed N_s (SI) = $3.814N_s$ (USCS)

Speed Characteristic This is defined as the peripheral speed of the runner divided by the spouting or free discharge velocity of the water:

$$\phi = \frac{u}{(2gH)^{1/2}}$$

where ϕ = speed characteristic

u = peripheral speed of runner, ft/s (m/s)

H = head, ft (m)

$g = 32.2 \text{ ft/s}^2 (9.807 \text{ m/s}^2)$

A plot of ϕ versus speed can be seen in Figure 11, which shows that ϕ becomes constant at after the specific speed has dropped into the impulse-runner region. Theoretically, for maximum energy conversion, ϕ should equal 0.5 for impulse runners. However, because of small losses in the runner, this value is set at approximately 0.46. Control of the hydraulic turbine can be accomplished by means of a governor or a flow or load control system.

Torque All hydraulic turbines have maximum torque at zero speed; therefore, they have ideal starting characteristics. The pump can be accelerated to design speed by gradually opening the turbine gates or inlet valve while keeping within the limitations of hydraulic transients.

DATA REQUIRED FOR TURBINE SELECTION

The turbine supplier should have the following information in order to select the best combination of size and type runner:

1. Head available, head range, and head duration.
2. Power and speed required to drive pump.

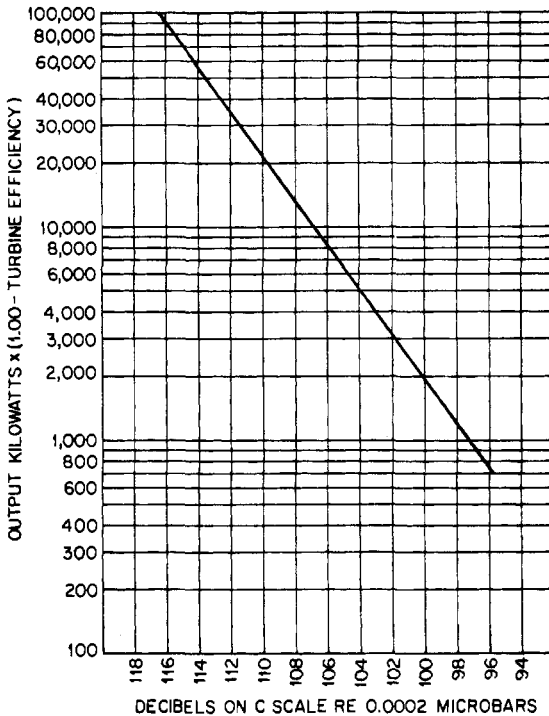


FIGURE 12 Sound level readings taken 3 ft (0.9 m) from main shaft. Hydraulic losses are approximated output in kilowatts $\times$ (1.00 - turbine efficiency). (From L. F. Henry: "Selection of Reversible Pump/Turbine Specific Speeds." Paper presented at meeting on pumped-storage development and its environmental effects, University of Wisconsin, September 1971.)

3. Description of fluid to be handled, including chemical composition and specific gravity.
4. Possibility of adding air to system. Turbine will operate satisfactorily without air, but air may be added to system to reduce pressure fluctuations (normally one-third of rpm in frequency) at part load and possibly to smooth unit operation at full load.
5. If corrosive fluid is to be handled, description of the materials required.
6. Controls from pumping process that will affect turbine operation. If speed is a controlling factor, the possibility of using a speed-varying device should be considered.
7. Back pressure on the turbine.
8. Noise-level trends for the turbine. Figure 12 illustrates noise levels recorded on some rather large turbine units. Ear protection is required for noise levels above 85 db.

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9.1.5 GAS TURBINES

RICHARD G. OLSON

THERMODYNAMIC PRINCIPLE AND CLASSIFICATIONS

The gas turbine is an internal combustion engine involving steady flow processes carried out in separate components. The basic processes involve compression of air in a compressor, introduction of the compressed air and fuel into the combustion chamber(s), and finally expansion of the gaseous combustion products in a power turbine. Figure 1 illustrates a simple gas turbine.

The Brayton, or Joule, Cycle Figure 2 shows an ideal Brayton, or Joule, cycle illustrated in *PV* and *TS* diagrams. This cycle is commonly used in the analysis of gas turbines.

The inlet air is compressed isentropically from point 1 to point 2, heat is added at an assumed constant pressure from point 2 to point 3, the air is then expanded isentropically in the power turbine from point 3 to point 4, and finally heat is rejected at an assumed constant pressure from point 4 to point 1. From Figure 2, the compressor work is $h_2 - h_1$, the turbine work is $h_3 - h_4$, and the difference is the net work output. The heat input is $h_3 - h_2$. One can then derive an expression for thermal efficiency as follows:

$$\eta = \frac{(h_3 - h_4) - (h_2 - h_1)}{h_3 - h_2}$$

In reality, the cycle is irreversible and the efficiency of the compression, combustion, and expansion must be taken into account. However, an examination of the thermal efficiency equation shows the need of high compressor and turbine efficiencies in order to produce an acceptable amount of work output. The importance of h_3 is also readily apparent. In noting that h_3 is directly proportional to temperature, one can appreciate why continuous

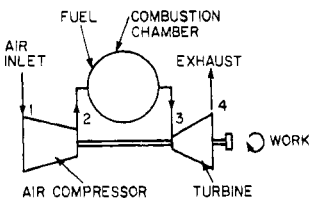


FIGURE 1 Components of simple gas turbine

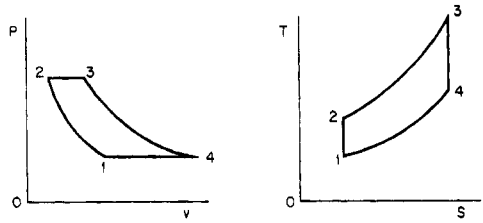


FIGURE 2 PV and TS diagrams for an ideal Brayton, or Joule, cycle

emphasis is being placed on the development of materials and techniques that permit higher combustion temperatures and correspondingly higher turbine inlet temperatures.

Classifications In an *open-cycle* gas turbine, the inlet air mixes directly with the combustion products and is exhausted to the atmosphere after passing through the power turbine. The *closed-cycle* gas turbine uses a heat exchanger to transfer heat to the working fluid, which is continuously recirculated in a closed loop. A *combined cycle* uses the principles of both the open and closed cycles. In current practice, the combined cycle uses an open cycle to provide shaft work while the heat from the exhaust is partially recovered in a waste-heat boiler. The heat recovered then proceeds through a standard steam power cycle until heat is rejected to the most readily available low-temperature reservoir.

Of the above configurations, the open-cycle gas turbine is most extensively used today for driving centrifugal pumps. This is probably due to the important consideration of minimum capital investment for each power output.

Two distinct types of open-cycle gas turbines have evolved: the *single-shaft* and *split-shaft* versions. The single-shaft gas turbine was developed primarily for the electric power industry and uses a compressor and a power turbine integrated on a common shaft. As the unit is used continuously at a single rotational speed, the compressor and power turbine efficiency can be optimized.

The split-shaft gas turbine was developed primarily for mechanical drive applications where output power and speed might be expected to vary. Figure 3 illustrates such a turbine. A typical curve of power output versus shaft speed is illustrated in Figure 4.

Split-shaft gas turbines are available in *conventional* and *aircraft-derivative* versions. The conventional gas turbine evolved from steam turbine technology and is illustrated in Figure 5. Figure 6 shows a modified jet engine used as a source of hot gas to a power turbine.

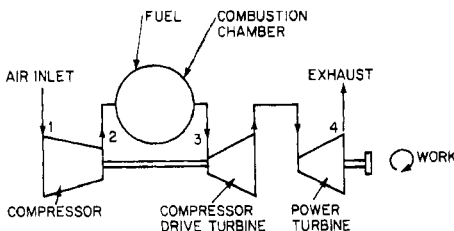


FIGURE 3 Components of a split-shaft gas turbine

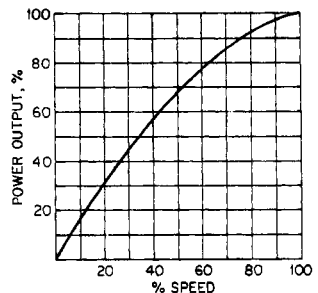


FIGURE 4 Typical curve of power output versus shaft speed for a split-shaft gas turbine

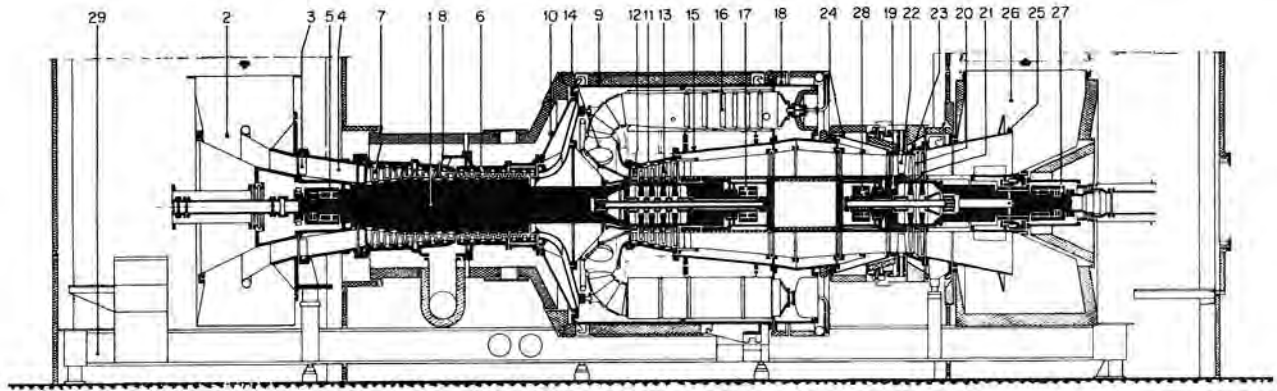


FIGURE 5 Gas turbine cross section: (1) gas generator turbine rotor, (2) inlet air casing, (3) compressor inlet guide vanes, (4) conical inlet casing with bearing support, (5) compressor radial-axial bearing, (6) compressor stator, (7) compressor stator blades, (8) compressor rotor blades, (9) combustion chamber main casing, (10) intermediate conical casing with diffuser, (11) gas generator turbine stator blade carrier, (12) gas generator turbine stator blades, (13) gas generator turbine rotor blades, (14) hot gas casing, (15) intermediate casing, (16) nine incorporated combustion chambers, (17) compressor radial bearing, (18) central casing, (19) power turbine rotor, (20) power turbine stator blades, (21) power turbine rotor blades, (22) power turbine adjustable rotor blades, (23) power turbine stator blade carrier, (24) conical intermediate piece, (25) outlet diffuser, (26) outlet casing, (27) power turbine radial-axial bearing, (28) power turbine radial bearing, (29) metallic foundation (Turbodyne)

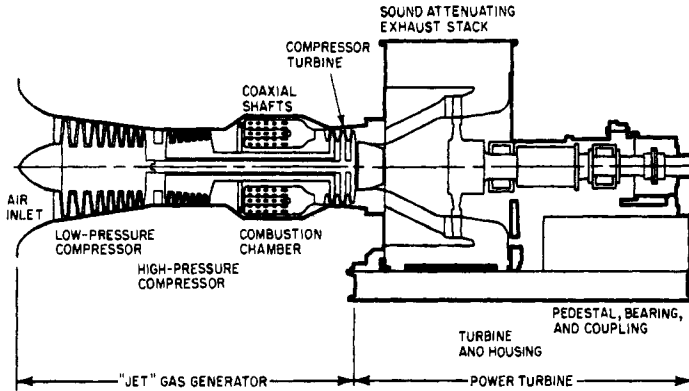


FIGURE 6 Cross section of aircraft-derivative gas turbine driver (Turbodyne)

Note that the jet engine combines the compression, combustion, and power turbine necessary to drive the compressor.

RATINGS

In the evolution of the gas turbine as a prime mover, various organizations have put forth standard conditions of inlet temperature and elevation to allow direct comparison of various gas turbines.

Four common standards exist:

ISO (International Standards Organization): sea level and 59°F (15°C)

NEMA (National Electrical Manufacturers Association): 1000 ft (304.8 m) above sea level and 80°F (27°C)

CIMAC (Congres International des Machines a Combustion): sea level and 59°F (15°C)

Site: actual elevation and design temperature at installation site

With the evolution of higher combustion temperatures and with the greater need for power over relatively short daily periods, new ratings have developed: emergency (maximum intermittent), peaking (intermittent), and base load. These classifications are based on the number of hours per unit of time that a gas turbine is operated and are related to the material used in the power turbine blading. A common standard is to use materials suitable for 100,000 h of continuous operation. Higher temperatures are permitted, but at the sacrifice of the life of the material and an increase in maintenance costs.

In a pump-driving application, the cycle of operations should be considered in specifying a gas turbine driver. A typical curve of gas turbine output as a function of inlet temperature (Figure 7) clearly indicates the necessity of specifying an accurate design temperature. Figure 8 is a typical correction curve for altitude.

FUELS

A wide range of fuels, from natural gas to the bunker oils, may be burned in simple-cycle gas turbines. In most cases, units can operate on gas or liquid fuels, and some turbines have automatic switchover capability while under load.

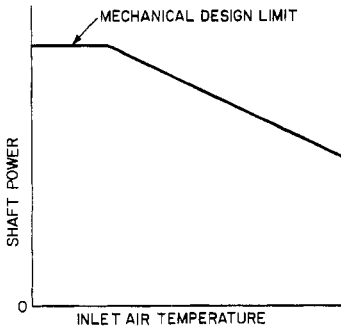


FIGURE 7 Typical curve of gas turbine output versus inlet temperature

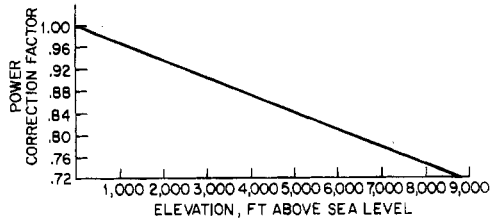


FIGURE 8 Typical curve of gas turbine power correction versus altitude (ft $\times$ 0.3048 = m)

Although it would appear advantageous to choose the cheapest fuel available, there may be disadvantages in such a choice: impurities such as vanadium, sulfur, and sodium definitely result in high-temperature corrosion; solid impurities and a high ash content will lead to erosion problems. One is therefore often faced with a trade-off between the cost of treating a fuel and the higher original cost.

With ever-increasing firing temperatures, research is continuing in the field of high-temperature corrosion with or without accompanying erosion problems. Present approaches to the problem include the use of inhibitors in the fuel as well as coatings for power turbine blades. Another solution that has been used on some crude oil pipelines is the topping unit. This is a small still that removes a specific fraction of the crude oil for burning as gas turbine fuel. It is good practice to include a detailed fuel analysis as part of any request for bids.

ENVIRONMENTAL CONSIDERATIONS

Legislation at both the federal and state levels has been concerned with the protection of our environment. An engineer must now consider the effects of the project on the environment.

Noise Any piece of dynamic mechanical equipment will emit airborne sounds that, depending on frequency and level, may be classified as noise.

The primary sources of noise in the gas turbine are the turbine inlet and exhaust and the accessories and support systems, such as fin fan lubrication oil coolers, auxiliary air blowers and fans, starting devices, and auxiliary lubrication oil pumps. It is common practice for manufacturers to provide inlet and exhaust silencers as well as some form of acoustic treatment for auxiliaries.

In considering the degree of acoustic treatment necessary for each installation, the practicing engineer must consider the following parameters:

1. Federal law (Walsh-Healy Act) limiting the time a worker may spend in a noisy environment
2. Local codes and their interpretation
3. Plant location and existing noise level at site
4. Site topography, including any noise-reflective surfaces
5. Applicable ASME standards and other relevant standards

Emissions The federal air quality acts require each state to develop a plan for achieving satisfactory air quality. Specifically, goals have been put forth to limit suspended particulate temperature matter and oxides of sulfur and nitrogen. Therefore,

the engineer must investigate existing regulations during the planning stage of a pump installation.

In general, gas turbines have low particulate emissions. The amount of sulfur oxides exhausted to the atmosphere is in direct proportion to the content of sulfur in the fuel, and current practice calls for elimination at the source. The formation of oxides of nitrogen is a direct result of combustion temperatures. Dry low-NO_x emission combustion exist based on lean combustion in pre-mixed burners.

GAS TURBINE SUPPORT SYSTEMS

Starting Systems A form of mechanical cranking is necessary to bring a gas turbine up to its self-sustaining speed. The amount of energy necessary will depend on each manufacturer's design. Available systems include electric motors, diesel engines, and gas-expander turbines.

Lubrication The gas turbine manufacturer normally provides a combined pump-turbine lubrication oil system. A main lubrication oil pump of sufficient capacity for the combined system is necessary, in addition to a standby pump in the event of failure of the main pump. A reservoir sized for a retention time of at least 4 minutes is usually specified. Filtration to 10- μ m particle size should be adequate for most gas turbines and pumps.

Lubrication oil can be cooled by various means, the selection of which depends upon local conditions. The simplest and cheapest method uses a shell-and-tube heat exchanger with water as the cooling medium. In arid regions, a fin fan cooler with direct air-to-water cooling is commonly used.

Inlet Air Filtration The degree of inlet air filtration needed is primarily a function of the size and number of particles in the atmosphere surrounding the installation. In most cases, a simple tortuous-path precipitator will suffice. For dirtier atmospheres or in arid regions where sandstorms occur, inertial separators followed by a rolling-media filter should be considered.

Any filtration will result in a loss in performance because of the pressure drop across the inlet filter. Conservative design practice calls for a face velocity of 500 ft/min (150 m/min) for a rolling-media filter and 1700 ft/min (520 m/min) for a tortuous-path precipitator.

Control Most manufacturers offer control packages which provide proper sequencing for automatic start-up, operation, and shutdown. During automatic start-up, the sequencer receives signals from various transmitters to ensure that auxiliaries are functioning properly and, with the aid of timers, brings the unit on line through a planned sequence of events.

Key operating parameters—for example, output speed, gas turbine compressor speed, turbine inlet temperature, lubrication oil temperature and pressure—are continually monitored during normal operation. Signals from a pump discharge-pressure transmitter can be fed into a speed and fuel controller to cause the unit to respond to changes in speed and output.

As with start-up, normal and emergency shutdowns are accomplished through the sequencer. Most standard control packages are easily adaptable to remote control by means of cable or microwave.

APPLICATION TO PUMPS

Gas turbines are available to drive centrifugal pumps in a wide range of speeds and sizes, from 40 hp (30 kW) to over 20,000 hp (15,000 kW). It is not practical to list here all the

available units because they are too numerous and are continually being upgraded and added to. A listing can be found in *Sawyer's Gas Turbine Catalog*, which is published annually.

Pipeline Service Crude oil pipeline service has seen a tremendous growth in the use of gas turbine drivers since the mid-1950s, and this trend will continue as crude oil production becomes more and more remote from its markets. The advantages of the gas turbine for this application are as follows:

1. Installed cost is usually lower than that of a corresponding reciprocating engine.
2. Variable-speed operation allows maintenance of a specific discharge pressure under a wide range of operating conditions, thus achieving maximum flexibility.
3. Normal gas turbine control system is easily adapted to unattended operation and remote control.
4. Operating experience has proved the gas turbine to have a high degree of reliability.
5. Gas turbines can be packaged into modules for ease of transportation and erection.

Water Flooding An important aspect of oil production is the use of secondary recovery methods to increase the output of crude oil reservoirs whose pressure does not allow the crude to flow freely to the surface. Flooding the reservoir with high-pressure water has been a primary technique for years. The development of high-pressure centrifugal pumps has allowed increased water flow into oil fields.

As most oil production is located in remote areas, the packaged gas turbine driver has become increasingly popular. Figure 9 shows a typical 3300-bhp (2460-brake kW) split-shaft gas-turbine-driven centrifugal water flood pump package. A cutaway view of a typical 1100-hhp (820-brake kW) gas turbine is shown in Figure 10; note the reduction gearing at the exhaust end for direct driving of a pump. Supporting systems, such as starter, lubrication oil pumps, governor, and fuel oil pumps, are driven off the accessory pods located at the air inlet end. Offshore platform installations require drivers with a minimum vibration as well as small, unbalanced inertial forces. The gas turbine fits both of these descriptions very well.

Cargo Loading Another interesting application of this prime mover is in the field of cargo loading, where units are currently in operation charging tankers with crude oil. Selection of a gas turbine pumping unit with critical speeds above the normal operating range allows great flexibility of operation, particularly during final topping operations.

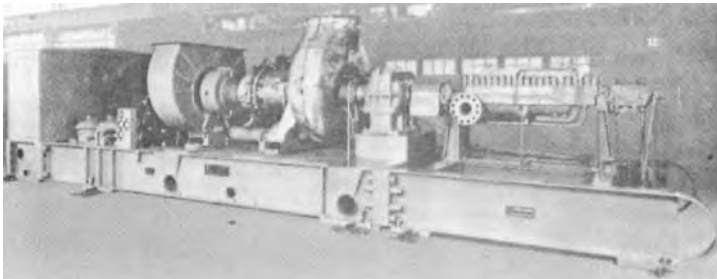


FIGURE 9 A 3300-bhp (2460-bkW) gas turbine-driven waterflood package (Solar Division of International Harvester)

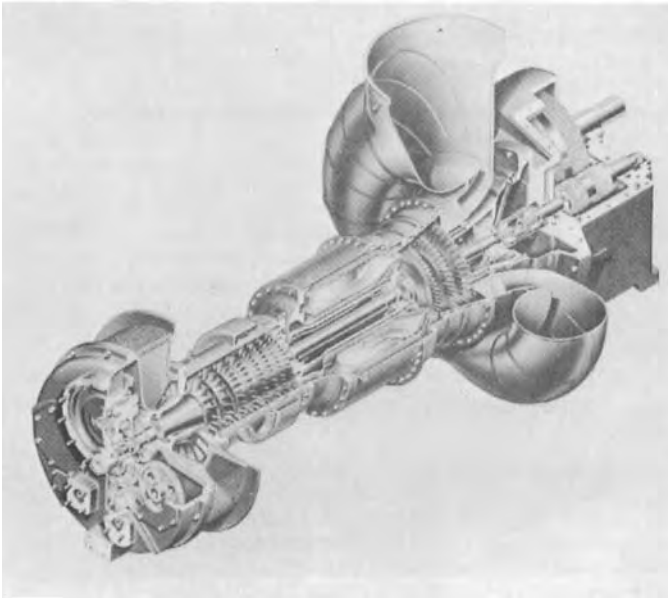


FIGURE 10 Cutaway view of 1100-bhp (820-kW) split-shaft gas turbine driver (Solar Division of International Harvester)

Application Considerations The application of gas turbine drivers can vary from a simple driver for one pump operating at constant flow and discharge pressure to a multiplicity of units operating at variable speed on a pipeline. For the purposes of this discussion, it is assumed that the pumping system has been analyzed, a pump selection has been made, and all possible operating conditions have been analyzed so brake horsepower (brake kilowatt) and speed requirements are known.

The brake horsepower (brake kilowatt) output of the gas turbine must equal or exceed that required by the pump. This output can be determined by the use of specific performance curves similar to Figure 7 as corrected for elevation (Figure 8). Gear losses as necessary are added to the brake horsepower (brake kilowatt) required by the pump. Intermediate brake horsepower (brake kilowatt) and speed requirements should then be checked against a gas turbine output versus speed curve (Figure 4).

A torsional analysis of the combined unit is made (usually by the gas turbine manufacturer) to ensure the absence of any critical speeds in the operating range. Table 1 is a typical data sheet recommended for use when purchasing a gas turbine to drive a centrifugal pump.

TABLE 1 Typical data sheet for gas turbine driven centrifugal pump

Information from purchaser	Information from manufacturer
Pumping requirements	Pumping requirements
Service _____	Service _____
Liquid _____	Liquid _____
Pumping temp. _____	Pumping temp. _____
Capacity (total) normal/max _____/_____	Capacity (total) normal/max _____/_____
No. pumps operating _____	No. pumps operating _____
Specific gravity at pump temp. _____	Specific gravity at pump temp. _____
Viscosity at pumping temp. _____	Viscosity at pumping temp. _____
Total head _____	Total head _____
NPSH available _____	NPSH available _____
Pump type, materials, and accessories	Pump type, materials, and accessories
Site conditions	Pumping requirements
Elevation, ft (m) _____	required _____ normal _____ max
Range of site ambient temperature	Speed, rpm _____ normal _____ max
_____ Dry bulb _____ Wet bulb	Efficiency _____ normal _____ max
Design, °F (°C) _____	Pump type, materials, and accessories
Maximum, °F (°C) _____	Gas turbine excluding gear
Minimum, °F (°C) _____	Design Max Min
Atmospheric air	Dry bulb temp., °F (°C) _____
Dust below 10 μ m, ppm _____	Output, hp (kW) _____
10 μ m and above, ppm _____	Heat rate, Btu/hp · h (J/kWh) (LHV) _____
Corrosive constituents:	Output shaft speed, rpm _____
Sulfur, ammonia, ammonium salts, salt or seacoast, other _____	Turbine inlet temp., °F (°C) _____
Noise specifications	Gear loss _____
City, state, federal, other _____	Net hp (kW) _____
Emission specifications	Total utility consumption
City, state, federal, other _____	Cooling water, gpm (1/s) _____
Utilities available at site	Electric power, kW ac/dc _____
Steam: Pressure, lb/in ² (kPa) _____	Steam, lb/h (kg/h) _____
Temp., °F (°C) _____	Compressed air, standard ft ³ /min (m ³ /min) _____
Quantity, lb/h (kg/h) _____	Shipping data
Electricity	Aux. items
V Phase Cycles	Turbine
ac _____	Shipping wt, tons (kg) _____
dc _____	Max erection wt, tons (kg) _____
Cooling water	Max maint. wt, tons (kg) _____
Source _____ quality _____	Length, ft-in (m) _____
Supply temp. _____ min _____ max	Width, ft-in (m) _____
Supply press, lb/in ² (kPa) _____	Height, ft-in (m) _____
Max return, °F (°C) _____	

TABLE 1 Typical data sheet for gas turbine driven centrifugal pump (*Continued*)

Information from purchaser	Information from manufacturer	
Fuel Gas _____ liquid _____	Accessories included (as required by purchaser)	
Analysis attached _____	Inlet air filter	yes no
Accessory items required (see list in right-hand column)	Inlet air silencer	yes no
	Exhaust silencer	yes no
	Exhaust duct	yes no
	Starting equipment	yes no
	Load gear (if required)	yes no
	Driven pump	yes no
	Coupling	yes no
	Fire protection system	yes no
	Equipment enclosure	yes no
	Baseplate or soleplates	yes no
	Combined turbine-pump	yes no
	Lub. oil system	yes no
	Main lube pump	yes no
	Auxiliary lub. pump	yes no
	Lub. reservoir	yes no
	Lub. filter	yes no
	Lub. oil cooler	yes no
	Unit control panel	yes no
	Auxiliary motor	yes no
	Control center	yes no

Source. Adapted from API Standard 616.

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SECTION 9.2

SPEED-VARYING DEVICES

9.2.1 PERMANENT MAGNET ADJUSTABLE SPEED DRIVES

**G. S. HIGHFILL
L. A. HALVERSON**

OVERVIEW

The permanent magnet adjustable speed drive is a coupling used where there is a need to connect a constant-speed driver to a load and obtain variable-speed operation. Pump drivers are usually electric motors and they inherently operate at a fixed speed. The inclusion of a permanent magnet adjustable speed drive into the drive train will transmit rotary motion and torque, and allow variable-speed adjustment for the pump/load. Variable speed technology has resulted in significant energy savings, improved pump performance, and reduced maintenance costs, compared with constant speed alternatives.

Two types of magnetic slip couplings are available in industry and they both function similarly. The heritage (earlier technology) product known as the eddy-current coupling is an electromechanical torque-transmitting device. The latest technology product known as the permanent magnet adjustable speed drive is a mechanical torque-transmitting device.

Both work on the principle of magnetic induction and torque is generated as a function of slip between the input and output members. The primary difference between these two types of devices is how the magnetic field is generated—one uses high strength rare-earth permanent magnets and the other uses a field winding that is excited by direct current, usually from a static power supply. The electrical power source is eliminated when using permanent magnets. Another difference is how the device is mounted in the drive train. Many permanent magnet adjustable speed drives can be shaft mounted, similar to a conventional spacer type coupling. This eliminates the base-mounted housing required for eddy-current couplings.

PERMANENT MAGNET ADJUSTABLE SPEED DRIVES

The permanent magnet adjustable speed drive (ASD) is a mechanical torque-transmitting device. The magnetic technology utilizes powerful rare-earth permanent magnets to transmit torque across an air gap; therefore, the input and output members are mechanically independent. A permanent magnet ASD is nonenergized. The drive unit consists of two primary component assemblies—a magnet rotor that revolves freely within a conductor rotor. When the driver rotates, the relative motion and interaction between the magnet rotor and conductor rotor creates eddy currents that develop torque across the air gap. Varying the width of the air gap provides adjustable torque and variable output speed to the load. A typical shaft-mount air-cooled permanent magnet ASD is shown in Figure 1.

Rare-earth permanent magnets are made of a combination of neodymium, iron, and boron (NdFeB). The magnets are typically plated for corrosion protection. NdFeB magnets have a high energy product, permitting small size and high torque transmission capability, and they have a half life of over 2000 years. NdFeB magnets will retain their performance up to 250 to 350°F, but they can demagnetize at prolonged exposure to higher temperatures.

The magnet rotor assembly contains two aluminum magnet rotors populated with several permanent magnets and is connected to the load (output) shaft. An even number of magnets installed with opposing polarity (north and south in turn) is normal. The conductor rotor assembly is comprised of copper disks mounted to steel plates and is connected to the driver (input) shaft and rotates at constant speed. When the conductor

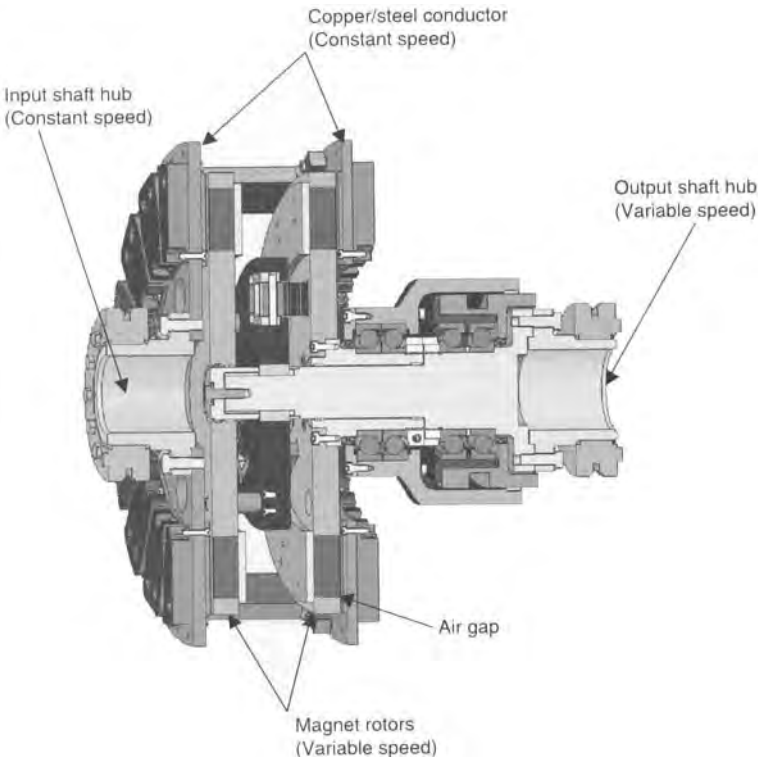


FIGURE 1 Cutaway of a shaft-mount air-cooled permanent magnet ASD (MagnaDrive Corp.)

rotor rotates relative to the magnet rotor, the magnetic flux from the magnet poles permeates the air gap and creates eddy currents in the copper conductor disks. The eddy currents create poles that interact with the magnet poles, whereby the magnetic flux develops a tangential force tending to turn the magnet rotor in the same direction as the rotating conductor. The net result is a torque available at the output shaft for driving a load. The magnet properties, rotor geometry, and materials of construction determine torque capability.

The magnet rotors move axially about a center pivot to increase or decrease the width of the air gap between the conductor rotors, thereby varying the amount of torque transferred and providing speed control. The center pivot is locked onto a center shaft where a cam driver and follower allow positioning of the magnet rotors through uniform motion. The drive mechanism is actuated through an external control arm that is supported about the center shaft by a pair of antifriction bearings. The center shaft is fastened to the output member and properly aligned by a machined register fit to the shaft coupling hub. Relative position between input/output rotors is maintained by the driver and load shafts, or when installed inside an enclosure, by a pair of rolling element bearings. Air gap, output torque, and load speed are usually not proportional, therefore, load torque characteristics and speed range must be known for proper permanent magnet adjustable speed drive size selection.

EDDY-CURRENT COUPLINGS

The eddy-current coupling is an electromechanical torque-transmitting device installed between a constant-speed prime mover and a load to obtain adjustable-speed operation. In most cases, the eddy-current coupling has an appearance very similar to that of its driving motor except that the coupling has two shaft extensions. One shaft, which operates at constant speed, is connected to the motor; the other shaft, providing adjustable speed output is connected to the load. A typical self-contained air-cooled eddy-current slip coupling is shown in Figure 2.

The input and output members are mechanically independent, with the output magnet member revolving freely within the input ring or drum member. An air gap separates the members and a pair of rolling element bearings maintains their proper relative position.

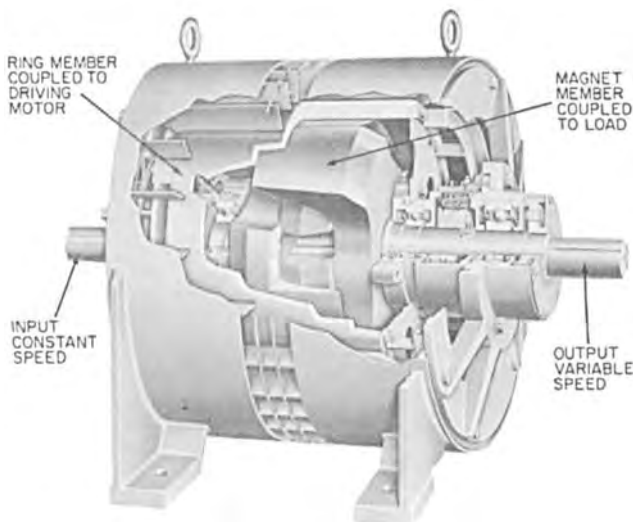


FIGURE 2 Cutaway of an ampli-speed eddy-current coupling (Electric Machinery Mfg.)

The magnet member has a field winding that is excited by direct current, usually from a static power supply. Application of this field current to the magnet induces eddy currents in the ring. The interaction between these currents and magnetic flux develops a tangential force tending to turn the magnet in the same direction as the rotating ring. The net result is a torque available at the output shaft for driving a load. An increase or decrease in field current will change the value of torque developed, thereby allowing adjustment of the load speed. Field current, output torque, and load speed are usually not proportional, therefore, load torque characteristics and speed range must be known for proper eddy-current coupling size selection.

Most eddy-current couplings have load speed control. An integral part of this system is a magnetic pickup that provides an indication of exact output speed and enables control of load speeds within relatively close tolerances regardless of reasonable variations in load torque requirements. Slip-type adjustable speed drives without load speed control may experience fluctuations in load speed when torque requirements vary.

CONFIGURATIONS

The permanent magnet ASD can be incorporated as an integral part of the drive system, mounted between parallel shafts or connected to right-angle gearing. They are available in horizontal (Figure 3) and vertical (Figure 4) configurations. The drive can be shaft mounted, same as a conventional coupling, using hubs secured directly to the motor and load shafts, or the drive may be mounted inside an enclosure, similar to an eddy-current drive, with two shaft extensions that are flexibly coupled to the driver and pump. When mounted inside an enclosure, the permanent magnet ASD has an appearance very similar to an eddy-current coupling (Figure 5). The most economical selection is a shaft-mount drive unit. Horizontal machines up to 400 horsepower are frequently shaft-mount air-cooled and larger power sizes are usually mounted inside an enclosure. Vertical motors are

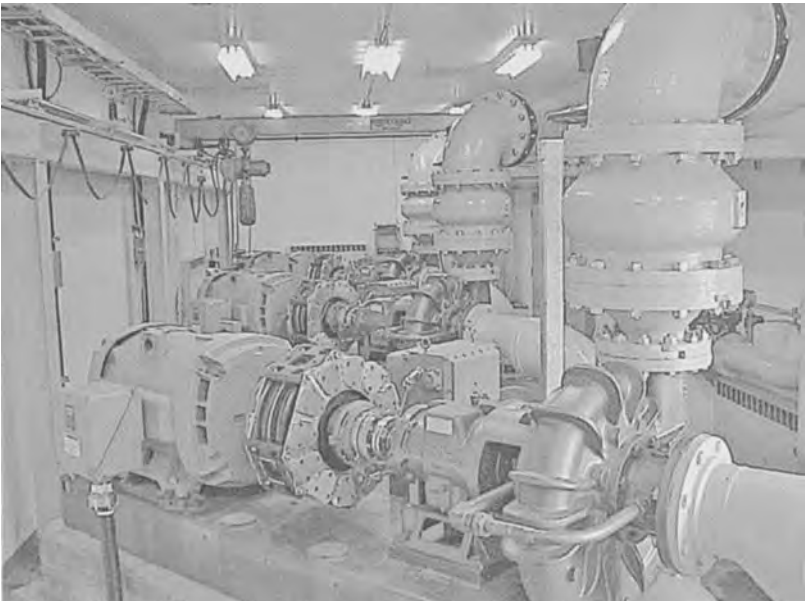


FIGURE 3 Horizontal centrifugal pump driven by 200-hp, 1780-rpm induction motor and permanent magnet ASD (MagnaDrive Corp.)



FIGURE 4 Vertical centrifugal pump driven by 350-hp, 1180-rpm induction motor and permanent magnet ASD (MagnaDrive Corp.)

close-coupled to limit overall height and to prevent vibration problems. Pump hydraulic thrust requirements can be accommodated with thrust bearings, typically located in the bottom of the permanent magnet ASD.

Enclosures (housings) will vary depending on the size of the permanent magnet ASD and on the type of cooling required. Air-cooled installations in clean or dirty atmospheres need open or drip-proof enclosures where heat rejection is not a problem. Enclosures are weather protected and suitable for outdoor installations. Both horizontal and vertical installations above 500 horsepower require water-cooled, sealed enclosures (Figure 6). No special considerations are necessary except to provide a clean, pressurized water supply and employ caution in outdoor installations where freezing can occur. Cooling water may be discharged on site, otherwise a coolant recirculation system is required.

PERFORMANCE FUNDAMENTALS

The permanent magnet adjustable speed drive, like eddy-current couplings, is a constant torque device that operates on the slip principle and is classified as a torque transmitter. This means that input and output torque are essentially equal and speed change is achieved by modifying the amount of torque transferred through the device. Power is a



FIGURE 5 Vertical centrifugal pumps driven by a permanent magnet ASD (front), an eddy-current coupling (center), and a direct coupled motor (back) (MagnaDrive Corp.)

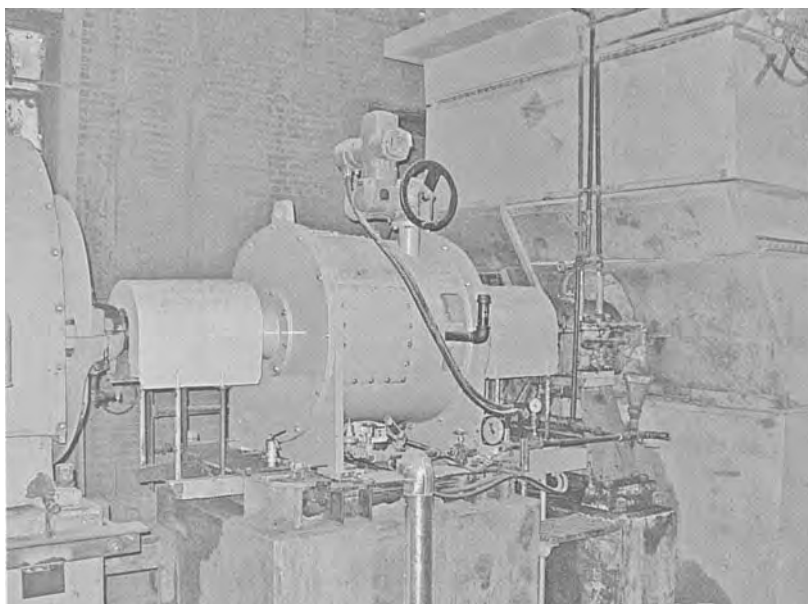


FIGURE 6 Horizontal induced draft fan driven by 700-hp, 1180-rpm induction motor and water-cooled ASD (MagnaDrive Corp.)

function of speed and torque. The fundamental formula for power is given below for both USCS and metric units.

$$\text{hp} = \frac{\text{rpm} \times T}{5252}$$

$$\text{kw} = \frac{\text{rpm} \times T}{9545}$$

where rpm = shaft speed

T = torque, ft · lb (N · m)

A permanent magnet adjustable speed drive must slip in order to transmit torque. Applying the above power equation separately to the input shaft and the output shaft shows that there is a difference in power. This difference in power is called slip loss, and is in the form of generated heat. There are also windage and frictional losses, but they are usually small enough to be neglected. Common values of slip for a centrifugal pump application are from 1 to 4%. The power formula can be used to determine slip horsepower.

$$\text{hp}_{\text{slip}} = \frac{(\text{rpm}_1 - \text{rpm}_2) \times T}{5252}$$

$$\text{kw}_{\text{slip}} = \frac{(\text{rpm}_1 - \text{rpm}_2) \times T}{9545}$$

where rpm₁ = motor speed

rpm₂ = load speed

T = load torque, ft · lb (N · m)

Permanent magnet adjustable speed drives and eddy-current couplings typically control centrifugal equipment such as pumps and fans. This type of equipment has a variable torque load that increases approximately as the square of the speed. Load power increases approximately as the speed cubed. This type of load increase is specifically for systems where the total system head is developed from frictional resistance (dynamic losses). Examples of these systems include pumping between reservoirs having the same levels or in a closed loop. Figure 7 shows the load power curve as a function of motor speed for such a system and illustrates the motor power curve and slip loss, calculated from the equations above. It can also be seen that efficiency is approximately equal to the percentage slip.

Many centrifugal equipment systems must first reach a static head level before flow begins. Figure 8 is an example of such a system. Load power curve OAB illustrates the closed discharge power. At point A the static head is overcome, and the pump load then rises along curve AC. Typically, even large static head systems will use the same permanent magnet ASD or eddy-current coupling appropriately sized for the ideal system shown in Figure 7. However, note that the peak slip loss value is reduced.

Performance is typically rated for continuous operation with ample torque for start-up and over-speed based on the horsepower and speed of the driver. The coupling's ability to transmit torque is a function of slip speed. Figure 9 shows power transmission as a function of load speed—note that high load speeds equate to low slip speeds. The initial part of the curve at full speed (low slip speeds) is often approximated as linear and defined by a single “torque per rpm slip” factor that can be used for estimating slip at the maximum speed position. The peak torque is also called the “breakdown” torque value, as further increases in slip speed will cause a reduction in transmitted torque (and power).

Slip-type couplings are best suited for controlling speed of centrifugal equipment, as discussed above. The power capacity of a permanent magnet ASD is determined by both the torque capacity and the amount of heat that can be removed from the conductor plates. All slip couplings and drives generate heat that must be dissipated by air-cooling or, for

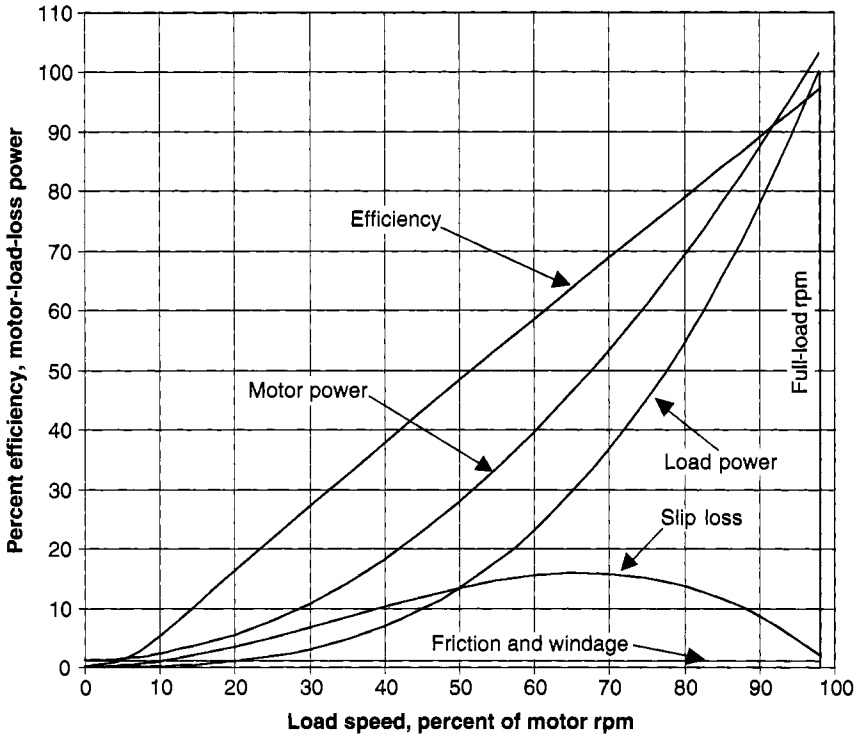


FIGURE 7 Load, power, and slip characteristics for a friction-only pumping system

high horsepower products, water-cooling. The heat is dissipated through the conductor plates (where eddy currents are developed) and must be removed to avoid overheating the magnets and/or causing thermal damage.

Two methods of heat dissipation are employed. Air-cooled units are primarily cooled by convection into the surrounding air, making their heat capacity a function of ambient air temperature and conductor rotating speed. At a given motor speed, each unit is rated for its heat capacity at 70°F ambient temperature, in horsepower. The coupling can dissipate its rated heat capacity in slip heat continuously. Water-cooled units rely on a system to supply coolant to the conductor plates. The water is pumped into an annulus that feeds water directly to the conductor to draw out the slip heat. A continuous supply of cool water must be available, or a circulating system with heat exchanger must be provided.

Slip loss can be calculated for each operating point from the torque required, using the slip hp equation. With a machine that has a cubic power relation to speed, the maximum heat generated (in power) is approximately 16% of rated power at full load speed, and is generated at 67% motor speed (Figure 7).

A permanent magnet adjustable speed drive controls speed by varying the air gap between the magnet and conductor rotors. Increasing the air gap will lower the torque transmitted by the coupling and reduce the load speed. A fully engaged coupling will operate at the intersection between the maximum power versus speed curve and the system curve as shown in Figure 9. As the air gap is increased (opened), the system will now operate at the intersection between the new power versus speed curve and the system curve. Minimum speed (turndown) is achieved when the air gap is fully opened as dictated by the

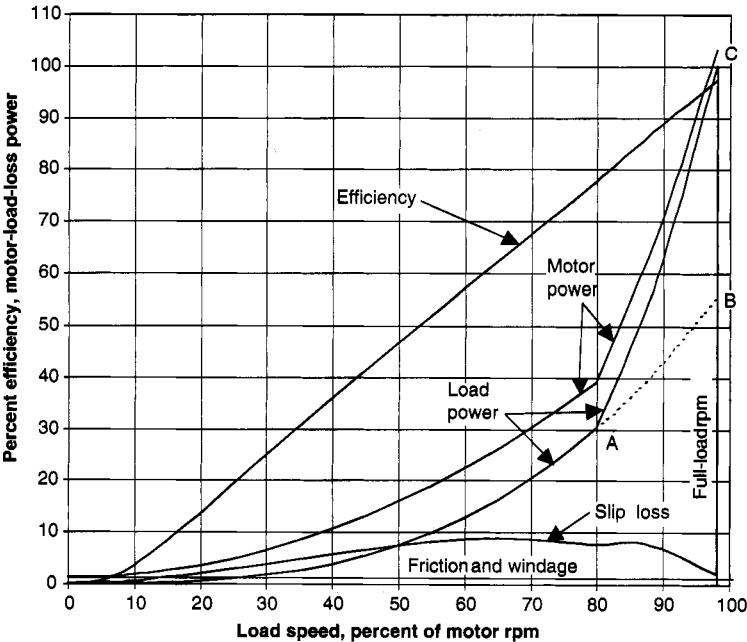


FIGURE 8 Load, power, and slip characteristics for a static head and friction pumping system

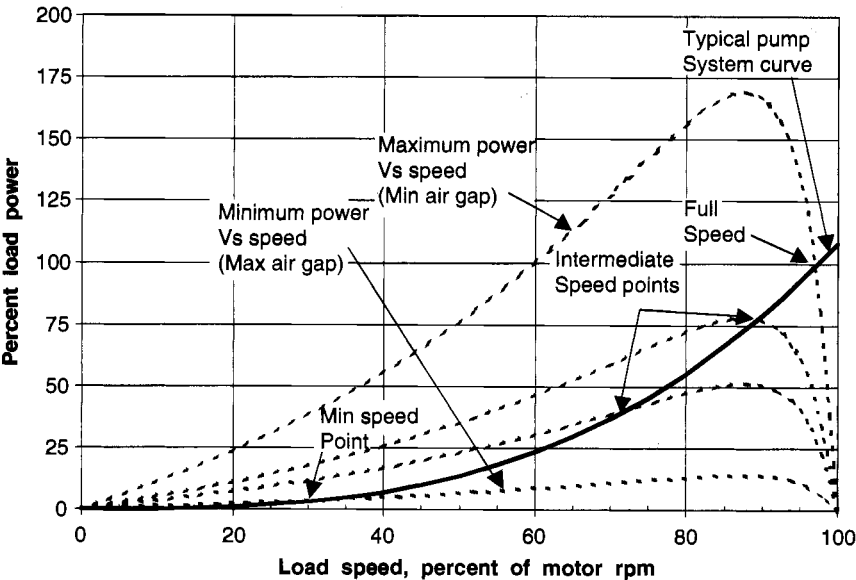


FIGURE 9 Typical load, power, and slip characteristics for a variable speed application showing intersections with the pump system curve

minimum power versus speed curve. Turndown speed ratios of 4:1 are typical and can adequately accommodate centrifugal pumps that will operate in their minimum flow region.

It is important for the system curve to have a steeper slope than the power versus speed curve. If the system curve is too flat or the power versus speed curve too steep, there could be multiple intersection points along the system curve. In certain speed ranges, the drive could provide more power than required by the system curve, causing the unit to accelerate through those ranges without providing control stability. Also note that constant torque applications must be carefully reviewed and may require special conductor materials to achieve desired control and turndown speed.

ENGINEERING CONSIDERATIONS

Permanent magnet adjustable speed drives are utilized to drive centrifugal pumps in many applications where variable flow and pressure is required. They provide vibration isolation between pump and driver because there is no mechanical connection between input and output members. For shaft-mount arrangements, this allows greater misalignment between equipment to accommodate thermal expansion, system fluctuations and pipe strain. When compared to flow control valves that restrict flow to desired levels, a permanent magnet ASD saves energy by reducing the rotational speed of the pump/load to match the variable process requirements. The motor operates near its synchronous speed, where maximum cooling efficiency is obtained, and output torque is lower so power consumption is reduced. Smooth transition between set points is critical to process control and by efficiently responding to any input signal through a mechanical drive mechanism, precise speed adjustment is attained. The permanent magnet ASD generates no electronic harmonic distortion and no voltage transients that are detrimental to the system and can disrupt electronic equipment. Likewise, they are not affected by lightning strikes or power fluctuations. They provide no-load or cushioned start/stop capability to the motor and pump, which reduces fatigue stress in the system, inrush current at start-up, and motor winding heat.

Drive systems should be given special consideration with regard to residual unbalance. In general, the major rotating assemblies of a permanent magnet ASD should be balanced to maintain the equipment within acceptable vibration levels. The type of balance and limits of imbalance vary with the component weight and radius, operating speed, and selected industry specifications. Generally, the rotating components are balanced to meet the requirements of ISO 1940-1 grade G6.3.

For an assembled drive unit, electromagnetic interference (EMI) is not a factor and is less than that typically found in an electric motor. However, for disassembled units, individual magnets and exposed magnet rotors require strict safety precautions and handling procedures—consult manufacturer for recommendations.

During maximum slip heat condition (~67% motor speed), conductor rotor surface temperatures can easily reach 250°F. Protective guards must provide sufficient airflow to allow adequate heat dissipation. Disconnected input and output shafts offer equipment protection. A failure or seizure of the equipment load will not result in shaft breakage or pump damage, but may result in unusually high slip speeds that can quickly overheat and damage the permanent magnet ASD. At elevated temperatures, magnets may become partially demagnetized and the conductor material may breakdown. In these instances, the drive unit should be instrumented with a noncontact optical or other type of temperature/speed sensor to provide alarm and shutdown circuitry.

Weight, length, and center of gravity should be considered for shaft-mount arrangements to evaluate overhung loads and estimate critical frequencies, especially with long, small diameter shafts. The foundation must be of adequate strength and stiffness to support the weight of the drive and its enclosure. Vertical configurations are much more flexible than most horizontal arrangements, making them more susceptible to resonance. The system critical (reed) frequency must always be reviewed to verify the pump discharge head has adequate stiffness to support the permanent magnet ASD and the motor without operating near the natural frequency of the installed equipment.

CONTROLS

The movement needed to change the magnet rotor position and decrease/increase the output shaft speed is produced by an actuator. The type of actuator motion necessary to control the adjustable speed drive is either linear or rotary. A typical arrangement results from linear translating motion provided by a quarter-turn actuator working directly through linkage connected to the drive mechanism control arm.

The choice of actuator depends on the service, usually modulating. In modulating service, the actuator must adjust to changing speed demands of the system. Actuator torque, stroke speed, its ability to hold a speed control setting and provide stability under fluctuating loads are important factors when selecting an actuator. Actuators may be manually powered (gear box), pneumatic/hydraulic powered, or, most commonly, electric motor driven.

A closed feedback control loop can be added to the system that receives an input signal (usually air pressure or electric signal) from a controller or process source and adjusts the drive's output speed to a prearranged corresponding value. Instrumented process signals may originate from load speed, flow or pressure sensors. Typically, the controller contains a signal isolator to allow precise speed adjustment from an automatic 4 to 20 mA or 0 to 10 VDC control signal. Position accuracy is held within 0.1% of the set speed throughout the operating range. Most actuators include limit switches and the drive mechanism contains mechanical stops, both of which protect the unit from overpowering and over engaging.

INSTALLATION

The basic permanent magnet adjustable speed drive is factory assembled and balanced. They can be installed and maintained by millwrights (no electrician necessary). Regular and preventative maintenance techniques for a permanent magnet ASD are very similar to other types of rotating equipment. Shaft-mount arrangements connect directly to the driver and load shafts using keyed or shrink-disc coupling hubs, similar to a conventional coupling. Enclosed arrangements mount on their own pedestal and require conventional couplings to connect to the driver and load shafts. For enclosed units, the foundation and pedestals must be of adequate strength and stiffness to support the weight of the drive, to maintain correct alignment between connected machinery, and to minimize the effects of vibration. During field mounting, the unit should be aligned following manufacturers recommendations and securely bolted in place.

The permanent magnet ASD can be installed indoor or outdoor without special provisions. The drive is a nonenergized mechanical device; therefore, under normal operating conditions, it may be applied in hazardous services, and used in corrosive or explosive environments. Only the actuator needs to be rated for hazardous locations in order to place the drive within an explosive environment. Other hazardous and corrosive environments should be evaluated for materials of construction, but versatility is not affected. Since the permanent magnet ASD does not require an external power source, it can be used where electrical power is not available. All that is necessary is a driving shaft and a load shaft. A manual gear actuator can be used to control speed.

LIMITED END FLOAT, THRUST CAPABILITY AND LUBRICATION

Permanent magnet adjustable speed drives have limited axial end float capability—a result of the air gap between rotors. Many horizontal motor-driven pump systems utilize motors equipped with sleeve journal bearings and no thrust bearing. For horizontal shaft-mount configurations, both pieces of machinery being connected must have a thrust bearing to limit the total axial end float no greater than 0.03 in. Alternatively, horizontal enclosed configurations are equipped with a thrust bearing eliminating the concern with machinery end float limitations.

Most vertical permanent magnet ASD are supplied with a thrust pot designed to handle the down-thrust and potential up-thrust generated by the pump/load and limit axial end float. A variety of external down-thrust load capabilities are available from the manufacturer. Bearing life is an important factor and most designs satisfy a minimum ABMA L_{10b} life of 40,000 h at an average output speed equivalent to the input speed. Bearings are angular-contact ball type or, where higher thrust values are encountered, spherical roller bearings may be used.

Bearing lubrication is generally accomplished either by grease or oil. Grease-lubricated bearings are the most common lubrication system for light to medium duty permanent magnet ASD. Oil lubrication is the common system for large horsepower water-cooled drives and vertical configurations with thrust pots. A typical system includes oil-lubricated bearing housing with a vented fill cap and an oil level sight glass to allow visual inspection of the oil level. Either too high or too low oil level can cause improper lubrication of the bearings, reducing their life.

APPLICATIONS

Permanent magnet adjustable speed drives are used extensively with centrifugal pump applications where the pump must deliver a wide range of flow over time. Examples of pumping systems which demand a range of flow include water and wastewater pumping in municipal installations, process pumping in a variable capacity process plant, boiler-feed and cooling-water pumping, circulating water and condensate pumping in power stations, stock pumping in paper mills, process water, reclamation and dewatering in mining and mineral processing, drainage, irrigation and flood control pumping, oil and gas processing, and HVAC chilled water and cooling water pumps in commercial and institutional buildings, and in oil and gas processing. Practically any pump application (or fan application), where cyclic constant-speed operation, fluctuations in process demand, valve throttling or other means of flow control are alternate considerations, can be conveniently solved with the use of a permanent magnet ASD as the variable-speed flow controlling device.

ENGINEERING SELECTION

Permanent magnet ASD are designed—and best suited—for driving equipment where power requirements vary exponentially with speed (rpm) -see Sections 2.1.1 and 2.1.3 for “similitude” and “affinity laws”. For any centrifugal pump application, the engineer can make an initial size selection based upon the motor speed and nominal horsepower (Table 1). The user's system will further define the selection process dependent upon several factors, including power, rotational speed, shaft separation, equipment orientation (vertical/horizontal, shaft-mount or enclosure), system influences, reliability, type of permanent magnet ASD required, and any special specification requirements. A system-wide assessment is important during the evaluation/selection process and the total cost of ownership model is a helpful tool when evaluating product, operating, maintenance, and downtime expenses.

The following denotes the basic information required by the permanent magnet ASD manufacturer for selection purposes.

- Driver type, horsepower, speed, and shaft end details
- Driven machine type, power required, variable speed range, shaft end details, and how torque changes with speed
- Site conditions, environment (indoor/outdoor), ambient temperature, and elevation
- Cooling medium available
- Control type
- Accessories
- Specifications

TABLE 1 Permanent magnet ASD selection chart, horsepower output

Numbers in Table represent typical ASD model sizes

Example: 75 hp at 1800 rpm = 14.5 ASD

Centrifugal Applications Only

Motor hp	Motor Speed (rpm)				
	3600	1800	1200	900	600
10	6.5	6.5	8.5	10.5	10.5
20	6.5	8.5	10.5	12.5	12.5
30	8.5	10.5	12.5	14.5	14.5
40	8.5	12.5	12.5	14.5	16.5
50	8.5	12.5	14.5	16.5	18.5
60	10.5	12.5	14.5	16.5	18.5
75	10.5	14.5	16.5	18.5	20.5
100	12.5	14.5	18.5	20.5	22.5
125	12.5	16.5	18.5	22.5	24.5
150	14.5	18.5	20.5	24.5	26.5
200	14.5	18.5	22.5	26.5	31
250	16.5	20.5	24.5	26.5	31
300	16.5	22.5	26.5	31	31
350	18.5	24.5	31	31	31
500	20.5	26.5	31	31	33
750	31HS	31	31	33	33
1000	31HS	31	33	33	33
1500	31HS	33	33	33	35
2000	31HS	33	33	35	35
2500	—	33	35	35	35
3000	—	33	35	35	37
4000	—	—	35	37	37
5000	—	—	35	37	—
6000	—	—	—	37	—

Note: 1 hp = 0.746 kW

FURTHER READINGABMA 9-1990, *Load Ratings and Fatigue Life for Ball Bearings*.ISO 1940-1, *Mechanical Vibration—Balance Quality Requirements of Rigid Rotors—Part 1: Specification and Verification of Balance Tolerances*.Volk, M. W. *Pump Characteristics and Applications*, Marcel Dekker, Inc., New York, 1996.

9.2.2 SINGLE-UNIT ADJUSTABLE-SPEED ELECTRIC DRIVES

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J. R. HENDERSHOT

In past years, the electric motor industry has been dominated by the use of essentially single and constant speed motors when line fed. There were many mechanical methods used to achieve adjustable speed using a single-speed prime mover. There was also widespread use of dc motors for adjustable speed over the last century because only the voltage must be adjusted for speed change. Although operation at speeds well below synchronous can be accomplished through adjustment of the secondary resistance in wound-rotor induction motors, the ubiquitous squirrel-cage ac induction motor requires the adjustment of the frequency to change speed. This was not so easily accomplished until the recent utilization of pulse width modulation (PWM) inverters with insulated gate bipolar transistors (IGBTs) for power transistors plus microprocessors.

Electric drives are available featuring a single rotating element and some associated control for performing the adjustable-speed driving function. The use of a single rotating element differentiates this type of drive from the eddy-current coupling motor, hydraulic coupling motor combinations, and adjustable-speed belt drives, all of which are tandem drives.

The speed of the single-unit adjustable-speed drive is controlled through the interaction of the control and the motor. For this reason, one must consider these two elements as a drive and not consider the motor alone.

Section 9.1.1 described motors that will be further discussed in this section. Previous discussions of control were limited to that required for starting and protection; this section will also describe specific controls required for speed control. The discussion will cover the following types of drives:

- Ac adjustable-voltage drives
- Wound-rotor induction motors with several different types of secondary controls
- Adjustable-frequency drives

- Modified Kraemer drives
- Dc motors with silicon-controlled rectifier (SCR) power supplies

All have physical features, operating characteristics, or prices that make them particularly valuable in some specific segment of the pump driver spectrum.

ALTERNATING-CURRENT ADJUSTABLE-VOLTAGE DRIVES

This drive consists primarily of an adjustable voltage, constant frequency control, and a motor (M) conveniently configured as shown in Figure 1. The motor must possess high slip characteristics and other characteristics that allow it to work successfully with the associated control. The motor is designed for operation and tested with its associated control. Because the motor has high slip characteristics, insulation is of Class F rating.

The fundamental parts of the control are a circuit protective device, such as a circuit breaker, an SCR assembly, and firing circuitry, identified as FC in Figure 1. Other parts are required to complete the control but serve as auxiliaries. The main function of this control is to provide a voltage to the motor at a level that will ensure a desired motor speed. The control also protects the motor under abnormal operating conditions and the motor cable under short-circuit conditions and provides a current-limit function so the motor draws a maximum of some preselected value, such as 150% of normal, under all operating conditions.

The SCR receives impressed voltage, usually at 60 Hz. The SCR can be turned on (or become conducting) by means of current pulses received from the firing circuits that energize the circuits at different points on the sinusoidal wave. After conductance starts, it continues until voltage disappears at the end of the half-cycle. The SCR must be pulsed again in order to become conducting. This occurs while the voltage is in the negative phase of its sinusoidal generation and is performed by an SCR connected in parallel to the first and having reverse polarity. The negative loop conductance continues until the voltage again returns to zero. The firing controls are designed to turn the SCR on repetitively sometime during each voltage half-cycle. Figure 1A shows the shape of the applied voltage between the SCR and the motor for the wave illustration shown. In this case, the SCR is turned on at the beginning of each half-cycle.

By delaying the firing, or turning on, of the SCR until later in voltage generation, shorter intervals of applied voltage and lower levels of voltage appear across the motor. The solid line of Figure 1B illustrates applied motor voltage when the voltage half-wave is half-completed. Figure 1C illustrates the motor-applied voltage when the half-waves are approximately 75% completed. Notice in these illustrations that the frequency of the voltage applied to the motor does not vary; only the voltage magnitude does.

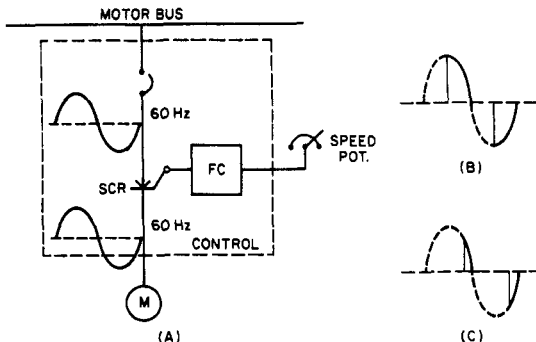


FIGURE 1A through C Block diagram for ac adjustable-voltage drive (General Electric)

The exact point in the half-wave when firing occurs is controlled by a low-energy electronic signal that may come from a potentiometer, as in Figure 1A, or from some process instrument signal. By increasing the signal level, voltage applied to the motor increases; on the other hand, a decrease in signal level decreases the voltage level.

Figure 1A shows the SCR on the utility side of the motor as a convenience in illustration. In reality, the SCR is generally placed at the motor neutral point. This reduces voltage from the SCR to ground and allows the motor impedance to protect the SCR somewhat from damage caused by transient voltage spikes entering from the electric utility line. However, placing the SCR at the motor neutral point does require the use of six motor conductors instead of the conventional three.

Figure 2 illustrates representative motor speed-torque curves at rated and other voltages as well as a pump speed-torque curve varying as the square of the speed. The motor possesses approximately 10% slip under rated torque and voltage conditions. Note that motor torque does not break down at speeds lower than breakdown torque speed. These various characteristics help to identify the motor as one designed to operate specifically for this application. Note that by reducing applied motor voltage, motor torque decreases, causing the pump to operate at lower speed.

Ratings of these drives are limited essentially to low power levels. Figure 3 provides a guide to those available in open construction only. All ratings with maximum rated speeds existing within the solid envelope are available. Some, but very few, are available outside of this envelope. Totally enclosed motors are more restricted in supply than open motors and are limited to approximately 40 hp (30 kW) maximum. In addition to these restrictions, load torques must not exceed values varying as the square of the speed; thus this drive is not a candidate for driving a constant-torque load. Table 1 gives some pertinent application information.

WOUND-ROTOR INDUCTION MOTORS

Liquid Rheostat Controls This form of drive consists primarily of a full-voltage, non-reversing (FVNR) starter, a wound-rotor induction motor, and a liquid rheostat, all of which integrate into a configuration as illustrated in Figure 4. The FVNR starter switches power to and from the motor stator as well as provides generally accepted protection to

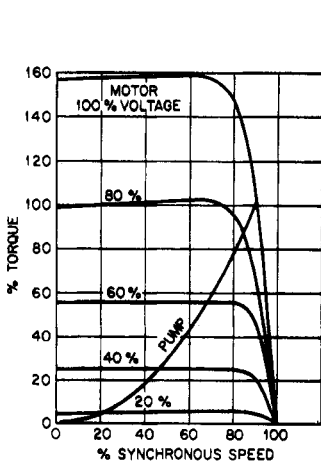


FIGURE 2 Torque versus speed for ac, adjustable-voltage drive (General Electric)

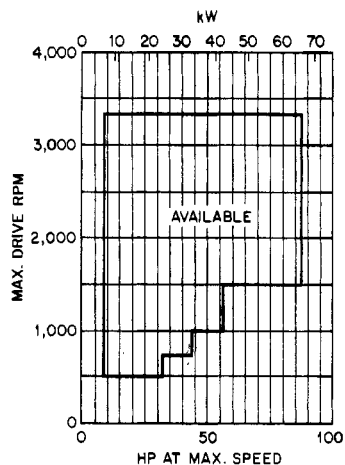


FIGURE 3 Availability of ac adjustable-voltage drives

TABLE 1 Alternating-current adjustable-voltage drive data

Drive element	Power rating ^a	Voltage rating	Max rated speed, rpm	Speed range, %	Enclosure	Mounting	No. of speed points
Motor	See Figure 3 for open frame ratings; limited availability outside these limits	200	1640	100	Vertical: shielded drip-proof; TEFC not generally available	Vert. or horiz.	Infinite
		230	1095				
		460	820				
			655				
			545				
			& others				
	Totally enclosed ratings limited as shown under Enclosure				Horizontal: drip-proof, TEFC to 40 hp (30 kW) approx.		
Control	5–150 hp (3.7–112 kW)	200	...	...	NEMA 1	Wall or floor	Infinite
		230			NEMA 12		
		460					

^aDrive is capable of operating with power varying as the cube of speed maximum. Specifically, these ratings are not suitable for constant-torque applications.

the motor from short circuits, overloads, and so on. Secondary cables connect the motor rotor and fixed electrodes in the liquid rheostat. The fixed electrodes exist in separate cells of the rheostat, one for each phase. Movable electrodes, one located in each cell, are suspended from a horizontal bar, and this bar, the vertical-electrode suspension bars, the movable electrodes, and the electrolyte filling the cells complete the Y, or common point in the external motor rotor circuit. When the upper electrodes are moved up or down, the secondary circuit resistance of the drive varies and this causes a change in the motor speed-torque characteristic.

As already pointed out, the motor starter provides the normal protective functions for the motor. Control circuitry allows the motor to be started with maximum secondary resistance in the rotor circuit, thus drawing minimum motor current from the power supply.

Figure 4 illustrates a motor (M3) operating a pulley that changes the position of the movable electrode. This motor could be controlled manually by a push button or automatically through a controller position providing either a raise-lower or modulated voltage signal. If desired, a pneumatic cylinder may substitute for motor M3 to provide power to move the electrodes.

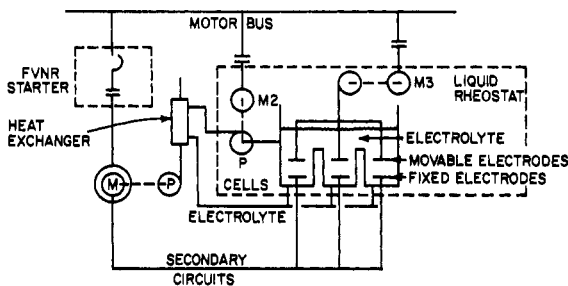


FIGURE 4 Block diagram for wound-rotor induction motor with liquid rheostat secondary power (General Electric)

The electrolyte receives the drive slip losses dissipated in each of the cells, and of course this heat must be removed into some heat sink capable of dissipating it. In some installations, an electrolyte pump driven by motor M2 circulates the electrolyte through a heat exchanger before returning it to the individual cells. Returning cooled electrolyte re-enters the cells under the fixed electrodes and passes through holes in the electrodes before passing vertically through the cells. The heat exchanger primary coolant can be tap water or mill water, which provides a good means of passing drive slip losses as heat directly out of the station, or it can be the pump discharge water, which conveys heat directly away from the building, as illustrated in Figure 4.

The liquid rheostat is factory-assembled with the rheostat, electrolyte pump, and its motor and electrode-positioning assembly packaged in a single steel enclosure suitable for control lineups except for ratings above 700 hp (522 kW). The heat exchanger may be of the sleeve (or wrap-around) type, as illustrated in Figure 4, and comes as a separate device to be attached to station effluent piping. Other forms of exchangers, such as liquid-to-air or shell- and-tube heat exchangers, are available.

Changing the position of the movable electrode results in a change in motor speed-torque relations, as shown in Figure 5. Each curve shown (except the pump curve) represents a motor characteristic for a discrete secondary resistance. Notice the number associated with each curve; it represents the percentage of secondary resistance external to the motor rotor; 100% ohms provide 100% motor torque at zero speed. In examining individual curves, note that as the percent resistance increases, the slope of the motor speed-torque curve decreases, thus allowing the motor to slide down the pump speed-torque curve and assume a lower speed.

This drive utilizes resistance only in the motor secondary circuit as a means of controlling speed. Reactance is available as a substitute for resistance in other forms of secondary controllers, but its use reduces the drive power factor and thereby increases motor current, reduces efficiency, and increases motor heating. As a further effect, the additional motor current may require the use of a larger motor frame to accommodate it.

Figures 6A, B, and D provide clues to the availability of enclosed vertical and horizontal wound-rotor induction motors. Figure 6A illustrates the point that vertical, totally enclosed wound-rotor induction motors are not generally available but may be available in random powers and speeds. Totally enclosed water-to-air-cooled (TEWAC) construction becomes available at some minimum point as illustrated. In large ratings, shown on the right, TEWAC construction becomes very important as it provides a convenient way of capturing motor losses and expelling them in cooling water external to the building. Both curves of Figure 6A terminate at 1800 rpm synchronous speed because higher speed ratings are generally unavailable.

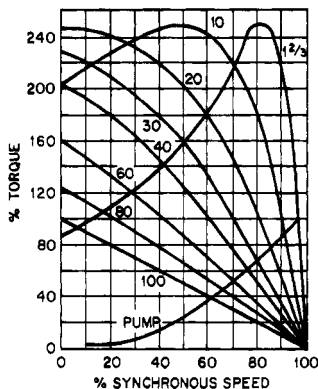


FIGURE 5 Torque versus speed for wound-rotor induction motor with variable secondary resistance (General Electric)

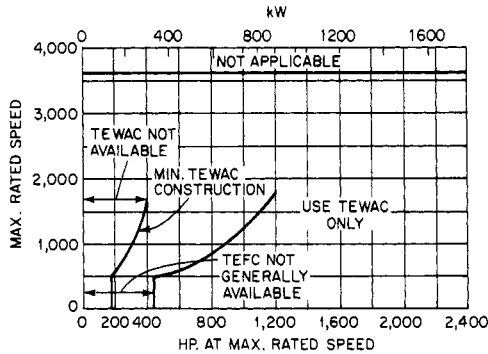


FIGURE 6A Approximate availability of totally enclosed vertical wound-rotor induction motors (General Electric)

Figure 6B delineates the left side, where totally enclosed fan-cooled (TEFC) construction should be used for horizontal wound-rotor induction motors because it is the only one available, the middle area, where TEFC construction should be used because it is less expensive than TEWAC construction, and, finally, the right side, where TEWAC construction is less expensive than TEFC construction. Again, motor speeds are limited to 1800 rpm synchronous.

Vertical wound-rotor induction motors are not available in explosion-proof construction, as their omission from Figure 6C implies.

Finally, the areas of availability and unavailability are delineated in Figure 6D for horizontal wound-rotor induction motors in explosion-proof construction. Again, the curve terminates at 1800 rpm synchronous speed for reasons already expressed.

Delineation of motor availability is an important criterion in drive selections. Table 2 presents significant data useful in drive selection.

Tirastat II* Secondary Controls This drive utilizes the same wound-rotor induction motor and FVNR starter as the preceding, but substitutes a Tirastat II controller for the

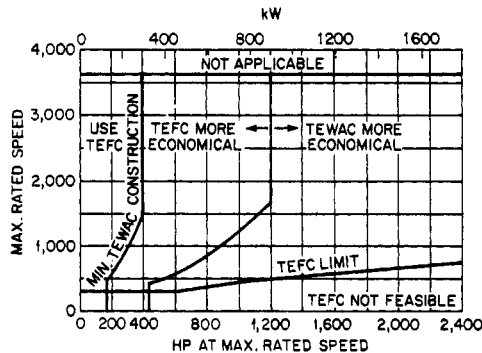


FIGURE 6B Approximate availability of totally enclosed horizontal and vertical squirrel-cage induction motors and horizontal wound-rotor induction motors. Wound-rotor induction motor speeds limited to 1800 rpm maximum (General Electric)

*Trademark of General Electric Co.

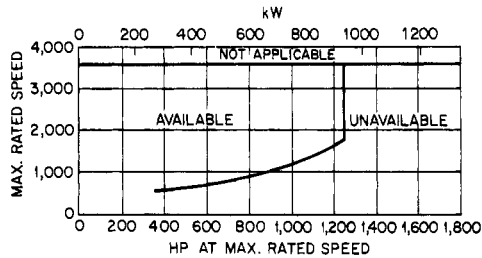


FIGURE 6C Approximate availability of explosion-proof vertical squirrel-cage induction motors (General Electric)

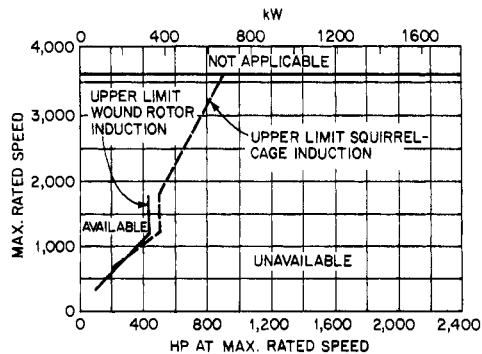


FIGURE 6D Approximate availability of explosion-proof horizontal squirrel-cage and wound-rotor induction motors (General Electric)

TABLE 2 Wound-rotor induction motor with liquid rheostat drive data

Drive element	Power rating ^a	Voltage rating	Max rated speed, rpm	Speed range, %	Enclosure	Mounting	No. of speed points
Motor	No limits for open motors; see Figure 6A, B, D for restrictions by enclosure	Any NEMA standard	All established by number of motor poles and supply frequency	60	Vertical: shielded dripproof, TEFC generally not TEWAC available per Figure 6A Horizontal: dripproof, TEFC, explosion-proof per Figure 6D	Vert. and horiz.	Infinite
Control	25 hp (18.7 kW) min; no stated max	For any motor secondary voltage		60	NEMA 1	Lineup or singly Floor only	Infinite

^aDrive may be designed for constant torque or for torque varying at the square of the speed.

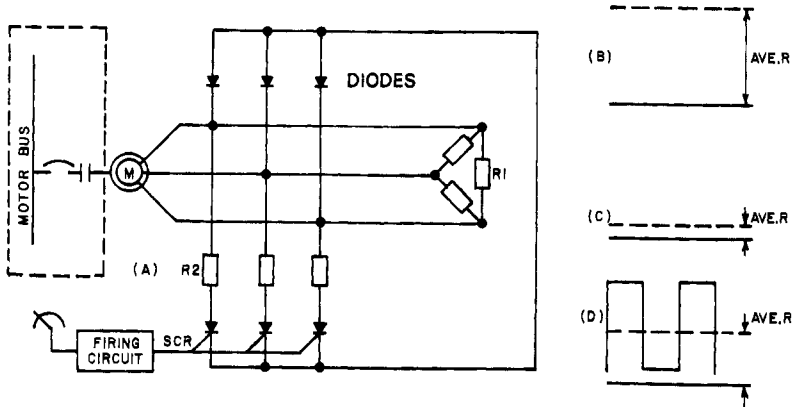


FIGURE 7 Block diagram for wound-rotor induction motor with Tirastat II secondary control power (General Electric)

liquid rheostat. Figure 7A shows the configuration with the wound-rotor induction motor and starter identical to those of Figure 4.

The Tirastat controller in its simplest form consists of the components shown in Figure 7A. Resistors identified as R1 are permanently connected across the motor secondary phases. Additional resistors (R2) are connected across motor phases through SCR and diodes. The SCR are individually turned on (become conducting) by minute current pulses generated by the firing circuit. As the voltage loop across any SCR drops to zero, the SCR shuts off and does not turn on again until the firing circuit refires it. The diodes allow return of the current to the motor but block out current reversals to the SCR.

Drive speed varies as a function of changing controller average resistance. Thus, resistance can be varied by adjusting time on-time off ratios in the SCR to give the desired average resistance. Figure 7B shows controller secondary resistance with the SCR not firing (conducting). In this condition, the controller resistance value equals the resistance of R1. Figure 7C shows controller resistance with all SCR firing continuously. The decrease in resistance is due to additional resistance placed in parallel with resistor R1. Figure 7D shows controller resistance varying from maximum to minimum in approximately equal time periods. The average resistance value then approximates half of the sum of R1 and R2. The time on-time off of the SCR can be varied automatically by the controller to provide the average resistance in the motor circuit necessary to give the desired motor speed.

The motor starter provides normal motor protective functions as well as short-circuit protection of the motor cables and starter. The Tirastat II controller monitors motor current and limits it to some preselected value, such as 150% of normal, under all conditions.

Packaging of the secondary controller is of particular interest. Both packaging and configuration lend themselves to mounting the resistors on or near the control enclosure or removing them some distance from the enclosure. By placing the resistors outdoors or at some indoor location away from the operations, heat losses can be released to the environment with impunity to operators.

The speed-torque ability of this drive would be very close to that illustrated by Figure 5. The comments contained in the text describing that figure apply equally well here. Table 3 presents significant data useful in drive selection.

Contact Secondary Controls Many pump drives utilize a very simple combination of FVNR starter, wound-rotor induction motor, and a form of contact making secondary control. Figure 8 shows the configuration with a motor and FVNR starter identical with those of Figures 4 and 7A.

The main difference between this drive and the two described previously lies in the construction of the secondary controller and the characteristics of the drive. Resistor R in

TABLE 3 Wound-rotor induction motor with Tirastat II controller drive data

Drive element	Power rating ^a	Voltage rating	Max rated speed, rpm	Speed range, %	Enclosure	Mounting	No. of speed points
Motor	No limits for open motors; see Figure 6A, B, D for restrictions by enclosure	Any NEMA standard	All established by number of motor poles and supply frequency	60	Vertical; shielded dripproof, TEFC generally not available, TEWAC available per Figure 6A. Horizontal dripproof, TEFC, explosion-proof per Figure 6D.	Vert. and horiz.	Infinite
Control	5 hp (3.7 kW) min. 600 hp (448 kW) max.	For any motor secondary voltage	...	50 70	NEMA 1, NEMA 12	Lineup or singly	Infinite

^aDrive may be designed for constant torque or for torque varying as the square of the speed.

Figure 8 is a three-phase resistor connected across the slip rings of the motor. Its resistance rating is selected to provide minimum motor torque at standstill or adequate torque at minimum speed. Contacts in the form of magnetic contactors, drum, cam, or dial switches are closed by a logic device (generally magnetic) to short-circuit part or all of the resistor. Thus, instead of a modulated or infinitely variable resistor as encountered with the other two drives, this drive modifies secondary resistance in discrete steps, providing discrete speed-torque curves instead of an infinite family. The number of speed-torque curves the control generates will depend directly on the number of contacts provided in the secondary-control circuit.

Either a manual switch or some automatic contact-making device actuated by the process can be utilized to actuate the control logic.

Secondary controls are normally packaged in the factory with resistors and contact-making secondary controller integrally assembled.

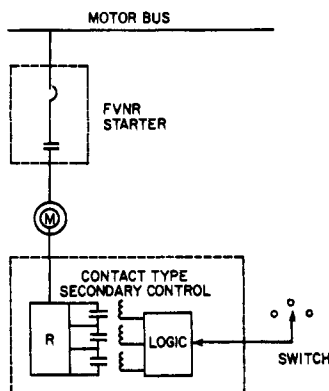


FIGURE 8 Block diagram for wound-rotor induction motor with magnetic secondary control power (General Electric)

TABLE 4 Wound-rotor induction motor with contact-type secondary control drive data

Drive element	Power rating ^a	Voltage rating	Max rated speed, rpm	Speed range, %	Enclosure	Mounting	No. of speed points
Motor	No limits for open motors; see Figure 6A, B, D for restrictions by enclosure	Any NEMA standard	All established by number of motor poles and supply frequency	60	Vertical; shielded dripproof, TEFC generally not available, TEWAC available per Figure 6A. Horizontal dripproof, TEFC, explosion-proof per Figure 6D.	Vert. and horiz.	
Control	No limits	For any motor secondary voltage	...	75	NEMA 1	Lineup or singly	Discrete only

^aDrive may be designed for constant torque or for torque varying as the square of the speed.

The speed-torque capability of this drive would be very close to that illustrated in Figure 5, with the exact number of curves determined by the number of contacts of the contact-making secondary controller and the shape of the curves determined by the resistance selected. Table 4 presents significant data useful in drive selection.

ADJUSTABLE-FREQUENCY DRIVES

Until recently, the most significant step in the improvement of adjustable drives was to replace the autotransformer and rectified variable voltage power supply with SCR drives. Although dc motors are seldom used for pump drives, except in automotive applications and other special cases, the SCR and PWM adjustable speed dc drives are very highly developed and available from a number of suppliers.

Pulse Width Modulation Frequency Inverters Adjustable-frequency drives consist of an adjustable-frequency inverter control and a constant-speed motor, as shown in Figure 9. The inverter control has generally been built in either pulse width modulation or square wave construction. An inverter control consists fundamentally of a circuit-protective device, a diode bridge, and an inverter section having an SCR and a firing-control (FC) section for the SCR, all packaged in a steel enclosure. Special power supplies, motor protective devices, and electrical protective devices complete the package.

Only low voltages are used to energize the motor bus. A diode bridge rectifies voltage and an inverter inverts the direct current into an adjustable-frequency adjustable voltage that varies linearly with frequency. This voltage is not necessarily sinusoidal; it can also consist of a number of dc pulses of positive or negative polarity, as shown between the inverter and the motor in Figure 9. The firing controls modulate the width of the pulses and the number of pulses per half cycle to vary the apparent frequency and to maintain motor voltage at a constant volts-per-cycle. The firing controls automatically introduce additional pulses or withdraw pulses as bandwidths reach their limits.

Firing circuits composed of solid-state devices trigger the SCR in accordance with logic controlled by the setting adjustment of a speed potentiometer (Figure 9) or the signal from the process. The control provides normal protection of the motor as well as short-circuit protection of the control, motor, and cables. Also, the control monitors and limits current drawn by the motor under all conditions to a preadjusted value, such as 150% of normal.

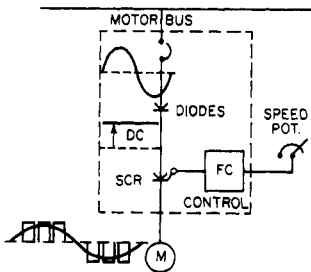


FIGURE 9 Block diagram for ac adjustable-frequency drive (General Electric)

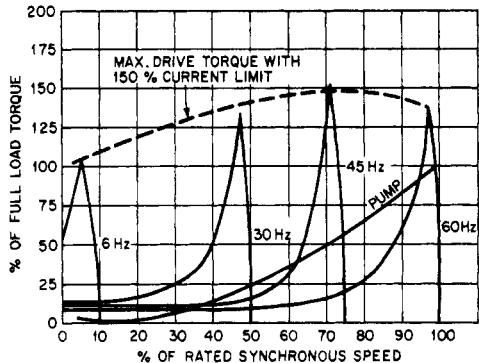


FIGURE 10 Torque versus synchronous speed for adjustable-frequency drive (General Electric)

Figure 10 displays some representative speed-torque characteristics of the drive utilizing a squirrel-cage induction motor and a representative centrifugal pump. Note that each characteristic intersects the zero-torque point at a different speed value. This differs from those of other drives, illustrated in Figures 2 and 5. The steepness of the slope of each characteristic as it rises from its zero-torque value indicates its low-speed regulation and provides a clue as to its ability to maintain speed with little fluctuation as load torque varies slightly. Motor speed is a function of frequency adjustment; the voltage is adjusted only to accommodate moderate changes in motor impedance. Table 5 lists significant application data.

Solid State Adjustable Frequency Inverters The opportunity for adjustable speed for ac induction motors has been realized by the development of the adjustable frequency control as well as the more recent vector drives. Similar adjustable speed drives are now

TABLE 5 Adjustable-frequency drive data

Drive element	Power rating ^a	Voltage rating	Max rated speed, rpm	Speed range, %	Enclosure	Mounting	No. of speed points
Motor	All NEMA ratings for open motors; see Figure 6B, C, D for restrictions by enclosures	460 or 230	All established by number of motor poles and supply frequency	100 plus limited over-speed	Vertical shielded dripproof, TEFC per Figure 6B, explosion-proof per Figure 6C. Horizontal: dripproof, TEFC per Figure 6B explosion-proof per Figure 6D.	Vert. or horiz.	Infinite
Control	Up through 800 hp (597 kW); higher ratings available as custom-built units	460 or 230	Capable of producing 150% of rated frequency but at reduced torque	150	NEMA 1 or NEMA 12	Floor	Infinite

^aDrive may be designed for constant torque or for torque varying as the square of the speed.

available for the control of switched reluctance (SR) and permanent magnet (PM) brushless motors. These three technologies are well suited for pump applications that can benefit from adjustable speed.

There are many manufacturers who offer variable frequency with constant or variable voltage IGBT-driven PWM inverters for reliable and performance programmable adjustable speed control of ac induction motors. These drives are available in power up to the hundreds of horsepower (kilowatts). They offer many standard features that can be useful for driving pumps. Some of these are summarized as follows:

- Wide input voltage range allowing standardization of motors.
- Dual frequency operation (50 or 60 Hz)
- PWM output for sine wave voltage at selected output frequency
- Constant or variable torque
- High starting torque at programmable linear acceleration/deceleration
- Over-current protection
- Phase to phase and phase to ground short circuit protection
- Maximum output frequency of 120 Hz for double motor speed
- External contacts for other uses
- Analog speed output proportional to frequency

The use of these types of adjustable speed drives provides pump speed ranges up to nearly twice the base speed of the ac motor. The details of an actual selection of a system should be discussed with an application from a supplier in order to select the right product and size for the pump. An example of a low-power ac inverter is shown in Figure 11.

These classes of solid state adjustable ac motor drives take the 50 or 60 Hz ac line voltage from the grid (wide voltage tolerance range OK) and first rectify it into a dc voltage. Some filter capacitors are sometimes used to attain a smooth dc voltage. Then the dc is converted into sine waves, one for each phase, at a frequency that is selectable to achieve the speed of the motor and pump according to the number of poles in the motor. The drive



FIGURE 11 Low-power AC inverter (motor controller) for three phase motors with sine wave outputs and an output frequency range from 1 to 120 Hz (courtesy of Leeson Electric Corp.)

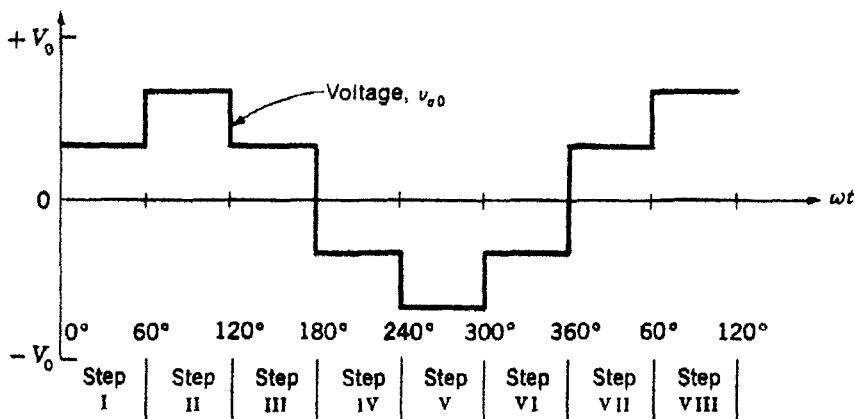


FIGURE 12 AC inverter (6) step simulated sine wave shown for one phase

is designed to maintain a constant volts/Hz relationship. Speed can be controlled to about 10% by inputting the frequency setting. The sine waves can be approximated, as shown in Figure 12, by the use of a simple six-step transistor switching method. Alternatively, they can be very precisely generated using a PWM switching topology at a high carrier frequency to achieve a very low-distortion sine wave. The dynamic response of these systems is sufficient for many pump applications. However, if there are sudden load changes and speed must be accurately maintained, such as for metering pumps, a higher dynamic response will probably be needed. For applications that require high acceleration, the constant volts/Hz algorithm will not allow a fast enough dynamic response because the system bandwidth will likely be too low.

Another problem with these drives is that they usually need a voltage boost at low speeds because of the low frequency. This would be a problem only for metering pumps run over a wide speed range, which is very rare.

Flux Vector Drives The highest performance class of solid-state adjustable speed drives for ac induction motors are known as *flux vector drives*. The rectification requirement of the incoming ac power is the same as is the regeneration of sinusoidal currents for each of the three phases to power the motor. The very important difference between them is that the phases are controlled in a closed loop fashion. The control block of the flux vector drive must receive rotor pole angular position feedback information. With this information, the inverter driven ac induction motor can be made to operate like a servomotor. Most flux vector drives utilize a digital signal processor (DSP) for all of the control functions using the software commands programmed by the supplier (user or supplier can modify). These include the various control loops such as speed (within 0.5% if required), acceleration (linear or to a function), current, torque, power, and several more over a very wide speed range. Drives of this type are available for high power with output frequencies of 800 Hz from several suppliers and up to 3 kHz from a few suppliers.

Flux vector drives can achieve a dynamic response in the 10-millisecond range and very smooth speed regulation, down to zero with ordinary induction motors. The concept of the vector control is to observe the present position of the rotor poles and formulate the control to achieve a dc servomotor performance. The control scheme synthesizes the two currents in the motor. If the calculation is done correctly, one of the synthesized currents controls the flux in the motor and the other controls the torque. In other words, one current vector in the stator phases lies along the vector of the rotor flux and the other current vector lies in quadrature or at 90° out of phase. The DSP constantly receives rotor pole position information and constantly recalculates this relationship at all speeds in spite of load changes.

An important feature of flux vector drives is that with simple and minor software modifications they can easily be used to drive PM brushless motors. The drive has transistor gate drivers controlled by the commands from the DSP that receives shaft angle position information from a shaft sensor such as an encoder, resolver, or hall sensors. The poles on the rotor do not move with respect to the shaft and rotor current (true synchronous operation without "slip") as do the poles in the induction motor. It is actually much easier to use this vector flux vector inverter for a PM brushless than for an ac induction motor. This is because there are seldom any calculations required to run the brushless motor unless phase advance is needed. However, for pump drives, this requirement seems unlikely.

MODIFIED KRAEMER DRIVES

Kraemer drives have been used as heavy industrial drives for many years. Although functioning very successfully, each has three rotating units requiring maintenance, and each has significant losses.

In recent years, the advance of semiconductors has simplified the drive configuration to that of Figure 13, which involves an FVNR starter, a wound-rotor induction motor, a solid-state converter, and an acceleration section. The acceleration section and contactors 1 and 2 can be omitted if the converter rating is large. The FVNR starter switches power to the motor and protects both the motor and the converter. Contactor C2, if provided, serves as a synchronizing contactor between converter and power line and is closed only when converter and line frequencies and voltages are compatible. This contactor may be placed on either side of the converter unit.

Under running conditions, rotor slip energy of low voltage and frequency (Figure 13B) flows to the low-frequency side of the converter. The converter unit inverts this voltage to a fixed line frequency. Thus all motor rotor slip energy except converter unit losses are returned to the power source, thereby improving drive efficiency.

Under starting conditions, acceleration occurs with contactor C2 open and C1 closed. Accelerating contactors progressively and automatically short-circuit the resistor R to allow the motor to accelerate to some preselected speed. When converter output voltage and frequency match line values, contactor C2 closes and C1 opens automatically. The drive then operates as already described.

The converter generally consists of a diode bridge and SCRs with their firing circuitry for inverting. Required auxiliaries, such as special power supplies, complete the package.

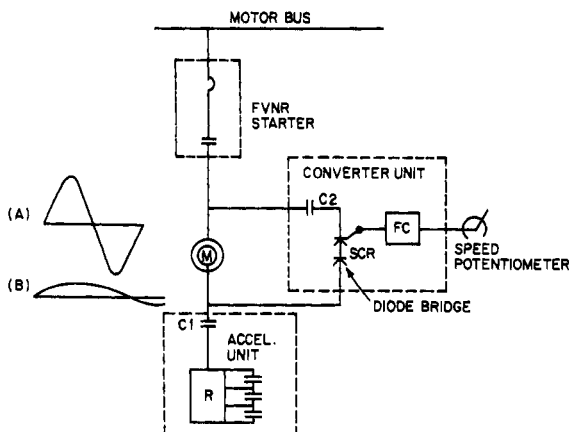


FIGURE 13 Block diagram for modified Kraemer drive

Generally all control materials are packaged in a single lineup to facilitate building and installation.

The motor starter provides normal protection of the motor as well as short-circuit protection of control, motor, cables, and converter. In addition, the converter monitors and limits current drawn from the line under all conditions to a preadjusted value, such as 150% of normal.

DIRECT-CURRENT MOTOR WITH SCR POWER-SUPPLY DRIVES

General industries use many dc motors with SCR power supplies. These drives perform very well in the exacting circumstances they normally face. Some of these drives are used to drive pumps. They are configured as shown in Figure 14 and consist of a dc shunt wound motor and an SCR power supply.

The dc power supply unit rectifies motor bus voltage to an adjustable dc voltage level, which in turn energizes the armature of the motor. The second function of the power supply unit is to rectify motor bus voltage and apply this constant dc voltage to the motor shunt field. By adjusting the dc voltage to the motor armature, the speed of the motor can be adjusted to a desired value.

The SCR power supply unit consists primarily of a short-circuit protective device, a switching contactor, SCR power modules, firing circuitry, a speed regulator, and a shunt field rectifier. The firing circuitry responds to a low-energy-level speed potentiometer or some process variable. The SCRs in turn respond to the firing circuitry and translate those signals into the correct dc voltage to be applied to the armature of the motor. Control circuitry is included to monitor motor current and limit it to a preselected value, such as 150% of normal.

The SCR power supply unit comes packaged for easy installation in the field.

Speed-torque curves for the drive and for a representative centrifugal pump are shown in Figure 15. The steepness of the motor torque provides a clue to its stiffness; that is, its ability to resist speed change because of a change in load torque. Motor torques are limited to some ceiling, such as 150% of normal, by the current limit circuitry of the drive. Table 6 provides significant data useful for drive selection.

DRIVE COMPARISONS

When comparing drives, factors to be considered include the following:

- **Drive costs—both initial and operating cost.** Cost considerations must include not only initial (installed) cost but also drive efficiency and power factor effects. Power factor may

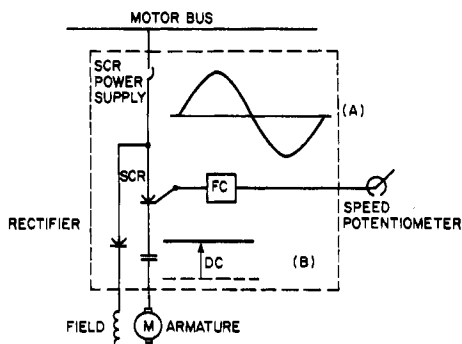


FIGURE 14 Block diagram for dc motor with SCR power supply

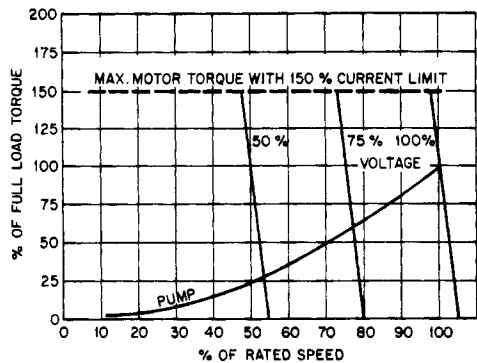


FIGURE 15 Torque versus speed for dc motor with SCR power supply

be of critical importance in influencing total costs highly dependent on the terms of the rate structure applied by the power supplier.

- *Operating characteristics.* All the drives discussed in this section compare favorably and all are considered generally suitable to perform functionally as pump drives.
- *Mechanical features* include enclosure, mounting, bearing capabilities, and arrangement (horizontal, vertical). Features are highly application related.
- *Mechanical simplicity* is one of most importance to operations personnel. It can consist of the number of wear points (parts) in rotating and control equipment. The rationale is

TABLE 6 Direct-current motor with SCR power-supply drive data

Drive element	Power rating*	Voltage rating	Max rated speed, rpm	Speed range, %	Enclosure	Mounting	No. of speed points
Motor	Open horizontal machines can be built at least up to 1500 hp (1120 kW); see Enclosure for other limits	240 500 550	Rating increases inversely with power rating; can match almost all pump speed requirements	At least 95	Vertical: open/dripproof, upthorough 300 hp, (224 kW) totally enclosed up through 200 hp (149 kW) Horizontal: open/dripproof, totally enclosed up through 200 hp (149 kW)	Vert. or horiz.	Infinite
SCR power supply	Can be built at least up to 1500 hp (1120 kW)	Any NEMA ac input voltage; will match motor voltage	...	At least 95	NEMA 1; NEMA 12	Floor	Infinite

*Drive can power either constant-torque loads or those varying as the square of the speed.

based on the concept that parts subject to wear are the eventual causes of failure; the smaller the number of parts, the less cause for failure.

- *Heat emission—removal requirements* can be critical depending on where the equipment is to be located. Heat can be emitted by both the equipment and the drive. If they are located in an enclosed area or building, heat emission can be an important issue.

In the final analysis, no simple routine can be suggested for drive selection. Each drive must be reviewed in the light of the application and its requirements. By reviewing the capabilities and features of the various drives, one has a starting point leading to successful drive selection.

The relative importance of these factors seldom remains stable but changes as application circumstances change.

9.2.3 VARIABLE SPEED FLUID DRIVES

MELBOURNE F. GIBERSON
CONRAD L. ARNOLD

INTRODUCTION

In almost all pumping applications, some means is required to provide variable flow and/or variable pressure of a liquid as it leaves a pump and enters a process application. Two alternative approaches exist, either (a) a variable speed drive for a pump, or (b) a constant speed drive for a pump with a discharge control valve that can provide a variable discharge flow to the process application.

The primary emphasis of this chapter is to focus on variable speed fluid drives for pumps such as boiler feed pumps, condensate pumps, and pumps for other fluids and applications. Nevertheless, for completeness, other types of variable speed drives for load equipment are briefly discussed.

TYPES OF VARIABLE SPEED DRIVES

Variable speed drives that are available today include fluid drives, mechanical drive steam turbines, electronic variable frequency drives, and magnetic drives.

In almost all situations, a pump and the associated variable speed drive will be substantially more efficient and more reliable than a constant speed pump with a discharge control valve. The choice of which variable speed drive to use usually depends upon the application, including such characteristics as the environment (temperature, dustiness, remoteness), the expected reliability, the importance of having critical parts not becoming obsolete, maintainability, capital cost, and overall efficiency.

Magnetic drives are discussed in Section 9.2.1. They have applications up to 300 or 400 hp (200 or 300 kW) or perhaps slightly higher. They are limited in power due to the difficulty of dissipating heat losses.

Electronic variable frequency drives can be used in a variety of applications. They are very common for low-power applications of less than 1000 hp (750 kW), and in other cases they can be used in applications over 10,000 hp (7500 kW). They can also be used for

applications up to several thousand rpm. With gear boxes, they can go to very high speeds or very low speeds. However, these drives consist of very sophisticated solid state electronics to convert incoming three-phase ac power at constant frequency (50 or 60 Hz) to dc power and then to invert the dc power back to variable frequency three-phase ac power. It is quite typical for isolation transformers to be used on the incoming power, for substantial air-conditioning packages to be used to cool the electronics, and for step up transformers to be used to convert the voltage to suit the motor. For high-power applications, the motors are usually made specifically for variable frequency drives in order to be able to handle the very high frequency components that arise in the solid-state electronics. A characteristic of these applications that should be addressed is the very high frequency pulsations that are inherent in the inverter process (dc to ac), as these pulsations affect both the electrical system and the mechanical system, and these pulsations are emphasized during rapid changes of the rotational speed of the motor and pump.

Mechanical drive steam turbines are used to provide variable speed power to pumps of many different applications, small to very large. In main power plant applications, steam from an extraction port of the main turbine can be used to drive variable speed steam turbines that drive boiler feed pumps.

Variable speed fluid drives of the type known as "hydro-kinetic" are the most important ones for driving larger pumps, and for this reason, they are the focus in this chapter. This type of fluid drive can be made from a few horsepower to over 40,000 hp (30,000 kW). However, today, most hydro-kinetic fluid drives are made for powers over 300 to 400 (200 to 300 kW). For powers under this level, electronic variable frequency drives and the other types of fluid drives are used more often. (As an aside, the terms fluid coupling and hydraulic coupling are sometimes used interchangeably with fluid drive.)

A hydro-kinetic variable speed fluid drive usually takes input power from a constant speed power source such as a motor of either induction or synchronous type and delivers variable speed power to a process pump. The fluid drive output shaft speed (pump speed) is controllable in step-less speed changes in a range, typically, from 20 to 97.5% of the input shaft speed. The output shaft speed is extremely smooth, that is, there are no torsional excitation pulsations. An interesting and valuable application for certain main power plants occurs when input power is taken from the main turbine-generator shaft into a fluid drive and the fluid drive delivers variable speed power to a large boiler feed pump. This arrangement is called a "shaft-driven fluid drive and boiler feed pump."

It should be noted that every type of variable speed drive can be used with a constant ratio gearbox to achieve the actual speed range desired by the pump application. However, the only type of variable speed drive that includes a gearbox within the drive is the type called "variable speed geared fluid drive."

DESIGN OF HYDRO-KINETIC VARIABLE SPEED FLUID DRIVES

The design of a "hydro-kinetic" variable speed fluid drive with an oil-conditioning system is shown in Figure 1. The primary mechanical components of a variable speed fluid drive assembly are (a) an outer housing or tank that supports the bearings, holds the oil, and provides a protective cover, (b) an input element comprising an input shaft and coupling hub, an impeller, an intermediate casing, and an outer casing (or scoop tube casing) all attached to each other and rotating together, (c) an output element comprising an output shaft, a coupling hub, and a runner all attached to each other and rotating together, (d) journal and thrust bearings that support each shaft, (e) oil used as a power transmitting medium between the impeller and runner and as a lubricant for the bearings, (f) a moveable scoop tube that slides in and out to control the amount of oil that remains resident in the rotating casings, and (g) an oil-conditioning system.

The impeller and runner are structures with vanes and pockets between the vanes shaped much like a half of a grapefruit with the linings that separate the fruit intact and the fruit removed. The impeller and a runner face each other with a small gap between them, on the order of 1% of the diameter of the impeller (or runner), but they do not touch each other mechanically.

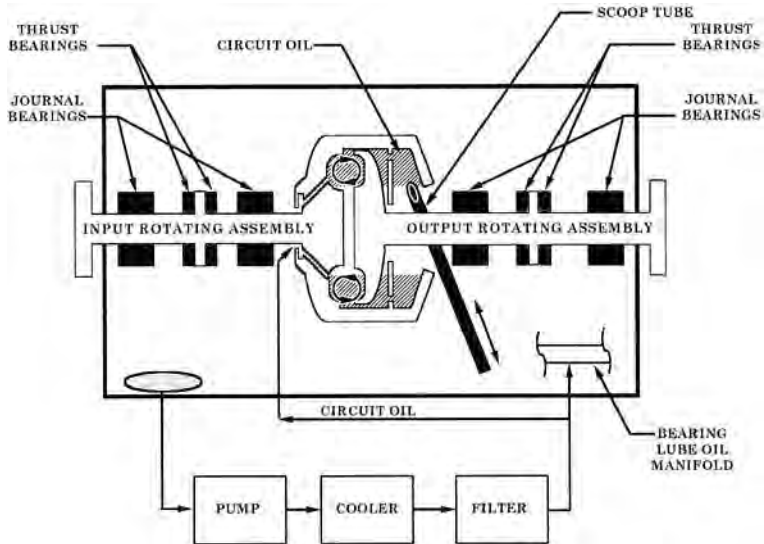


FIGURE 1 "Hydro-kinetic" variable speed fluid drive

The impeller and the attached rotating casing structure retain the working fluid, usually oil, as shown in Figure 1. The runner is located within the envelope of the rotating input element. The actual mechanism causing the runner to rotate is the kinetic action of the oil: the impeller transmits energy to the oil and the oil transmits energy to the runner, as described below.

Drives of the variable speed fluid drive type are sized according to the equation:

$$\text{Power (hp)} = \frac{K \times (N_{\text{out}})^3 \times D^5}{10^{14}}$$

where

K = approximately 2.1 (for hp), but depends on many design details such as exact pocket shape, vane angles, and the circuit oil flow rate

$N_{\text{out}} = 0.975 \times N_{\text{in}}$ (rated speed point) (rpm)

D = Hydraulic diameter of the impeller and runner (inches)

Figure 2 is a chart showing power (hp and kW) as a function of speed for various hydraulic diameters. The actual sectors that are applicable for different hydraulic diameters will vary slightly depending upon the many design details of a drive. Consequently, this chart should be considered to represent trends for different fluid drives, but should not be considered to represent a specific fluid drive.

A fluid drive of the hydro-kinetic variable speed design operates as follows. The tank contains a level of oil, often described as ISO VG 32, light turbine oil. An oil pump supplies cooled and filtered "lube oil" to the bearings and "circuit oil" or "working oil" to the "circuit," the cavity comprising the pockets between the vanes of the impeller and runner, as shown in Figure 1.

Because the impeller and runner are rotating, the circuit oil is thrown outward under centrifugal force, so that the oil forms a torus in the rotating casing assembly. The impeller is being driven by a driver, generally at a fixed speed, and the impeller rotates faster than the runner. Consequently, oil that is in the impeller is subject to greater centrifugal acceleration than oil that is located in the runner pockets. Consequently, a parcel of oil

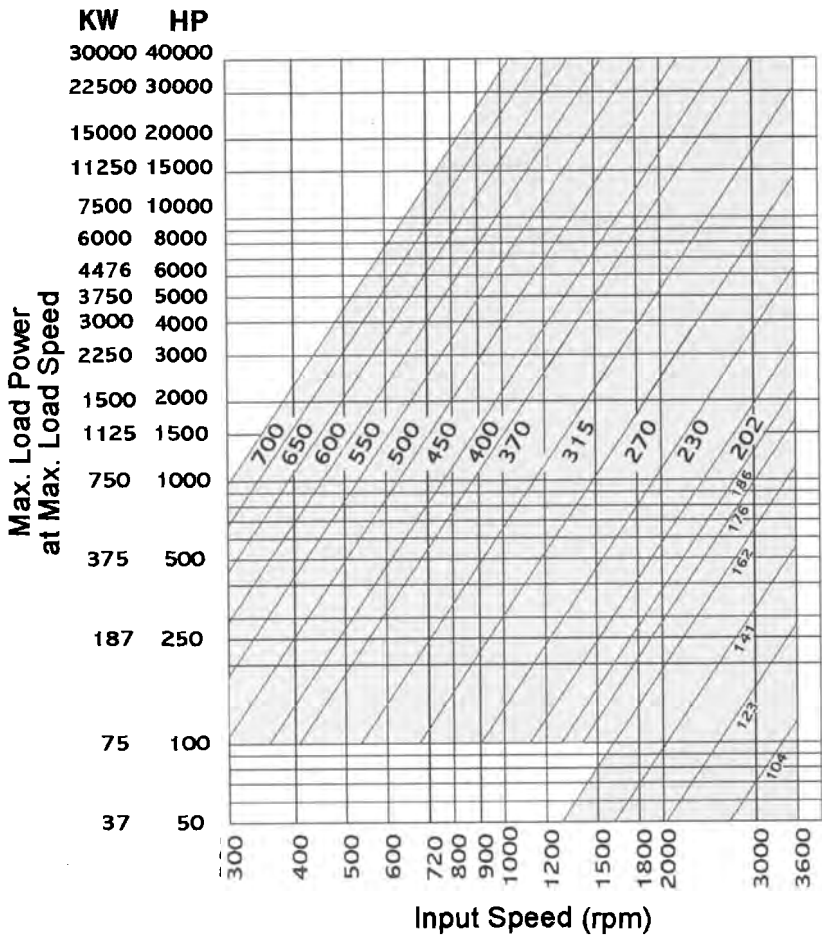


FIGURE 2 Sizing chart for fluid drives. Hydraulic diameters are shown on diagonal (For example, “370” signifies 37.0 in)

that starts in an impeller pocket moves in a generally circular path (relative to the rotating pocket) to the radially outer portion of that impeller pocket, gaining momentum along the path, and leaves the impeller pocket traveling across the gap with both circumferential and axial velocity components. Then the packet of oil enters a runner pocket where, because the runner is rotating slower than the impeller, the oil packet hits the wall of that runner pocket and imparts momentum to the runner vane. Then the packet of oil moves radially inward along a generally circular path (relative to the runner pocket) until it leaves the runner pocket in a path with axial and circumferential velocity components, and enters an impeller pocket where, because the impeller is rotating faster than the runner, the wall of that impeller pocket hits the packet of oil and imparts momentum to it. Then the packet of oil again follows a circular path (relative to the impeller pocket) radially outward within the impeller pocket until it again crosses the gap to enter the runner.

The combination of both circular paths within the pockets and a circumferential path of the pockets around the shaft centerline provides a general trajectory of each packet of oil that is a spiral that travels in circles, as shown in Figure 3.

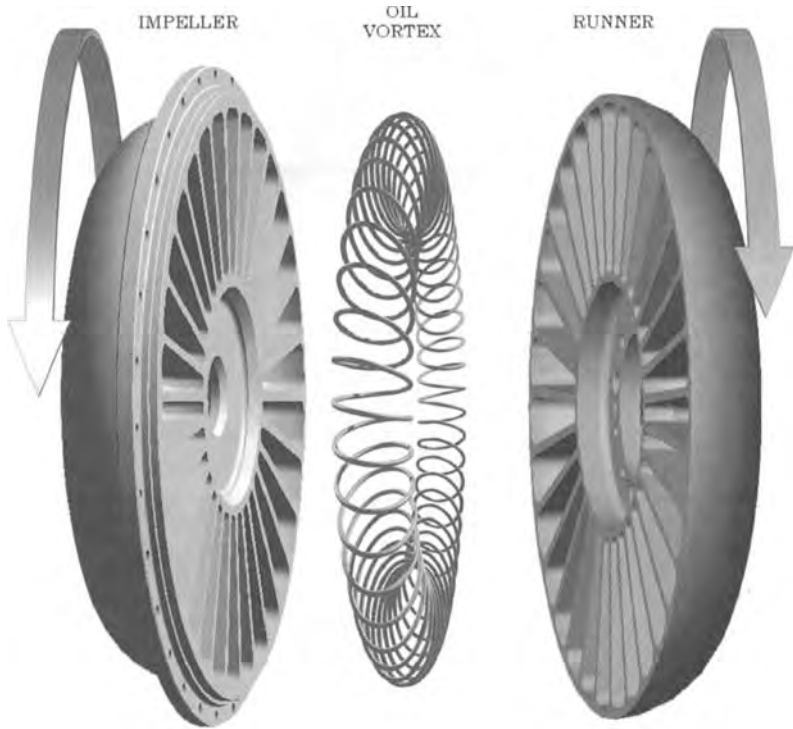


FIGURE 3 The trajectory of each packet of oil is a spiral through the impeller and runner as they rotate about the shaft centerline

Note that in a hydro-kinetic fluid drive, the energy is transmitted from impeller to runner using the principal of “momentum exchange”: The vane of the impeller hits a packet (mass) of oil and increases its velocity, and then the packet (mass) of oil hits the vane of a runner and the velocity of the oil is reduced, transmitting momentum, and the process continues.

Fluid drives are “constant torque” machines, that is, the torque that is absorbed by the impeller from the input shaft to drive the circuit oil equals the torque that the circuit oil imparts to the runner that drives the output shaft and the load, which, in the present case, is a pump.

The degree of coupling between the impeller and runner depends upon the level of filling of the impeller and runner chamber, and the level of filling is controlled by the position of the scoop tube tip. The more mass of oil that can traverse the spiral path, the more coupling that exists. The degree of coupling is controlled by the radial position of the scoop tube tip.

A scoop tube is a hollow tube with a tip that is shaped to scoop oil from the inside surface of the rotating torus of oil and to drive the oil through the tube so that it exits the rotating fluid drive element.

Figures 4a and 4b show two different levels of filling. In Figure 4a, the scoop tube tip is moved radially outward (away from the shaft centerline) providing a small torus of oil with a small mass of circulating oil between the impeller and runner, with the result of low coupling.

In Figure 4b, the scoop tube is moved radially inward (toward the shaft centerline), providing a large torus of oil with a large mass of circulating oil between the impeller and runner with the result of high coupling.

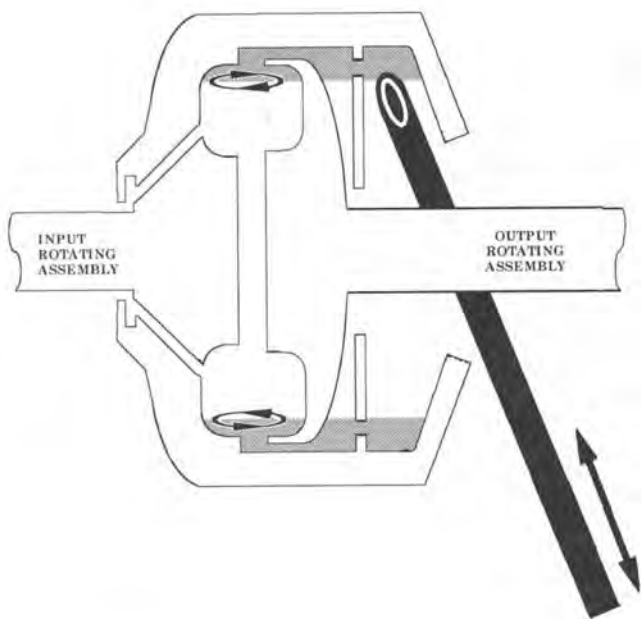


FIGURE 4A A small torus of circuit oil provides small coupling

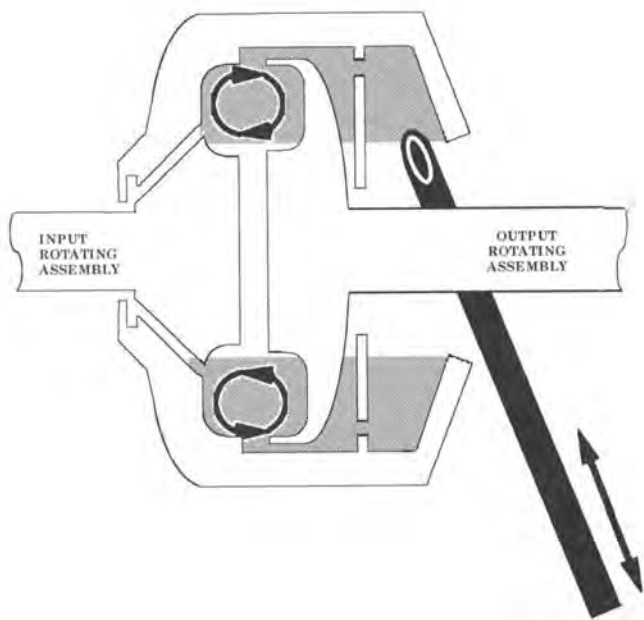


FIGURE 4B A large torus of circuit oil provides large coupling

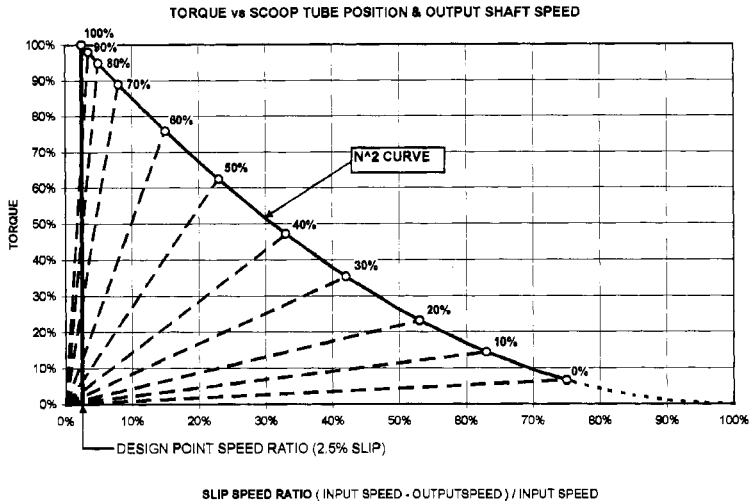


FIGURE 5 Chart of output torque ratio versus output speed ratio for different scoop tube positions

Figure 5 presents a generalized chart that shows the torque transmission of a fluid drive as a function of scoop tube position and output shaft (pump) speed. It is important to realize that the locations of the constant scoop tube position lines vary according to many design details including the pocket shapes in the impeller and runner, as well as vane thicknesses, vane angles, and the circuit oil flow. Consequently, this chart should be considered to represent the generalized trends of variable speed fluid drives and not any particular fluid drive.

FUNDAMENTAL EQUATIONS FOR FLUID DRIVES (USCS UNITS)

Several fundamental equations for fluid drives are given below:

$$\text{Torque in} = \text{torque out} = \text{torque (lbf-in)}$$

$$\text{Power in} = \text{torque in (lbf-in)} \times \text{input rotational frequency (rad/sec)}$$

$$\text{Power out} = \text{torque out (lbf-in)} \times \text{output rotational frequency (rad/sec)}$$

$$\text{Power loss} = \text{power in} - \text{power out} = \text{torque (lbf-in)} \times (\text{input rotational frequency} - \text{output rotational frequency}) \text{ (rad/sec)}$$

$$\text{Heat power (btu/h)} = \text{power loss (hp)} \times 2544 \text{ btu/hp} \cdot \text{h}$$

It is the heat that is generated by the lost power that requires the circuit oil in a fluid drive to be exchanged continuously.

Certain losses are not included in the above equations. These include bearing power losses and circuit oil handling losses. Bearing losses for the input shaft are constant and for the output shaft increase with increasing speed. Circuit oil handling losses are a function of scoop tube position and circuit oil flow rate. Most fluid drives have a constant, or almost constant, circuit oil flow rate, so the circuit oil handling losses are greatest when the scoop tube tip is at the largest distance from the shaft centerline, which corresponds to minimum output shaft (or pump) speed, and these losses are the least when the scoop tube tip is closest to the shaft centerline, which corresponds to maximum output shaft (or pump) speed. The net result is that the sum of the bearing losses and circuit oil handling losses can be considered to be a constant, typically ranging from 3 to 4% of the rated power of the fluid drive.

The efficiency of a variable speed fluid drive of the hydro-kinetic type can typically be expressed as:

$$\text{Efficiency} = (\text{output shaft speed/input shaft speed}) - 3.5\%$$

It is very common to use a slip speed (input speed – output speed) of 2.5% at the design point, though in some applications, the slip speed at the rated operating point is from 2.0% to over 4%. Using a slip speed of 2.5% at the design point, the design point efficiency is $100\% - 2.5\% (\text{slip loss}) - 3.5\% (\text{mechanical losses}) = 94\%$.

If a set of gears is included within the fluid drive tank and it is located on the output side of the fluid drive circuit, the runner output shaft and a gear shaft can be combined, eliminating two journal bearings and a pair of thrust bearings. In this case, the power losses will increase by approximately 1.5%, rather than the normal 2.0 to 2.5% additional loss when an independent gearbox is used between the fluid drive and the pump.

COMPARISON OF EFFICIENCIES OF VARIABLE SPEED DRIVES

When comparing efficiencies for alternative types of variable speed drives, it is essential to determine the following factors:

- First, before the efficiency is calculated it is essential to determine if the equipment being evaluated will provide variable speed power under all of the conditions that are required.
- Second, determine if the equipment will respond as fast to a change of load as is required or desired.
- Third, determine if the efficiency of alternate types of equipment be evaluated at several operating points from low load, say 20%, to full load being included, or will the evaluation be based on only one operating point, such as full power.

Many types of equipment can be designed to operate well at or near a single operating point such as full load, but they do not perform well when operating away from the design point of full load.

A good way to evaluate alternative forms of variable speed drives is to examine the annual costs for operating each complete drive train at *several operating points*, giving weight for each load point in proportion to the number of hours that the equipment is expected to operate at each of these points during a year. The reason is that when operating at or near full load, there is high power, there is high efficiency, and there are usually a large number of hours, so the actual costs at this operating point are quite favorable. Then there are times when the drive train and pump will be operating at low load, at which time the efficiency is low, very little power is transmitted, and the number of hours at this point is limited, with the consequence that, when averaged with the high efficiency operating points at high load conditions, the average efficiency is quite satisfactory. When performing these evaluations, it is very critical that an accurate value of the actual power consumption for each evaluation point be obtained and used. This includes such things as (a) for steam turbines, the actual heat rate at "off design" conditions when extraction steam is not available in sufficient quantity and main steam or steam from a "pony boiler" may be needed, and (b) for electronic variable frequency drives, the air-conditioning load for the electronics as well as the transformer, wiring, and motor losses, especially the losses due to the very high frequencies.

CONSTRUCTION OF FLUID DRIVES AND THE OIL-CONDITIONING SYSTEMS

Variable speed fluid drives are built by a number of manufacturers around the world. Most manufacturers have developed a range of designs of fluid drives for specific applications,

and these that have become "standardized lines" for that manufacturer. Many applications require specialized designs that are not among the standardized designs, due to the environment, power, speed, or turndown conditions, and, upon request, manufacturers will often provide those specialized designs. Fluid drives can be made with horizontal shafting or vertical shafting.

The materials of construction of fluid drives vary widely. Outer housings and certain bearing housings are usually made of fabricated steel, may be split horizontally or may be stacked from one end, and may be made from thick steel or from thin steel. In certain cases, the outer housings and certain bearing housings may be made of cast iron or cast steel.

The impellers and runners may be cast aluminum of a variety of grades and heat treats, cast steel, fabricated steel, or milled from solid forgings of alloy steel. For a fluid drive with a hydraulic diameter of 25 to 27 in (64 to 69 cm), the impeller and runner pockets are each about 6 in deep axially and the gap between the impeller and runner faces is on the order of $\frac{1}{4}$ to $\frac{3}{8}$ in (6 to 9.5 mm). Pocket sizes and gaps for other fluid drive sizes are larger or smaller in proportion to the size of the fluid drive hydraulic diameter.

Shafts and other rotating casings are usually made of carbon steel or alloy steel. Heat treatment may or may not be used. The journal and thrust bearings can be rolling element design or Babbitted bearings, and the choice depends upon the speed, size limitations, operating duty, required life, and price. Rolling element bearings come in a range of duties and L_{10} lives. Babbitted bearings may be made of simple fixed shapes or they can be made with extremely robust tilting pad designs. Flexible coupling types range from gear to diaphragm styles. They may be supplied by the customer or the manufacturer, or, in some cases, the hubs may be made integral with the shafts. Instrumentation such as thermocouples or resistance temperature devices (RTDs) and vibration pickups, proximity or seismic, may or may not be included.

Oil-conditioning systems can be extremely simple, comprising one pump, electric or shaft driven, one cooler, one filter, and one sight glass to determine the oil level in the tank. Alternately, they can be quite complex, being built according to the highest level of American Petroleum Institute (API) specifications with duplex ac main oil pumps, a dc emergency pump, duplex oil coolers, duplex filters, all stainless steel piping, various isolation valves, pressure control valves, actuator for the scoop tube, additional lube oil requirements for the main motor and main pump, and with electronic transmitters for pressure, temperature, and oil levels, for interfacing with a distributed control system (DCS). Because there are no industry standards that broadly apply to all fluid drives, it is essential for the user to detail all of the desired features in the purchasing specification. As will be seen below, the heat loss in a fluid drive depends upon the specific operating points in the application, and several operating points should be provided to the manufacturer, including the maximum power and speed point, the minimum power and speed point, and two or three interim operating points, especially the lowest pressure with the maximum flow. The reason for providing such details is that there is no reason to use larger pumps and/or larger heat exchangers than are required to meet the user's operating conditions and, at the same time, to maintain the maximum oil temperatures below the oxidation limits of the particular oil to be used (generally around 185°F (85°C)). When gears are included in the mechanical design of the fluid drive, the service factors for gear sets located on the output side of the fluid drive do not need to be so high as when they are located on the input side, particularly when synchronous motors are involved.

Because there is such a wide range of design features that can be used in fluid drives, and because the technology content is the principal controller of price, the pricing can vary widely for different fluid drives for the same maximum power and maximum speed. In other words, specifications addressing only the max speed and max power are not sufficient to specify a fluid drive adequately, especially the oil system for a fluid drive, because the most difficult operating points for fluid drives are in the "turndown" operating points when the output shaft speed is in the neighborhood of $\frac{2}{3}$ of the input shaft speed. It is extremely helpful to the fluid drive manufacturer/designer if the user actively participates in developing the design details of the equipment.

When a variable speed fluid drive and its mating oil-conditioning system are properly designed for a given application, they become relatively simple mechanical systems with extremely high reliability and availability. It is quite common for variable speed fluid drives

to operate for over 10 years without inspection or any maintenance other than changing oil filters. When maintenance of a fluid drive is required, parts can be obtained or can be made as required. It is rare for a fluid drive design to become obsolete, which has been a problem on certain types of electronic variable frequency drives.

The speed of the output rotating assembly is controlled by the position of the scoop tube tip, which is typically positioned by a scoop tube actuator, being driven by a 4 to 20 mA signal from the plant control system, such as the central DCS, for the entire process unit.

It is interesting to note that the fluid drive of the hydro-kinetic design was invented by Dr. Herman Föttinger in Germany in 1905, and fluid drives of this design are now used all over the world. A textbook on fluid drives, also called fluid couplings, that may be useful to those who have an interest in technical details is entitled *Stromungskupplungen und Stromungswandler, Berechnung und Konstruktion*, by Ing. Maurizio Wolf, Springer-Verlag, Berlin/Göttingen/Heidelberg, 1962. Fluid drive designs have been tuned and adjusted and adapted to many applications, but the fundamental features, other than instrumentation, have not significantly changed.

POWER LOSS AND TEMPERATURE CONDITIONS

A **very simplified approach** for evaluating the heat loss of a variable speed drive assumes that the power required by a pump is only proportional to the cube of the pump speed (or output shaft speed). This is correct under certain conditions, but because pumps operate at points on the head-capacity plot that are at some distance from this simplified curve a more appropriate evaluation of the heat loss of a variable speed fluid drive for boiler feed pump service is given below via a series of head-capacity charts.

In the simplified approach, assume that the power of a pump is proportional to the cube of the pump speed (N) (or fluid drive output shaft speed):

$$(\text{Power out}) = (N_{\text{out}})^3$$

$$\text{And } (\text{Torque out}) = (N_{\text{out}})^2$$

$$\text{Now } (\text{Torque in}) = (\text{torque out}) = \text{torque}$$

$$\text{Hence } (\text{Power in}) = (\text{torque in}) \times (N_{\text{in}})$$

$$(\text{Power in}) = (N_{\text{out}})^2 \times (N_{\text{in}})$$

$$\begin{aligned} (\text{Power loss}) &= (\text{power in}) - (\text{power out}) \\ &= (N_{\text{out}})^2 \times N_{\text{in}} - (N_{\text{out}})^3 \end{aligned}$$

$$\text{The maximum loss occurs at } (N_{\text{out}}) = 2/3 (N_{\text{in}})$$

The maximum loss at this output shaft (pump) speed is approximately 15% of the rated input power. Including bearing and circuit oil handling losses, the maximum loss is on the order of 16 to 17% of the rated input power.

The circuit oil flow pattern within the fluid drive chamber consisting of the impeller pockets and runner pockets and the axial gap between them can be considered to be a spiral in general, Figure 3, but it is extremely turbulent.

The turbulence of the circuit oil is affected by the speed difference, the torque transmitted, and the amount of circuit oil in the rotating element. As indicated above, the speed at which the maximum power loss occurs is when the output shaft is at approximately 2/3 of the input shaft speed, which is also the speed at which the greatest turbulence occurs. Thus, for a fluid drive input speed of 3600 rpm, the maximum turbulence occurs when the output shaft speed (pump speed) is in the range of 2400 rpm. It is also important to note that for a given output shaft speed both the turbulence and the power loss increases with increased torque.

Five head-capacity charts are provided below that are based on the head-capacity plot for a main boiler feed pump for a 375-MW turbine-generator. The head is shown in units of psig because the intent of these charts is to show trends for understanding of the issues at hand rather than to be completely accurate. For specific applications, the grids can be

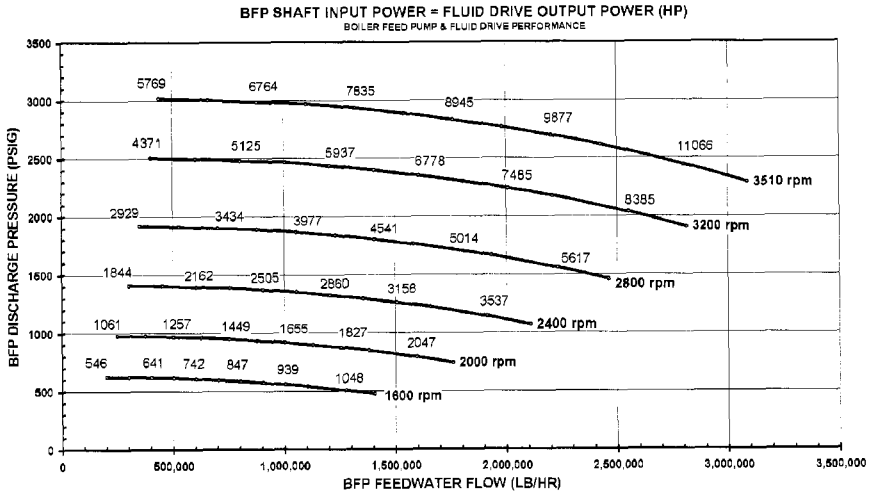


FIGURE 6 (Chart 1) Power absorbed by the pump shaft from the fluid drive output shaft

calculated with all of the normal details considered, but it will not change the principles at hand. The grids on the charts are made of constant speed lines (generally horizontal) and constant efficiency lines (generally curving upward). The grids are for the same conditions on all of the charts. Data are presented at certain intersection points of the grids.

Figure 6 (Chart 1) presents the shaft power absorbed by the pump (out of the fluid drive).

Figure 7 (Chart 2) presents the shaft power absorbed by the fluid drive (out of the motor or turbine-generator) at the same intersection points of the same grid.

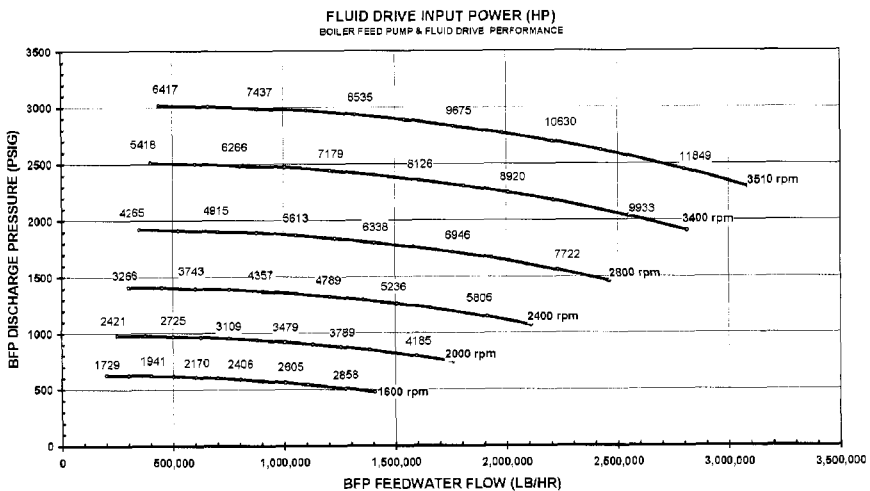


FIGURE 7 (Chart 2) Power absorbed by the fluid drive input shaft from the motor or turbine-generator output shaft at the same operating points of the grid

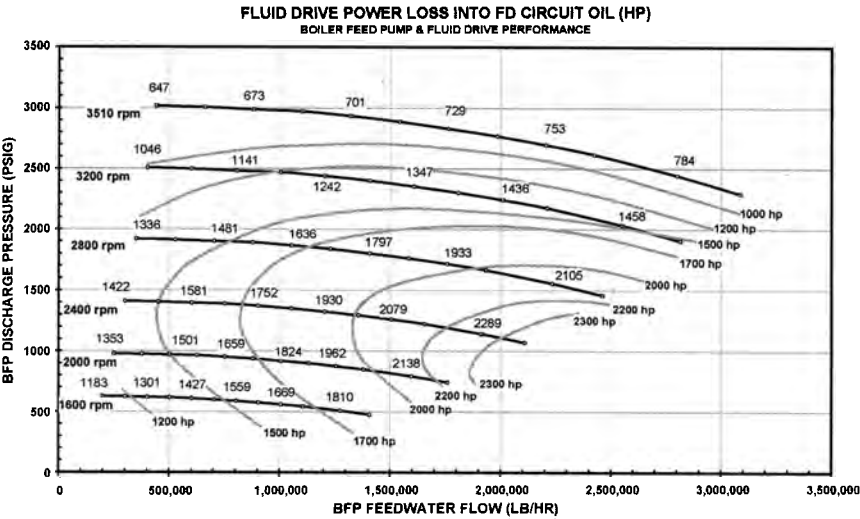


FIGURE 8 (Chart 3) Power lost to the circuit oil in the fluid drive element (equals power into the fluid drive minus power out of the fluid drive) at the same operating points of the grid

Figure 8 (Chart 3) presents the power lost to the circuit oil in the element (power into the fluid drive minus power out of the fluid drive).

Figure 9 (Chart 4) presents the circuit oil discharge (COD) temperature that results from the power lost in Chart 3 with the circuit oil input temperature of 120°F (49°C) and a constant circuit oil flow rate of 400 gpm (91 m³/h).

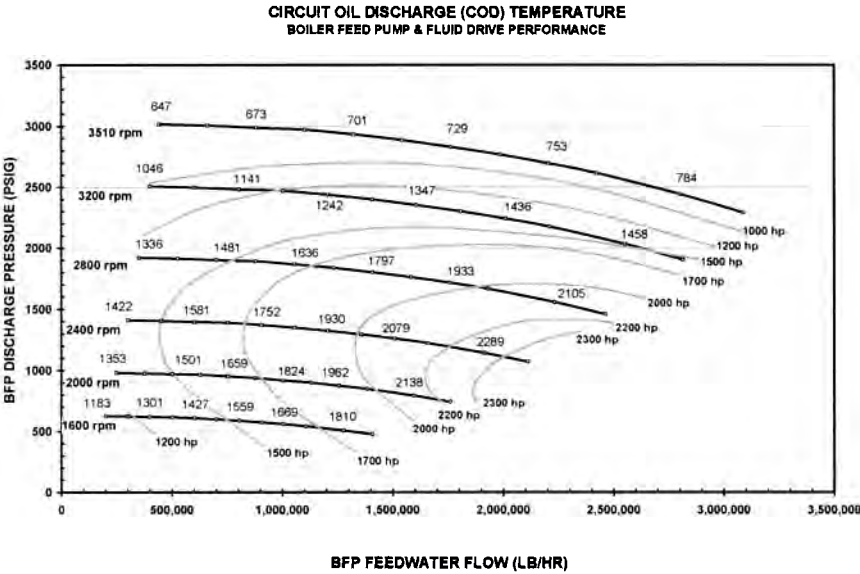


FIGURE 9 (Chart 4) The circuit oil discharge temperature that results from the power lost per Chart 3 with the circuit oil input temperature of 120°F (49°C) and a constant circuit oil flow rate of 400 gpm (91 m³/h)

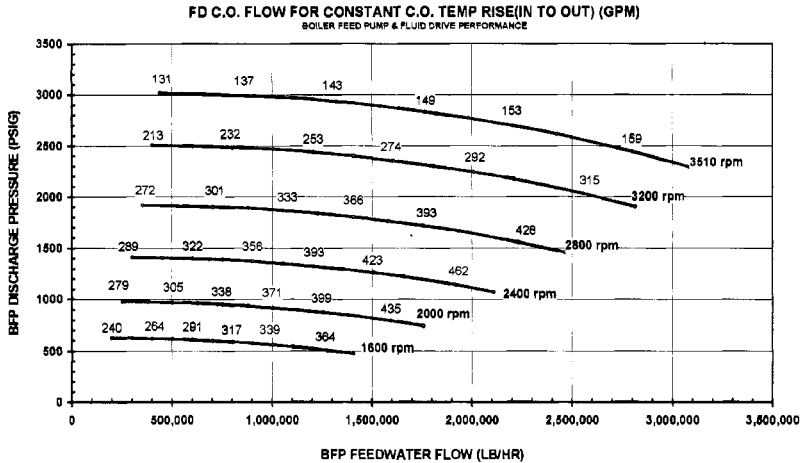


FIGURE 10 (Chart 5) Circuit oil flow rate (gpm) required to maintain a constant temperature rise of the circuit oil (the difference between the circuit oil temperature entering the fluid drive and the circuit oil discharge (COD) temperature from the fluid drive rotating element)

Figure 10 (Chart 5) presents the circuit oil flow rate (gpm) required to maintain a constant temperature rise of the circuit oil (the difference between the circuit oil temperature entering the fluid drive and the COD temperature from the fluid drive rotating element).

For instance, in Figures 6 through 10 (Charts 1 through 5), following the constant speed line for 2400 rpm from left to right, it can be seen in these charts that power transmission increases, the torque increases, the power (heat) losses increase, and the COD temperature rises. All of this is associated with increased circuit oil mass flow and associated increased turbulence of the circuit oil when traversing the constant speed line going from low boiler feed pump flow to higher pump flow.

CONSTANT THROTTLE PRESSURE OPERATION VS. “REDUCED PRESSURE” OR “SLIDING PRESSURE” OPERATION

In the 1960 era, almost all turbine-generators were originally designed for constant pressure at the throttle, examples being 1800 psig, 2000 psig, 2400 psig, and at super critical pressures above 3600 psig. The corresponding system resistance curve for the boiler feed pump is a line beginning at the rated operating point 11,066 hp on the 3510 rpm curve on the head-capacity chart of Figure 6 (Chart 1) and traverses the chart with a downward slope until it reached the minimum flow point for the pump at 2929 hp on the 2800 rpm curve. Consequently, the fluid drive of this example would never have to operate below 2700 rpm. The typical operating procedure, at least as anticipated in the design stage, was that the start-up pump would be used to get the unit up to minimum load and full throttle pressure. This process works, and is used for super-critical units that have higher operating pressure than this example, but it is almost never used by plant operators for subcritical T-G units. As can be seen by inspecting these BFP head-capacity charts, keeping the operating pressure high so that the output shaft speed is always above 2700 rpm has the result that the heat dissipation in the circuit oil is limited to less than approximately 1500 hp, as shown in Figure 8 (Chart 3). Two important consequences are (a) that the heat exchangers do not have to be sized for the higher heat dissipation (on the order of 2200 hp) that occurs if the fluid drive were to be operated at 2400 rpm and at maximum flow conditions of approximately 2,000,000 lbs/h at this speed, and (b) the circuit oil pump

size can be limited in size for similar reasons, both of which save capital cost, and physical space on the oil-conditioning skid.

Most operators of subcritical units operate these units in a different manner, as follows: The typical start-up objective is to get the T-G unit onto the main pump as soon as is practical. The pressure is limited to the "silica" limit, say 1200 psi at first, and the flow would be increased to the maximum possible by operating the throttle valves to the valves wide open condition. Two benefits are achieved by operating in this manner: A greater feed water flow will clear the boiler water faster and, hence, the boiler pressure can be increased more rapidly. The second benefit is that the electrical power generated is maximized during the water cleanup process. The heat dissipated by the fluid drive at 1200 psi discharge and 500,000 lbs/h is around 1500 hp, similar to the amount of heat dissipated at points along the system resistance curve for the constant throttle pressure application. However, as the flow is increased, the heat dissipated increases, and if the heat exchangers and circuit oil pumps are not adequately sized for "reduced pressure" or "sliding pressure" operation, then the fluid drive COD pressure will get hot, up to the 200°F (93°C) range, as shown on Figure 9 (Chart 4).

A reason for limiting the oil temperature to 185°F (85°C) is that, above this temperature, oil in the presence of air will degrade. Another reason for limiting temperature is that, if water enters the oil, the higher temperature, the faster sludge will develop, and sludge can centrifuge out into the casings causing unbalanced conditions with associated increased vibration levels.

When the fluid drive and the oil system are in the specification and/or design stage, the solution to avoid overheating conditions involves accounting for the heat to be dissipated and the circuit oil flow rates that should be used to limit oil temperatures to 180 to 185°F (82 to 85°C) at the hot operating conditions that correspond to low speed, high BFP flow, and adjusting the plant cooling water heater balance accordingly.

If the fluid drive is already in BFP service for a turbine-generator, and there is a desire to operate the unit with "reduced pressure" or "sliding pressure," then it is necessary to identify the additional heat to be dissipated, and then install additional heat exchanger and circuit oil pumping capacity. The most effective heat exchangers are usually large, single pass counter-flow designs. Heat exchangers of this design may not be the lowest cost by themselves, but they will likely obtain the lowest oil temperature leaving the coolers and they will use the least amount of plant cooling water, providing significant economic benefits.

While more heat is lost in a fluid drive on sliding pressure operation than on constant throttle pressure, the amount of power saved in the pumping process and in the reduction of throttling losses in the steam turbine is greater, providing a substantial net improvement in the overall cycle efficiency. For instance, the input power to the fluid drive in Figure 6 at full pressure and 350,000 lbs/h is 4265 hp, whereas on sliding pressure conditions for the same megawatt, which uses about the same feed water flow, the input power is about 2500 hp, a savings of 1765 hp, or 1.3 MW, which at \$70/MW represents around \$100/h. This savings is only the reduced power to drive the pump, and does not include reduced throttling losses, and reduced fan airflow and other savings.

Sliding pressure operation, including unconditional operation anywhere on the head-capacity map, usually has tremendous economic benefits. In most cases where the process can use these capabilities, the improved operating conditions can pay for the slight increase in capital cost for a heavy-duty fluid drive and oil system in a matter of a few months. A reference book for analyzing sliding pressure operation in steam turbines is entitled *Evaluating and Improving Steam Turbine Performance*, by Ken C. Cotton, Second Edition, published by Cotton Fact in 1998.

Use of a Control Valve to Maintain Uniform COD Temperatures At very low speed, low BFP flow operation, the minimum speed at low or zero percent scoop tube is substantially controlled by the circuit oil flow rate, control of the pump speed, and hence, the BFP flow is improved by minimizing the circuit oil flow to that required to keep the COD temperature below approximately 185°F (85°C). This suggests that circuit oil flows should be reduced, which is the opposite of the solution for the high heat condition, discussed above.

The compromising solution to this circuit oil flow issue is to install a COD temperature control valve that limits the circuit oil flow to the fluid drive in order to maintain a constant

COD temperature. Depending upon the application, this COD temperature set point is set in the neighborhood of 180°F (82°C). The results of the use of this COD valve are demonstrated in Chart 5, which is based on the heat dissipated in Chart 3 and upon a constant differential temperature (COD temperature—circuit oil supply temperature) of 70°F (21°C). In this chart, the COD temperature remains constant and the flow varies, whereas in Chart 4, the circuit oil flow is constant and the COD temperature varies. Additional benefits arise:

1. The circuit oil flow is limited in the low output shaft speed range, and this increases the effectiveness of the scoop tube to control the output shaft/pump speed, reducing the minimum output shaft/pump speed when the scoop tube is at the minimum power position, and yielding greatly improved feedwater control throughout the low speed range.
2. The circuit oil pump size can be increased to suit the specific application so that when the fluid drive is operated in the neighborhood of 2400 rpm and it is generating a large amount of heat, the circuit oil flow is adequate to limit the maximum temperature of the circuit oil discharging from the fluid drive element to approximately 180 to 185°F (82 to 85°C).
3. The power required to handle the circuit oil is reduced at operating conditions away from the high-heat zone, increasing the operating efficiency of the unit.
4. A constant temperature fluid drive maintains uniform elevation height of the input and output shafts and maintains uniform thermal expansion of the housing and all internal parts. This extends the life of the fluid drive by minimizing low cycle stresses, relative movement, or fatigue of the various parts.

TYPICAL VIBRATIONS OF HIGH POWER VARIABLE SPEED FLUID DRIVE ROTORS

In almost every fluid drive, the first critical speed of the input shaft and the first critical speed of the output shaft is each measurably above 3600 rpm (60 Hz). The mode of vibration of each shaft in response to unbalance is typically a rigid body conical mode. That is, there is almost no bending of the shaft and the critical speed of each shaft is controlled primarily by the stiffnesses of the oil films of the bearings and the structures that support the bearings for that shaft.

Depending upon the condition of the journal bearings, it is possible to experience some subsynchronous vibration such as "oil whip" which occurs at a fractional frequency, generally below half running speed. This type of vibration should not be confused with a very unique type of vibration that often arises in fluid drives, discussed below.

The journal at the input-inboard bearing of a high-powered variable speed fluid drive typically has the highest amplitude vibrations because it is this journal that is experiencing the largest vibratory forces, both from the heavy casings of the input rotor and from the unbalance of the turbulent oil contained in these casings. A typical vibration frequency analysis for this input-inboard journal appears as indicated in Figure 11.

The unique features of this figure are that three-vibration components usually appear in this analysis: (a) the synchronous component of the input shaft, which in this case is 60 Hz, (b) the output shaft rotational frequency, 39 Hz, and (c) a frequency (41 to 44 Hz) that is slightly above the output shaft rotational frequency corresponding to the average effective rotational frequency of the circuit oil in the rotating element.

ALTERNATE ARRANGEMENTS OF DRIVERS, FLUID DRIVES, AND BOILER FEED PUMPS FOR TURBINE-GENERATOR SERVICE

These days, there is great emphasis of improving the overall efficiency of fossil-fired generating plants. It is very common to design every new plant as if it is going to operate at only one point: full load. However, once built, many have to cycle to a reduced load at night,

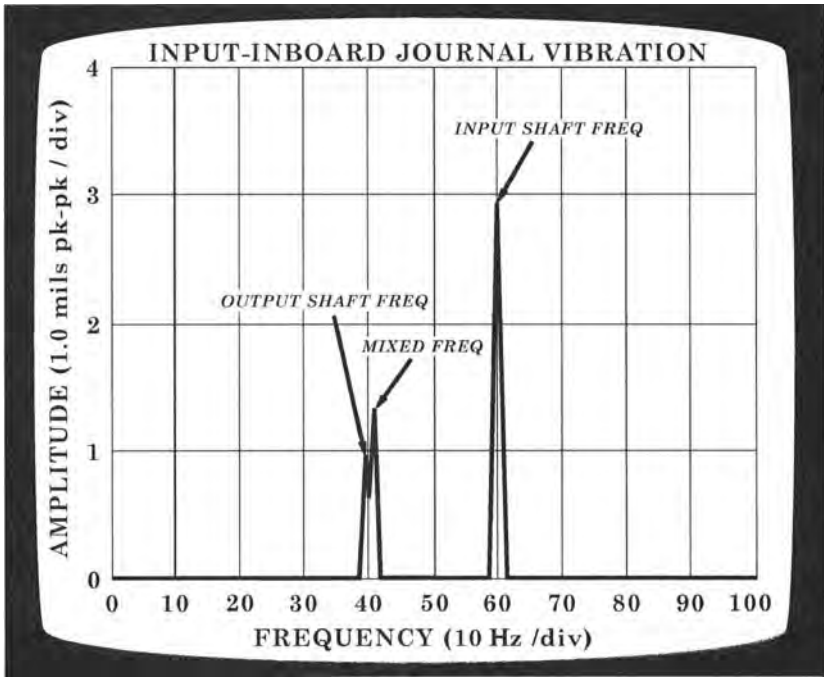


FIGURE 11 A typical vibration frequency analysis of the journal at the input-inboard bearing of a high power variable frequency fluid drive

and this includes those that use fossil fuels. In this case, having the unit designed so that it is relatively efficient at full load as well as over a range of operating points down to low load is more appropriate. To this end, an arrangement that should be considered is to have a shaft-driven fluid drive and 100% capacity boiler feed pump attached directly to a steam turbine-generator, and to have the entire plant designed for sliding pressure at the turbine throttle valve.

In the 1960s, many such steam turbine-generators were built this way, that is, with a single 100% shaft-driven fluid drive/boiler feed pump attached to either the turbine or generator extension shaft. A number of these units had heat rates at the design point on the order of 9300 btu/kW · h. Several were built at the 325 MW range. Ravenswood 3, owned by Consolidated Edison of New York, Inc., was the world's first 1000 MW unit, commissioned in 1965. It was built for sliding pressure and, incidentally, it had coal-fired boilers. This is a cross-compound unit. There are two fluid drive/boiler feed pumps, each rated at 18,000 hp. There is one set on each end of the HP/IP turbine/generator shaft line. With today's technology, a single fluid drive/boiler feed pump set on the order of 36,000 to over 40,000 hp could be used instead of the two that were used then.

Today, with the availability of three-dimensional computational flow dynamic analyses for steam path design, it should be possible to build large electrical generating plants using similar arrangements and obtain net heat rates below 9000 btu/kW · h.

The primary advantage of the "shaft-driven" design is purely to maximize efficiency of the steam power. In the shaft-driven arrangement, the steam power is converted to mechanical power at approximately 92% efficiency. Near the maximum power point, the efficiency of a fluid drive is around 93% at full load, and it will exceed 70% for the bulk of the hp-hours of operation, though it will drop below that for very low-load operation.

For a motor-driven fluid drive, the chain is longer and less efficient: The mechanical power at the turbine coupling to the generator is 92% of the steam power (the same as

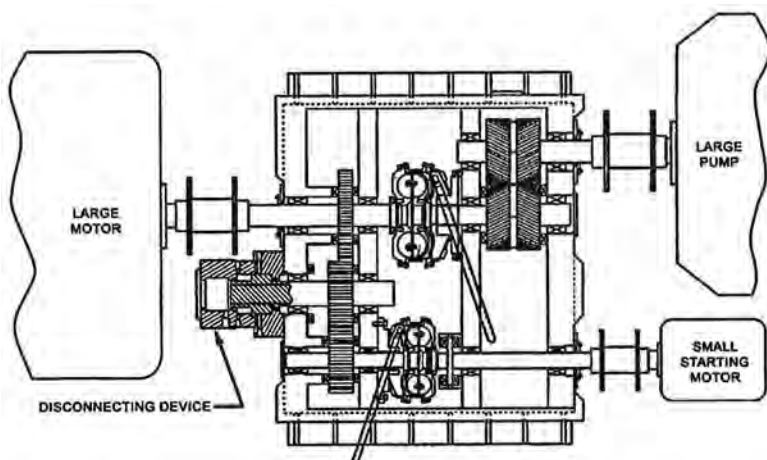


FIGURE 12 Variable speed geared fluid drive with parallel start-up capability that is used to start a very large motor which can then drive a very large pump (Turbo Research, Inc., U.S. Patent 7,171,870)

above), with additional losses of approximately 1.6% for the generator, 1% for the transformer, and perhaps 6% for the motor. The result is that for every megawatt needed to drive a fluid drive by a motor requires at least 9% more of the power from the turbine shaft. For a 1000 MW T-G that requires 36,000 hp from the turbine to drive a boiler feed pump via the "shaft-driven" feature would require 39,234 hp, or an additional 3234 hp to drive a motor-driven boiler feed pump. This is significant in heat rate calculations for a turbine-generator of this size.

VARIABLE SPEED FLUID DRIVE FOR LARGE SYNCHRONOUS MOTORS WITH A STARTING CAPABILITY

As previously mentioned, with today's technologies, it is possible to build very large 100% boiler feed pumps and comparably large fluid drives for turbine-generators on the order of 900 to 1300 MW. Figure 12 presents an example of a fluid drive design that can be used for the unique application of a very large synchronous motor driving a geared fluid drive/large boiler feed pump. This variable speed geared fluid drive has features (a) to be able to start a very large synchronous motor so that it can be connected to the grid smoothly without a large inrush current, (b) to separate the starting equipment from the gear train, and then (c) to use the large motor to drive a comparably large variable speed pump. This design permits the use of one large motor-driven pump to be used from start-up to full load operation. Alternative configurations of fluid drives can, of course, be adapted or developed to meet the needs of other applications.

9.2.4 GEARS

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USE OF GEARS WITH PUMP DRIVES

The main use of gearing in pump drives is to change the speed of the prime mover to the speed applicable to the pump. In some applications, such as a turbine to a reciprocating pump, the speed is reduced, so this gearbox is typically described as a speed reducer. In another application, such as a motor to a centrifugal pump, the speed is increased, and this gearbox is typically described as a speed increaser.

The second reason to use gears with pump drives is to arrange the shafts from horizontal to vertical. A vertical pump may be combined with a prime mover that must operate horizontally, such as a diesel engine. A right-angle gear set can be incorporated into the drive of such a combination to change the power train from horizontal to vertical. The right angle can be an increaser, reducer or even a 1:1 ratio. The right-angle gear set can also be incorporated into a housing with additional gear sets. Figure 1 is a pump drive gearbox of this style.

It is also possible to use a gearbox to combine the power from shafts of multiple prime movers. In the case where a redundant system is needed for reliability, it is possible to run a pump with an electric motor and to have a diesel engine also installed. However, a clutch would disconnect the diesel engine through a different input shaft. If the electrical power fails, then the diesel engine can be started and the clutch can be engaged to allow the pump to be back in service quickly.

Some applications, such as one driven by an electric motor, need to drive a pump at variable speeds. In this case, a coupling with a variable drive feature, such as a fluid drive coupling, can be mounted on the input side of the gearbox. This allows the prime mover to accelerate to synchronous speed, and then the fluid drive can be filled to increase the pump to any speed.

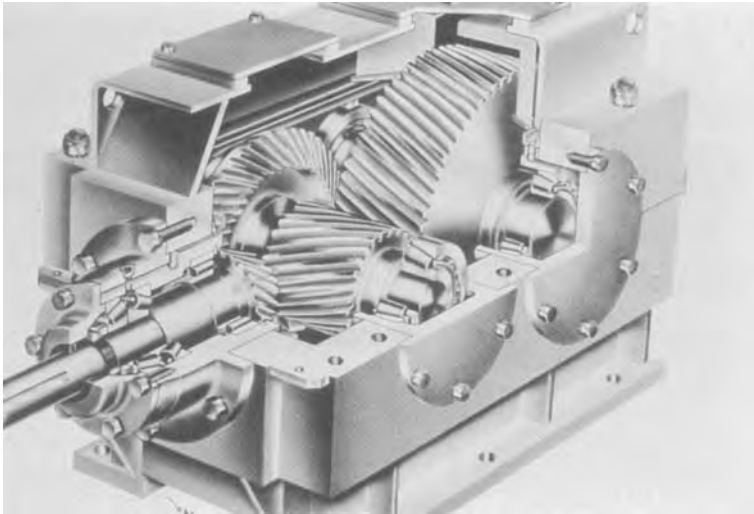


FIGURE 1 Typical pump drive gearbox (Lufkin Industries, Inc.)

HOW GEARS WORK

Gears are rotors that have teeth spaced uniformly around their circumference. When gears rotate around their center axis, the motion is equivalent to rolling contact between two cylinders or two cones. This contact surface is described as the pitch surface. Spur and helical gears have parallel axis and cylindrical pitch surfaces, while bevel gears have intersecting axis and conical pitch surfaces. The shapes of the gear teeth are mathematically defined, so they have a constant angular velocity when they roll through the mesh with another gear.

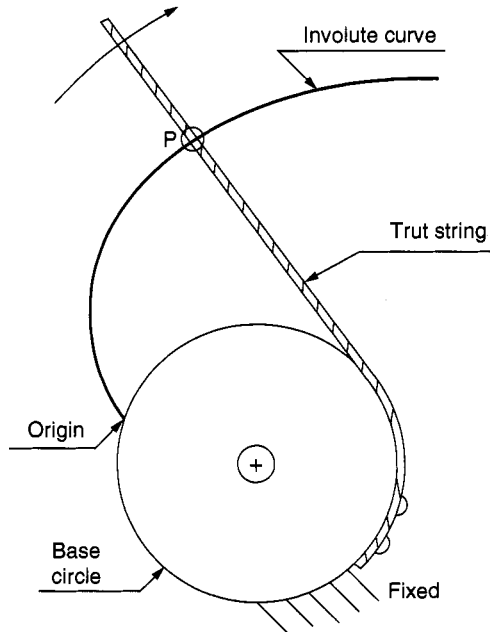
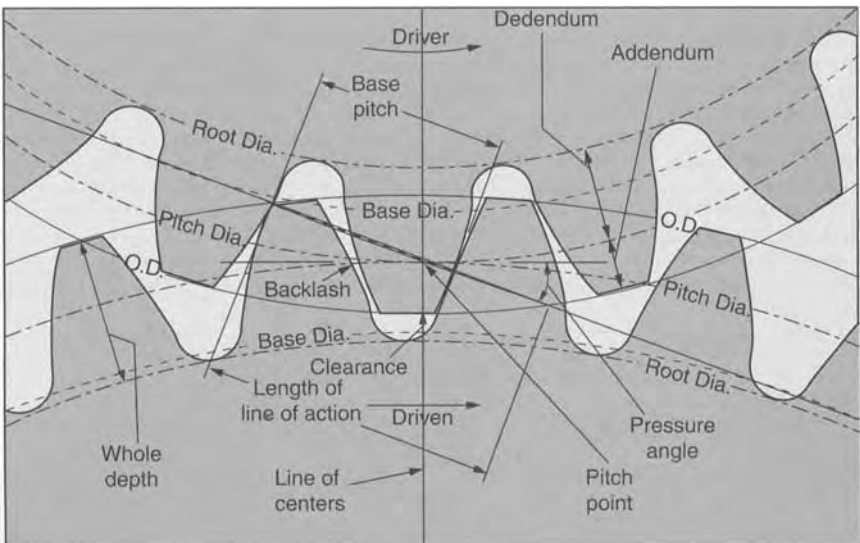
There are several different gear teeth shapes that have been developed over history that mathematically result in the transmission of constant angular velocity as they roll through mesh. By far, the most common of these shapes is the involute tooth form. This tooth form is simple to produce and small changes in the center distance result in very minor errors from constant angular velocity. Another tooth form that is used is the cycloidal tooth form. The cycloidal tooth form allows very high ratios, but it is limited to very low torques because of the severe undercutting of the teeth. They are commonly used for indexing applications such as in a clock. The involute tooth form is the only form that will be discussed further, because it is the dominant form used.

In a two-dimensional view sectioned through the gears in the transverse plane (perpendicular to the shaft centerline) the point of contact lies on a line called the line of action. The line of action for an involute tooth gear set is a straight line, whereas the line of action for a cycloidal tooth gear set is an S-shaped curve consisting of two circular arcs. In a three-dimensional view, the points of contact describe a surface defined as a zone of action. The shape of the points of contact is a plane for the involute tooth form.

The Involute The involute shape can be most easily described as the shape of the line that is produced by unwinding a string from a cylinder. See Figure 2 to better understand the principle.

It is important to understand how the involute shape can be applied to two different diameters of gears and yet the two gears roll together while transmitting smooth motion. The main requirement is that both sets of gear teeth have the same base pitch. Base pitch is defined as the distance between the same flanks of two adjacent teeth measured perpendicular to the involute surface. It is pictorially shown in Figure 3. While both the

Involute curve string method

**FIGURE 2** Generation of an involute (Lufkin Industries, Inc.)**FIGURE 3** Gear tooth nomenclature (AGMA)

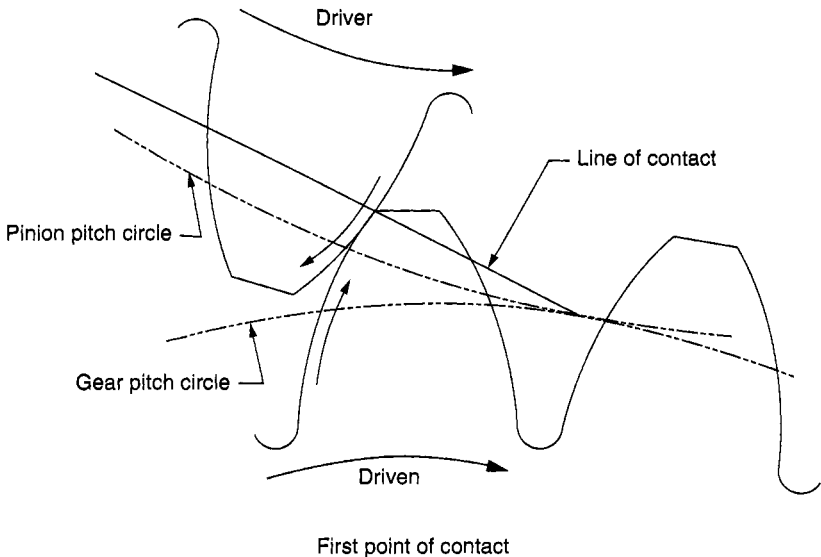


FIGURE 4 First point of contact (Lufkin Industries, Inc.)

large and small parts are generically called “gears” when they are separate, the small member is generally called a “pinion,” while the larger member remains known as the “gear” when they are put together in a gear set. By definition, the pinion will have a fewer number of teeth. Because a tooth of the pinion will predictably mesh with a tooth of the gear as the parts rotate, it can be understood how the gear will turn slower than the pinion as determined by the gear set ratio. The gear set ratio can be obtained by simply dividing the number of teeth in the gear by the number of teeth in the pinion.

Gear Tooth Stresses As the driving gear teeth push on the driven gear, two stress areas are of interest: the Hertzian contact stress in the contact area of the teeth and the resulting bending stress in the root of the tooth. The calculation of these stresses is difficult, so industry committees such as the American Gear Manufacturers Association (AGMA) and the International Organization for Standardization (ISO) continually update the standards defining the calculation methods to be used. If the contact stress is too high, the result is a gear set with distress on the working flanks. If the stress in the root of the tooth is too high, tooth breakage will occur. The levels of acceptable stress depend on the material that is used and the heat-treat process that is applied. Typical allowable stresses for various materials are discussed under “Gear Materials and Heat Treatment.”

As the gear teeth roll through the contact zone, the contact moves from the tip to the root of the tooth on the gear that is being driven and from the root to the tip on the gear that is driving. Visualizing Figure 4, it can be seen that the ratio of sliding to rolling is high at the extremes of the line of contact, and that when the contact point coincides with a line between the centers of the gears, the contact is pure rolling motion.

PARALLEL SHAFT GEARING

Gear sets with parallel shafts consist of three types: spur, single helical, and double helical. The centerline of the shafts may be either horizontal or vertical. Figure 5 is a typical parallel shaft pump drive gearbox.

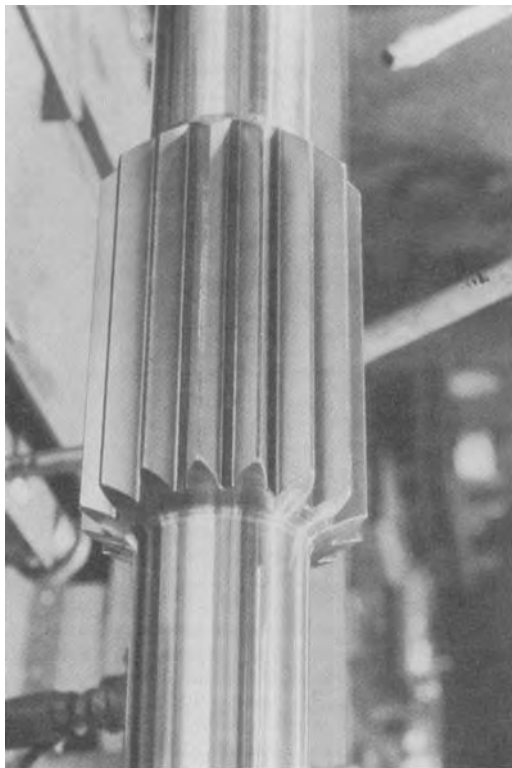


FIGURE 5 Spur gear (Lufkin Industries, Inc.)

Spur Gears Spur gears have the teeth parallel to the axis of rotation (refer to Figure 5). It is easy to see that the contact between the pinion and gear will be a line that extends the entire length of the tooth. The tooth that is loaded is bent out of true position, because the tooth loads are high. The bend of the tooth causes the next two teeth that come into mesh to be too far apart. This tends to cause an impact when the next tooth is loaded. It is also possible to visualize that only one tooth will be in contact a portion of the time, and then two will be in contact for the remaining portion of the time. The resulting change in the stiffness of the gear mesh, as well as any inaccuracy in the tooth form, cause vibration and noise. While theoretically giving smooth angular movement, these two features limit the use of spur gears to lower speeds and powers. Some advantages of the spur gear are that they are easy and inexpensive to make, they have no axial force from the gear mesh and they are relatively insensitive to axial location.

Single Helical Gears A more gradual contact between mating gears is obtained by slanting the teeth in a way to form helices that make an angle (see Figure 6). This type of gear is defined as a helical gear. The tooth contact between the teeth of mating helical gears is gradual; therefore, starting at one end and moving along the teeth so at any instant the line of contact runs diagonally across the teeth. The effect of the helix angle is to have tooth contact on several teeth at the same time; and the higher the helix angle, the more teeth are in contact. Instead of one tooth in contact in the case of a spur gear, it is possible to get six teeth in contact with a helical gear. When there are more teeth in contact, the change in torsional stiffness is minimized. Also, the more teeth that are in

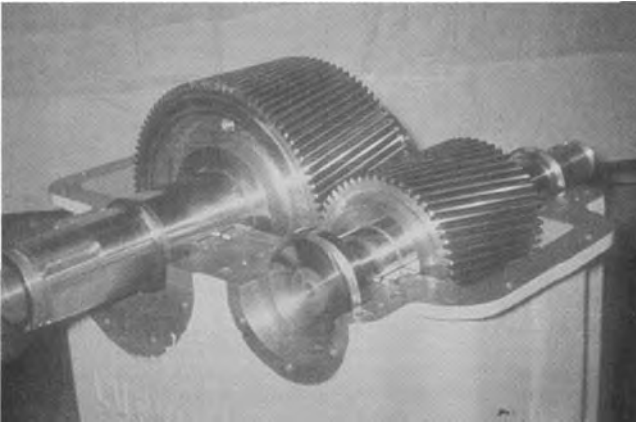


FIGURE 6 Single helical gearbox (Lufkin Industries, Inc.)

contact, the more the machining deviations of the teeth shape are averaged. The end result is that the helical gear can transmit more torque and run much smoother and quieter at higher speeds.

The helical gear generates an axial force when the gear set transmits torque. The axial force increases as the helix angle increases. Housings and bearings for helical gearing must be designed to accept these loads. The axial force creates an overturning moment on the gear that changes the magnitude and direction of the bearing loads (see Figure 7).

The bearings used on a single helical gear may be either a sleeve hydrodynamic bearing or a rolling element bearing. The determining factor is usually the velocity of the bearing journal. If the velocity is slow, the bearings are typically a single row tapered roller bearing mounted on either side of the gear teeth. If the application is faster, then the

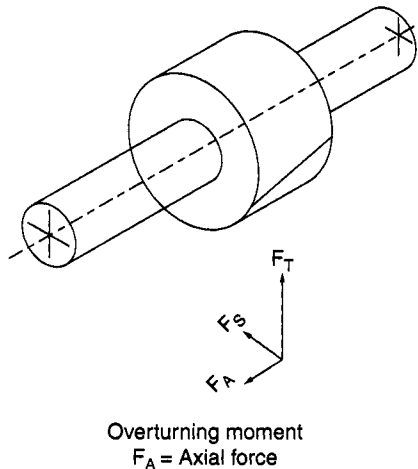


FIGURE 7 Single helical overturning moment (Lufkin Industries, Inc.)

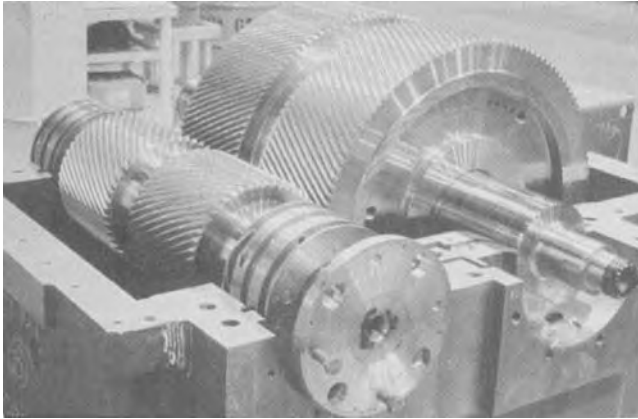
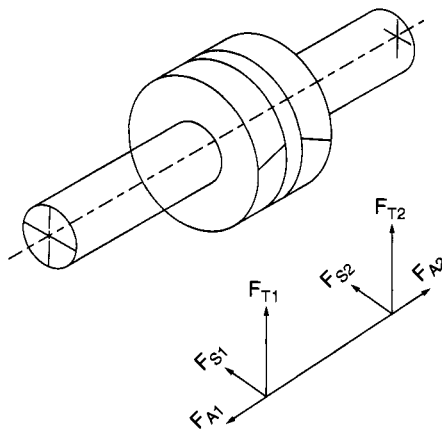


FIGURE 8 Double helical gearbox (Lufkin Industries, Inc.)

choice may be a sleeve hydrodynamic bearing with an integral flat face to absorb the thrust mounted on either side of the gear teeth. If the bearings have large internal clearances, as typical of a sleeve bearing, this will need to be taken into account when the gear tooth contact is checked.

Double Helical Gear A common type of gear will have two sets of opposing helical teeth (see Figure 8). Each set of teeth has the same helix angle and base pitch. This type of gear is defined as a double helical gear. The axial force generated when transmitting torque is equal and opposite, so that no thrust bearing is needed and the overturning moment is eliminated (see Figure 9). Higher helix angles can be used when the axial force is equalized, resulting in higher numbers of teeth in contact. This provides smoother operation.



Overturning moment = zero (0)

$$F_{A1} = F_{A2} = \text{Axial force}$$

FIGURE 9 Double helical overturning moment (Lufkin Industries, Inc.)

Axial Position The different requirement for thrust bearings between single and double helical gearing is quite interesting. In a double helical design, typically a thrust bearing would be used on the low-speed shaft (although some designs have no thrust bearing, and rely on limited end float couplings to keep the gear set assembly centered in the gear-box). If there is an axial force coming through the high-speed coupling, the axial loads between the two helices are unbalanced. The effect is to unload one helix and overload the other. The force coming through the high-speed coupling is usually because the axial coupling spacing is set incorrectly. It is important to remember that a double helical gear set will run in the center of the endplay, whereas a single helical will run up against the thrust bearing. Lack of centering the double helical pinion can be the cause of unbalancing the helix load sharing through axial misalignment. Thermal considerations often require that a disc or a diaphragm coupling be prestretched when the coupling is installed and the machinery is cold. If this prestretch is more than half the endplay of a double helical gear set, then the gear set is in the "bound mesh" condition until the machinery warms up. Under these circumstances vibration generally occurs on start-up. A good clue to see if this is the cause is to see if the frequencies of the vibration agree with the machinery train torsional critical frequencies.

Apex Wobble A phenomena of double helical gearing known as "apex wobble" has been recognized for a long time. Due to machining deviations in the gear teeth, and the load sharing between the helices discussed above, the pinion will move axially in relation to the gear as the gear set rotates. The acceptable magnitude of the movement for new gear sets is 0.0015 to 0.002 in (0.038 to 0.051 mm) depending on the speed of the gear set. With the precision manufacturing methods in general use today this is seldom a problem.

Similar to single helical gears, double helical gears can use sleeve or rolling element bearing, as determined by the velocity of the journals. However, for double helical gearing, it is imperative that either the pinion or the gear be allowed to move axially in relation to the other.

Herringbone Gear A special type of double helical gear that has a continuous tooth (see Figure 10) is known as a herringbone or Sykes gear. The helix angle is generally 30° because of the limited availability of the tooling used to manufacture this design. Smaller gear sets can also be used as an oil pump, because they do not have the gap between the teeth like the typical double helical gear set. The benefits of double helical gearing apply to the herringbone design, but there are obvious differences. If the gear rotates in the direction of rotation so that the contact lines move from the outer edges toward the center, the oil pushed ahead of the contact zone will be trapped in the point between the helices. This direction of rotation is described as "wings leading." The resulting high pressure can break the teeth in the center of the gear. For this reason, herringbone gears rotated with the wings leading are generally limited to 1000 ft/min (5 m/s) pitch line velocity (PLV). The velocity of the gear teeth is the most important factor in the selection, design, and manufacturing. The velocity is commonly calculated at the pitch diameter and expressed in feet per minute or meters per second. A user is generally not given the PLV, but it is easy to calculate. See Eq. 1 to calculate PLV and Eq. 2 to calculate the pitch diameter of the pinion:

In U.S. customary units:

$$PLV = PD \times 3.14 \times RPM/12 \quad (1)$$

$$PD = (2 \times CD)/(\text{ratio} + 1) \quad (2)$$

In SI units:

$$PLV = PD \times 3.14 \times RPM/60 \quad (3)$$

where

PLV = pitch line velocity in feet per minute (meter per second)

PD = pitch diameter of pinion in inches (meters)

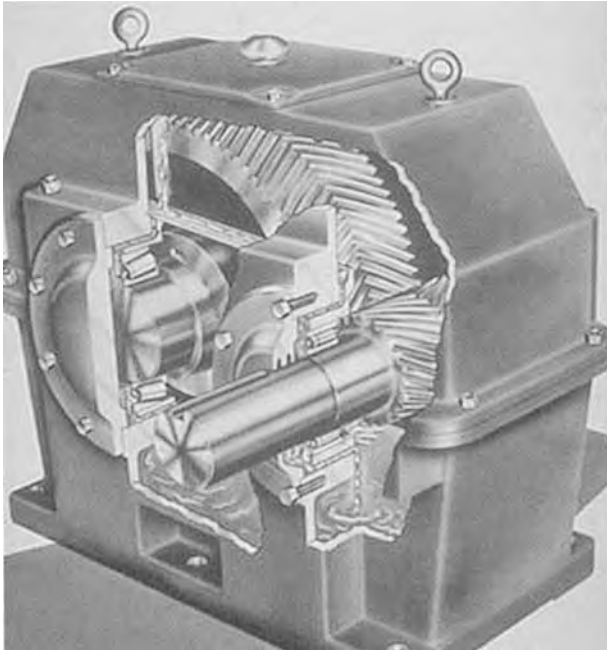


FIGURE 10 Herringbone gearbox (Lufkin Industries, Inc.)

Ratio = revolutions per minute of the pinion divided by revolutions per minute of the gear or number of teeth in gear divided by number of teeth in pinion

CD = center distance of the gear set in inches (meters)

RPM = revolutions per minute

For example, an application consisting of an electric motor at 1200 rpm driving a speed increaser to a centrifugal pump at 4000 rpm has 12 in between the shafts.

In U.S. customary units:

$$PD = (2 \times 12) / [(4000/1200) + 1] \\ = 5.538 \text{ inch}$$

$$PLV = 5.538 \times 3.14 \times 4000/12 \\ = 5796 \text{ ft/min}$$

The manufacturing process for herringbone gears is less sophisticated than typical double helical gears. The process results in much less accurate gear teeth. This has resulted in fewer herringbone gears being produced, because of the general increase in the power and speed of new machinery and user expectations for quiet and smooth operation.

Due to the speed limitations on herringbone gears, rolling element bearings are generally used. Like the double helical gears, either the pinion or the gear has to be free to move axially so that both helixes are loaded equally. Usually the pinion is mounted on cylindrical roller bearings and the gear is mounted with a tapered roller bearing on each side. This allows the pinion to move back and forth axially and maintain equal helix loading.

RIGHT-ANGLE GEARING

Gears of this type transmit power between two shafts. The shafts are usually at right angle to each other, but they can be at angles other than 90° . There are several special designs, but those most commonly used as pump drives are straight bevel, spiral bevel, and Zerol spiral bevels.

Straight Bevel Gears Straight bevel gears (see Figure 11) are the simplest form of bevel gears. They have teeth that are straight and tapered and would intersect at a common point at the axis if they were extended. These gears operate at speeds up to 1000 ft/min (5 m/s) PLV and are more economical than the other styles of right-angle gearing. They can be an increaser, reducer or a 1:1 reduction (often called “miter gears”). The axial force generated from the tooth loading is away from the other gear in either direction of rotation.

Spiral Bevel Gears Spiral bevel gears (see Figure 11) have curved teeth that have an inherently smoother meshing action than straight bevel gears. Whereas the contact moves from the top of the tooth to the root on a driven straight bevel gear, the contact also moves axially on a spiral bevel gear. This makes the spiral bevel meshing action much smoother and allows speeds of 8000 ft/min (40 m/s) PLV. The spiral direction or “hand” is difficult to describe. If you stand at the toe (the small end) of the pinion, and look toward the heel (the large end) the spiral is called a left-hand spiral angle if the teeth curve to the left, or a right-hand spiral angle if they go to the right. For example, the spiral bevel pinion shown in Figure 12 is a left-hand spiral angle. Spiral gears can run in either direction, but the axial force generated from the tooth loading will change direction. The hand of the spiral angle should be selected so the pinion tends to move out of mesh to avoid the possibility of running without backlash.

The bearing requirements for bevel gears are somewhat different than for the other types of gears, because the gear tooth contact changes with a slight change in axial position. Both the pinion and gear have to be restrained both in the radial and also the axial direction, instead of just the radial direction. The most common method is to use two single-row tapered rolling element bearings on each gear. Two other methods used are a two-row tapered rolling element bearing and a cylindrical bearing on each gear, or a dual angular ball bearing and a cylindrical bearing. Generally, the pinion is set with some preload on the bearings to reduce the possible axial movement of the pinion. A few high-speed bevel gear sets use sleeve bearings, but care has to be taken to reduce the axial movement.

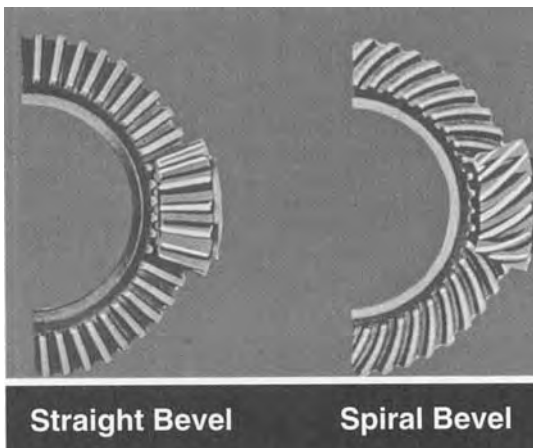


FIGURE 11 Straight and spiral bevel gear (AGMA)

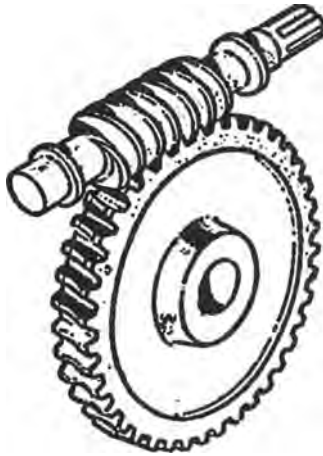


FIGURE 12 Worm gear and pinion (AGMA)

Zerol Spiral Bevel Zerol bevel gears have curved teeth like a spiral bevel, but the spiral angle is zero at the axial center of the tooth. Zerol bevel gears have a portion of the smoothness of a spiral bevel, because of the curved tooth. In addition, the spiral angle is zero, so the axial force generated from the tooth loading will not pull the pinion into the gear. It is common to upgrade a straight bevel gear set with a Zerol bevel gear set.

WORM GEARING

A worm gear set consists of a pinion, commonly called a worm, and a gear (see Figure 12). The pinion resembles a large screw thread. The gear resembles a narrow single helical gear. The gear may have an hourglass shape on the outside diameter to give line contact instead of point contact. This is described as a single enveloping gear set. The pinion may also have an hourglass shape on the outside diameter to give area contact instead of line contact. This is described as a double enveloping gear set. It is easy to see that the axial location of the pinion on a double enveloping gear set is critical to good contact. The gear will advance one tooth for every rotation of the pinion (assuming a single start thread) so the ratio is simply the number of teeth in the gear.

Worm gears typically utilize rolling element bearings. A common combination is to have a tapered roller bearing on either side of both the gear and the pinion. An alternate method is to use angular contact ball bearings on the pinion.

The advantages of the worm gear are the compactness of the gear set and the very high ratios. The disadvantage is the high sliding velocity resulting in limitations to PLVs (generally below 2000 ft/min - 10 m/s), as well as very poor efficiencies.

GEAR MATERIALS AND HEAT TREATMENT

Two of the most important factors in the successful operation of a gear set are the choice of material and heat treatment. While gears are built from material ranging from plastic to steel, pump drives generally are made from steel. The trend is to use the hardest and most durable heat-treating process available. Steels that have been quenched to form martensite and then tempered to restore a level of ductility offer the best combination of properties attainable. The size of piece that can be successfully quenched to produce

martensite depends on the amount of additional alloys, such as chromium, nickel, and molybdenum. Adding these elements move the “nose” of the time temperature transformation (TTT) curve to the right. This gives more time for the quench to be performed, but also could permit larger pieces to be successfully quenched.

The AISI numbering system for the basic steel types used for pump gears are as follows (the nominal percent of carbon is replaced with XX, e.g., 0.45% carbon is 45):

- 10XX Carbon steel
- 15XX Manganese 1.00 to 1.35%
- 33XX Ni 3.5%, Cr 1.5%
- 41XX Cr 1%, Mo 0.25%
- 43XX Ni 1.75%, Cr 0.75%, Mo 0.25%
- 86XX Ni 0.5%, Cr 0.5%, Mo 0.2%
- 93XX Ni 3.25%, Cr 1.25, Mo 0.12%

The methods used to heat treat the steel are through hardening, nitriding, induction hardening, carburizing, and flame hardening.

Through Hardening Through-hardened steels have a reasonably consistent hardness all the way through the gear. This allows the heat treating to be done before the gear teeth are machined. Suitable steels for through-hardened pump gears are the 41XX and 43XX series, with carbon content around 0.4%. The 10XX and 15XX series cannot be hardened deep enough to be used. Following are general guidelines for obtaining 321 BHN on various diameters of gears. This is a typical hardness for through-hardened gearing and a direct conversion from a nominal indentation diameter on a Brinell checker.

- | | | |
|--------------|-------|----------|
| • 4140 | 5 in | (127 mm) |
| • 4145 | 10 in | (254 mm) |
| • 4340 | 24 in | (610 mm) |
| • 4340 (mod) | 36 in | (194 mm) |

Nitriding Nitriding is a heat-treating method utilizing nitrogen, which is produced by cracking ammonia to diffuse into the surface and form hard nitride particles. Aluminum containing alloys are particularly good for nitriding since they already form aluminum nitride needles. Nitriding is a low-temperature diffusion process performed in the range of 900 to 975°F (482 to 524°C). The case depth is limited to typically 0.025 to 0.030 in (0.63 to 76 mm) deep since the diffusion rate at the modest temperatures is low. However, the relatively low temperature gives very low part distortion, but still results in a case with high hardness. The hardness is up to 45 R_c for 41XX and 43XX steels, and 64 R_c for special nitriding steels. After nitriding, some type of finishing operation is normally required to remove what is known as a “hard layer” or “white layer” from the flanks of the teeth.

Induction Hardening Induction hardening is a heat-treating process consisting of using high-frequency alternating current to quickly heat the tooth surface of the gear, then immediately quenching the part. There are two methods of induction hardening commonly used, and the selection primarily depends on the size of the tooth and the size of the gear. The first method, used for smaller gears and smaller teeth, consists of rotating the gear inside a coil and energizing the coil with high-frequency electric current. The current causes the tooth area to be heated; then this area is immediately quenched by submerging or spraying the gear. The second method, described as a tooth-to-tooth or scanning method, is used for large gears. It consists of a shaped electrical inductor passing through the tooth gap (see Figure 13) while it is energized with high-frequency alternating current. The tooth is immediately quenched, generally by a spray nozzle integrally built into the inductor head. The resulting hardness on the 41XX and 43XX series steels range from 50 to 58 R_c . The desired hardness depth depends on the tooth size, but a hardness depth

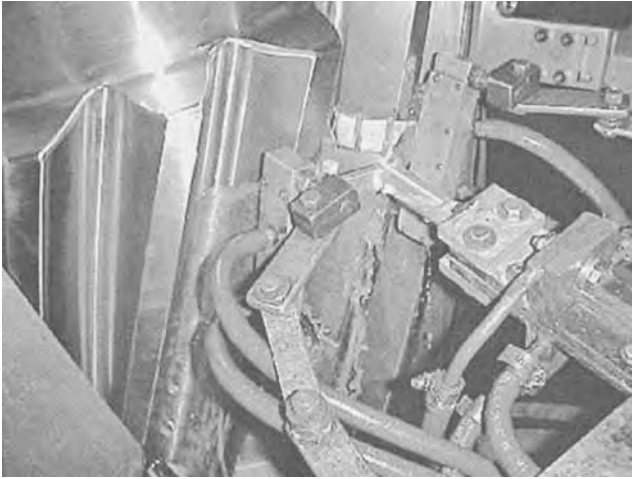


FIGURE 13 Induction hardening (Lufkin Industries, Inc.)

of 0.125 in (3 mm) is easily achievable. Tooth-to-tooth induction hardening is an economical and effective method to surface harden larger gears, but it can result in cracking if the process is not carefully controlled.

Carburizing The carburizing heat-treat process produces the highest allowable stresses per the ratings standards, and it is probably the most common method of manufacturing pump drives. High alloy, low carbon steels, such as 4320 and 9310, are heated to 1750°F (954°C) in a carbon rich furnace (see Figure 14).



FIGURE 14 Carburizing (Lufkin Industries, Inc.)

The carbon slowly diffuses into the surface and then the part is immediately quenched. The final material is a composite of a hard surface consisting of approximately 0.85% carbon and a softer core with the original 0.1% carbon. The difference in the carbon content is very important. An increase in carbon content from 0.1% to 0.85% reduces the solidification temperature by over 500°F (260°C). During the quench, the center transforms first to a relatively soft martensite because of the low carbon content. Minutes later, the case transforms to martensite, but the volume increase happening during this transformation is resisted by the center, which forces the case into a high compressive stress. In addition to the high hardness due to the higher carbon, this resultant compressive stress allows the gear teeth to operate successfully under even higher loads. The hardness of the carburized surface after quench is usually around 64 R_c . To gain more ductility, this is usually tempered at 350°F (177°C) resulting in the final hardness of about 60 R_c . The martensite transformation rarely goes to 100% completion and some residual austenite is usually present. Typically in gear applications the level of the residual austenite is restricted to less than 20%.

Carburized gears have the highest allowable load rating of any material; however, the distortion during the heat-treat cycle is severe and complicates the manufacturing process. The distortion requires substantial experience in carburizing and some type of finishing operation.

Flame Hardening In tooth-to-tooth progressive flame hardening, an oxyacetylene flame is applied to the flanks of the gear teeth. After the surface has been heated to the proper temperature, it is air or water quenched. This method has some limitations because the hardened case does not extend into the root of the tooth. The allowable Hertzian stress is higher, but the stresses built up at the interface may actually decrease the allowable root bending stress.

Gear Tooth Stresses The gear stresses for the calculation of the power rating of a gear set are defined by the rating standards, such as those provided by AGMA. The allowable Hertzian contact stress per AGMA for various materials is shown below:

Allowable Contact Stress Number (S_{ac}) psi

• Cast iron	65000
• Ductile iron	128000
• Through hardened	150000
• Induction hardened	190000
• Nitrided	163000
• Carburized	225000

The allowable root bending stress per AGMA for the same materials is shown below:

Allowable Bending Stress Numbers (S_{at}) psi

• Cast iron	8500
• Ductile iron	33500
• Through hardened	50000
• Induction hardened	55000
• Nitrided	52000
• Carburized	70000

GEAR TOOTH CONTACT

Gears have an unusual requirement that is not necessary for other types of machinery. The operating location of the shaft centerlines must be maintained very accurately with

respect to each other. This allows the gear teeth to have an even load across the entire face width. If the shafts are not correctly located, then part of the gear teeth will be very heavily loaded and fail will be imminent. Tooth alignment is controlled by the accuracy of the gear teeth, the housing, and the bearings. Twisting of the housing due to poor foundation conditions, incorrectly installed bearings, or poorly manufactured components will cause poor tooth contact.

Tooth contact should be checked on all new installations, after any disassembly of the gearbox, and after any major housing movements on the foundation.

The contact can be checked using two different methods. The methods are known as either the "soft" blue or the "hard" blue method. Soft machinist's blue can be lightly applied to a few teeth on the pinion, and then the pinion rolled by hand through mesh with the gear. The transfer of the blue from the pinion to the gear is referred to as the contact pattern. The hard blue method is performed by spraying hard blue or layout blue on the teeth of the pinion, then operating the gearbox and evaluating the wear pattern from the hard blue applied to the pinion.

The soft blue method is usually performed when the cover is removed from the housing, such as during the assembly or during the installation of a new set of gears or bearings.

Soft Blue First, clean the teeth of any debris or residual oil using a suitable cleaner such as electrical contact cleaner. Apply soft blue to a 1-in (25 mm) wide good quality paintbrush that has had the bristles shortened to a length of about 1 in (25 mm). Using the brush, spread the soft bluing all the way across three or four teeth of the pinion. The bluing will be too thick, so use tissue paper to smooth the bluing to a very thin and even layer. For double helical gear sets, center the pinion teeth axially on the gear teeth. This can be achieved by either using a dial indicator to measure the axial travel of the pinion in relation to the gear and then moving the pinion back halfway; or turning the pinion while allowing it to move axially with a slight drag on the gear. For single helical gear sets, position the pinion and gear against the proper thrust bearings. Using an assistant, apply a slight resistance to the turning of the gear by pushing down symmetrically on the teeth or bearing journals. Roll the blued area in and out of mesh, taking care to be smooth and consistent. Observe the blue that is transferred from the pinion to the gear. This is the contact pattern.

It is much easier to translate the soft blue pattern if a "lift tape" of the contact pattern is taken. Carefully place a piece of cellophane tape lengthwise on one of the gear teeth that the bluing was transferred to, press it down firmly, then pull it off by holding it firmly on both ends. Place the tape on a clean sheet of white paper and label it appropriately. At this point, the blue transfer is much easier to translate and also a permanent record has been taken.

Hard Blue The hard blue is similar to paint in that it dries after a short period of time. The cleanliness of the teeth before applying the hard blue is even more important than the soft blue. Be cautious to spray the blue lightly on the pinion. After the blue has dried, operate the gearbox, preferably at a light load for a short period or until its temperature stabilizes, then stop the gearbox and observe the wear pattern.

For parallel shafts, the guidelines in Figure 15 will be useful in determining whether or not the contact is acceptable. As a general rule, 80% minimum contact length is required. If you are interpreting hard blue contact, be aware that the higher the power that was transmitted during the hard blue test, the higher percent of tooth contact will be required.

LUBRICATING OIL SELECTION AND TESTING

The gear teeth will be the focus of this paragraph on lubrication. It is important to fully realize that many other components, such as bearings, will be using the same oil and may compromise the selection.

It is the intent that the oil will be selected so that it will separate the gear teeth to avoid gear tooth distress. The primary factors that will affect the oil selection are gear type, gear set PLV, gear tooth loading intensity, anticipated oil operating temperature, and gear tooth surface finish.

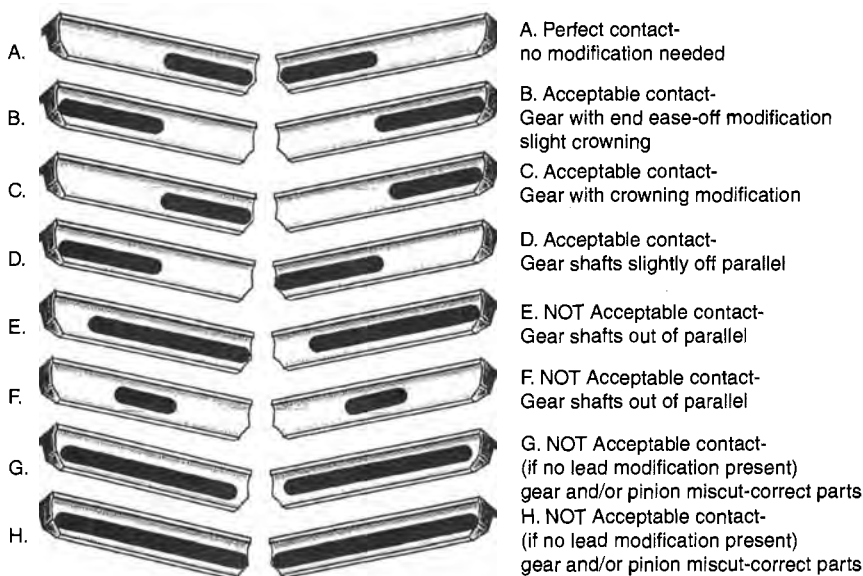


FIGURE 15 Various double helical gear set contacts (Lufkin Industries, Inc.)

Viscosity Selection It is generally accepted that the two most important oil characteristics that control the success of gear tooth lubrication are the viscosity of the lubricant and the additive package. Most people are familiar with the ISO VG series of oil viscosities, for example: ISO VG 220. The ISO series uses the middle of the viscosity range in centistokes as the generic designation. Fewer people are familiar with the AGMA 9005 Industrial Lubrication Standard¹ viscosity system, for example: AGMA 5, which also describes the oil viscosity. In recent years, the ISO numbering method is predominately used. A general guideline that has often been used to select a lubricant for a specific application is given in Eq. 4:

$$\text{Viscosity} = 7000/(\sqrt{\text{PLV}}) \quad (4)$$

Where viscosity = viscosity of oil in centistokes

For the example used in the calculation of PLV, a 4000-rpm pump driven by a 1200 rpm motor:

$$\begin{aligned} \text{Viscosity} &= 7000/(\sqrt{5796}) \\ &= 92 \text{ centistokes} \end{aligned}$$

A reasonable selection would be an ISO VG 100 lubricating oil. Analyzing this equation helps to understand the dependence of the viscosity on the PLV. As the PLV lowers, the oil needs to be much more viscous. Other methods to determine the oil viscosity are to use the tables in the AGMA 9005 Standard, or the method for calculating the elastohydrodynamic film thickness (EHL) using information from sources like the *Mobil EHL Handbook*.²

The additive package is much more important for lower PLV gears. For example, gear sets operating above 6000 PLV generally use a mineral oil with only rust and oxidation inhibitors added. These oils are generically labeled "R & O" oils. Lower speed gears generally use a mineral base lubricant with chemical additives. These additives, such as sulfur phosphorus or similar materials, produce a protective film to provide antiwear properties. Oils with the above mentioned additives are generically labeled EP (extreme pressure) oils.

There are exceptions to these guidelines. For instance, in a worm gear, the high sliding properties require special consideration. A "compounded gear oil" is usually recommended. Compounded gear oil is a combination of the mineral base lubricant with the addition of 3 to 10% fatty oils. Another exception is when using an EP oil with gearboxes that have clutches or backstops that share the same oil. Unfortunately, the same additives that help keep the gear teeth apart may allow the clutches and backstops to slip.

While the traditional petroleum lubricants remain more popular, the use of synthetic lubricants in gearboxes has gained acceptance in recent years. It is in general agreement that for extreme weather conditions, synthetics can be an advantage. Some synthetic lubricants claim that they substantially increase the efficiency of a gearbox. This is recognized in worm gears, but it is a controversial subject in parallel shaft and bevel gears.

There are two basic ways to lubricate a gearbox. One way is to use internal oil passageways, mesh sprays, and an oil pump to circulate the oil. The alternative method is a system consisting of dipping the gears and centrifugally throwing the oil into strategically placed troughs that lead to the bearings. Obviously, the centrifugal system is less expensive and very reliable and is often used in roller bearing applications. On higher speed gearboxes, such as above 4000 PLV, the turbulence is high, the foaming may become excessive and the efficiency may be degraded. In this situation, it is more attractive to use a pressurized system and avoid dipping the gears in the oil.

Oil Maintenance It is very typical for a gearbox manufacturer to request that the oil be changed every 4 months. With the industry trend toward more reliability and longer-run times between shutdowns, the requirements of the manufacturer and user may be viewed as being in conflict. It is probable that the manufacturer is concerned about particles that become suspended as wear occurs during the early life of a gearbox. This will occur especially in gears with an initial rough surface finish and that are relatively soft. These abrasive particles need to be removed so that polishing can occur. In response to the need for longer intervals between oil changes, oil analysis is a very helpful method to determine when the gearbox oil needs to be changed. The spectrographic analysis focuses on the various elements in parts per million, and the report is accompanied by suggestions from the analyzing vendor. The cleanliness analysis rates the amount of debris in the oil by the size of the particles and uses a code to indicate the number in a prescribed volume. A cleanliness rating requirement of 19/18/14 per ISO 4406³ is generally recognized as adequate for pump drive gearboxes.

GEARBOX VIBRATION

The tooth mesh frequency (TMF) is the most important aspect of gearbox vibration. As discussed in the "Parallel Shaft Gearing" section, every time a tooth engages a mating tooth, impact occurs. By multiplying the number of teeth in the gear by the speed of the gear, the TMF can be determined. A vibration analyst should know this frequency so that he can properly evaluate the gear set. A typical centrifugal pump drive could easily have a TMF as high as 4 kHz. At these high frequencies an accelerometer is very helpful.

The combination of the number of teeth in the two meshing gears is important. If the gear set has no common prime numbers except unity, then is it referred to as a "hunting tooth" gear set. Gear designers try to use hunting tooth combinations to avoid the vibration frequencies that result from not doing so. These frequencies are known as "assembly phase" and "tooth repeat" frequencies and are thoroughly discussed in *Component Identification of Gear-Generated Spectra* by John Winterton.⁴

Torsional/Lateral Correlation In addition to the axial force on a double helical pinion exciting the torsional criticals as previously discussed, there are other unique gearbox vibration phenomenon. One of the most interesting is the correlation between torsional and lateral vibrations. To understand this problem, it is essential that we understand the bearing loading orientation in a typical gearbox. From the discussion on "How Gears Work," the loading from the gear teeth will be along the line of action. The line of action

is approximately 20° off of a line perpendicular to a line connecting the centers. For a parallel shaft gear set with the shaft centerlines horizontal, the gear tooth forces will be about 20° off of vertical in the upward direction for one gear, and 20° off of vertical downward for the other (see Figure 3). If there is a power pulse coming through the equipment train, the force on the gear teeth will see this pulse as alternating tooth loading. This fluctuation changes the bearing load, which then changes the shaft location in the bearing. The resulting lateral shaft vibration is the result of the torsional vibration.

Up versus Down Load Another unusual gearbox vibration problem can be aggravated by the orientation of the gearbox, or what is known as the "hand" of the gearbox. As we look into the coupling end of the pinion, imagine that we could have the gear either on the right side or the left side of the pinion. Which side we select is important to the anticipated vibration of the gearbox. If the pinion is rotating clockwise (still looking into the coupling end) and it is assumed the application is a speed reducer, and the gear is on the right side, the pinion will be loaded up and the gear will be loaded down. This gearbox will be a better design than if the gear had been on the left because the uploaded element will move upward at a lower torque. As an example, an uploaded pinion may move at 5% torque, but an uploaded gear at 40% torque. In summary, select the orientation of the shafts so that the heavier member is loaded down, and the lighter member is loaded up. See Figure 16 to see the difference in a typical pump drive application.

A typical bearing used on a high-speed gearbox is a hydrodynamic sleeve bearing. The uploaded shaft usually has a stability feature such as a pressure dam. There is an increased chance of assembling a pressure dam bearing on the uploaded shaft incorrectly. The pressure dam should be located in the bottom of the bearing and help push the shaft up into the operating position. If it is either in the wrong half, or turned backward, vibration and failure will be imminent. Figure 17 gives the load direction and the bearing orientation for both the gear and the pinion for the different possible configurations.

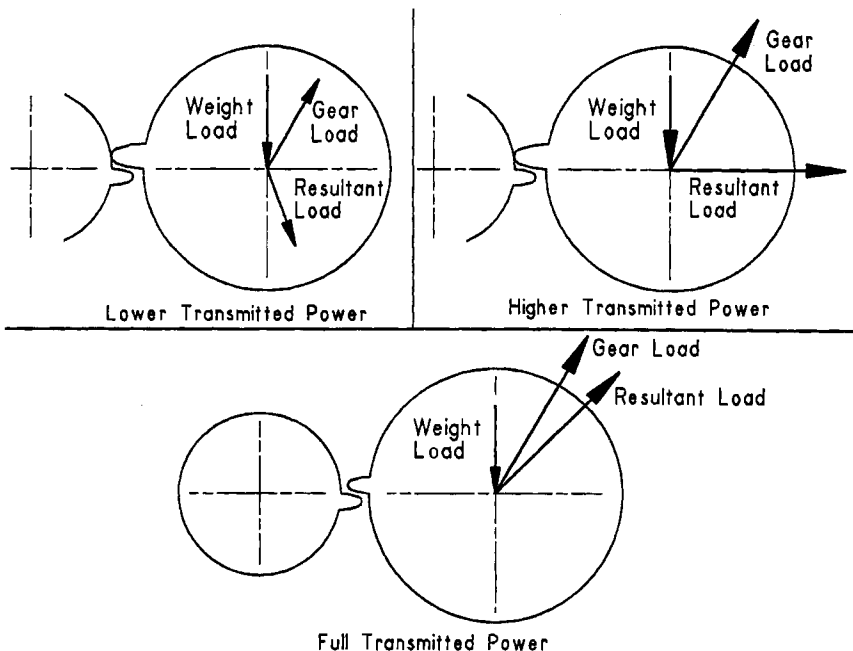


FIGURE 18 Uploaded versus downloaded gear (Lufkin Industries, Inc.)

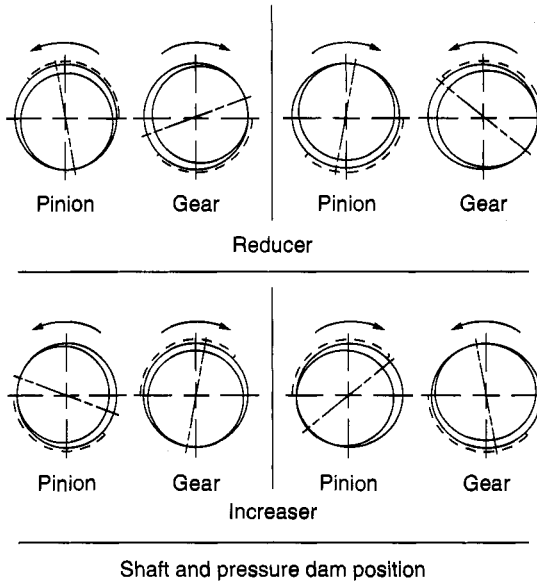


FIGURE 17 Pressure dam bearing orientation (Lufkin Industries, Inc.)

TYPICAL GEAR TOOTH FAILURES

Gear tooth failures have been thoroughly studied and documented. It is probably more helpful to steer the reader toward these documents, than to give a few samples that could lead the reader into an incorrect solution.

The most commonly used document is the ANSI/AGMA Standard 1010, *Appearance of Gear Teeth-Terminology of Wear and Failure*.⁵ Figure 18 is a typical failure picture from the standard.

There is also a document, ISO 19825, *Gears-Wear and Damage to Gear Teeth*⁶ that is considered very useful.



FIGURE 18 Removing a gear band (Lufkin Industries, Inc.)

GEAR REPAIR

The purpose of this paragraph is to explain the gear set and gearbox repairs that are different than repairs of other types of machinery. Of course, this will include gear tooth repair, but will also include comments on housing and bearing repair.

It is suggested that some type of nondestructive test be performed on the gears to give some basis for a decision to determine if it is repairable. The optimum inspection is a wet magnetic particle inspection, but the circumstance may only allow a dry magnetic particle or liquid penetrant inspection.

Three major methods to repair gear teeth are recutting the teeth, regrinding, and rebanding.

Recutting Recutting is a popular method to repair gears, however, it only applies to through-hardened gears. This process consists of machining some material from the outside diameter of the gear, then using a hob or shaper (with the same cutting tool as was originally used) to sink the teeth further into the part by the same amount as was previously removed. Then a new pinion is manufactured that is larger than the original pinion by the same amount that was machined off the gear. The pinion teeth are cut to a full depth, but not as deep as the original pinion.

The recutting process generates a slightly different involute shape than the original. The new gear involute is virtually identical. The teeth on the new pinion are slightly thicker at the root than on the original, which means that the bending stress is lessened. The pinion involute is usually the limiting rating factor, so the rating on most gear sets is actually improved by this process. A new gear is expensive and has a long delivery. By salvaging the gear, both the cost and the downtime are substantially less than a new gear set.

Regrinding Regrinding the tooth is often an alternative to recutting, and it can be used to repair both through-hardened and surface-hardened gears. It cannot be used on herringbone gears because they lack a groove between the helices that is necessary for tool clearance during the regrinding. The regrinding process consists of moving the identical involute shape back along the circumference of the pitch diameter until the distressed gear tooth material has been removed. It is a general practice to regrind the active flank of both the gear and pinion a minimal amount. Of course, the repaired gear set now has more backlash than the original gear set. If the gear set is surface hardened, then some thought needs to be given to the remaining case depth. For instance, regrinding a nitrided gear set is not commonly done, because the original case depth is quite small (and not known by the user) and the softer core material may be exposed to higher stresses than it can accommodate. In a typical regrind, 0.003 in (0.08 mm) of material is removed from the loaded flank of the pinion and the same amount is removed from the gear. Using this process, the gear and pinion is salvaged, so the cost and downtime are less than both a new gear set and a recut gear set. In recent years, this process has become very common and has become known as "kiss grinding."

Rebanding A "banded" gear is a two-piece gear described as a gear with a center section, generally made of cast iron, and a much thinner hardened alloy steel band mounted around the center section. The band is retained in place by an interference fit and the gear teeth are machined into the band after the two parts are assembled. Rebanding is the process of machining the alloy steel band from the center section of a two-piece gear and replacing it with a new band. It is easy to remove the band by machining a narrow groove across the band and nearly through to the outside diameter of the center section, causing the band to fracture (see Figure 18).

Rerimming Rerimming is often used to describe a similar situation as rebanding, except the gear has an alloy steel rim welded on an assembly consisting of side plates instead of an alloy steel band shrunk on a cast iron center. The removal and reassembly necessary to rerim is more difficult and less attractive than rebanding. Neither rebanding nor rerimming is common, because the cost and downtime are similar to a new gearset.

Housing Repair Gearbox housings have much different requirements than housings for other machines. The number of load cycles is extremely high (a steam turbine driving a gearbox would typically have 10 giga cycles on the pinion teeth during its life) and the stresses are also high. Therefore, the need for shafts to be held parallel (or perpendicular for a right-angle gear) to minimize uneven tooth stress is absolutely critical.

The inspection process for a gearbox should include measuring the bearing bores for roundness, size, and parallelism. A common rule of thumb is to allow a maximum of 0.003 in (0.08 mm) skew between the centerlines of the bores in a plane perpendicular to a plane through both gear axis and a maximum of 0.005-in (0.13 mm) difference between the center distances. If the housing needs to be repaired, remilling the split line to create stock in the bores and then reboring the box is the usual method of repair. If the design uses a carrier that fits between the housing bore and the outside diameter of the bearing, the repair is much simpler because a new carrier can be made. A nice improvement to an old gearbox is to have the carrier bore offset from its outside diameter so the carrier can be turned to adjust the gear set parallelism as based on the soft blue information.

Bearing Repair When repairing a sleeve type hydrodynamic bearing, take care to avoid affecting the parallelism of the gears because of machining the bore nonconcentric with the outside diameter. This will dramatically affect the gear tooth contact. It is possible to measure the wall thickness in the loaded zone of the repaired bearing, and to evaluate the amount of error one is introducing into the gear tooth contact. Unfortunately, this is not often done, and the problem with the repaired bearing is noticed during an emergency when there are very few options available.

SUMMATION

It was assumed that the reader of this chapter would either be operating a pump driven by a gearbox and would be interested in achieving the maximum life possible; or may have gearbox problems and be looking for answers. With this assumption in mind, the structure of the chapter was intended to recreate the life of a typical gearbox. It started with gear tooth theory, discussed the types of gears, the material options, their assembly and lubrication, typical vibrations, the failure modes, and then typical methods of repair.

Gear specialists are quick to agree that gearboxes are more complex and technically developed than most users would imagine. Large amounts of reference material are available if a deeper knowledge of gearboxes is desired. Some of the favorites are listed in the "Further Reading" section below.

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1. ANSI/AGMA 9005-E02, *Industrial Gear Lubrication*, Dec. 2002. American Gear Manufacturers Association, 500 Montgomery Street, Suite 350 Alexandria, Virginia 22314.
2. *Mobil EHL Guidebook*, Fourth Edition, copyright 1992. Mobil Oil Corporation, 3225 Gallows Road, Fairfax, Virginia.
3. International Organization for Standardization, ISO 4406, Particle Count Analysis.
4. John G. Winterton, "Component Identification of Gear-Generated Spectra," Bently Nevada Orbit Journal, (June, 1991) pp. 11-14.
5. ANSI/AGMA 1010-E95, *Appearance of Gear Teeth-Terminology of Wear and Failure*, Dec. 1995. American Gear Manufacturers Association, 500 Montgomery Street, Suite 350 Alexandria, Virginia 22314.
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FURTHER READING _____

ANSI/AGMA 6013-A06 "Standard for Industrial Enclosed Gear Drives." This standard includes design, rating, lubrication, testing, for spur, helical, bevel, and worm gears in various configurations. American Gear Manufacturers Association, 500 Montgomery Street, Suite 350 Alexandria, Virginia 22314. 2006. ISBN: 1-55589-822-X.

ANSI/AGMA 6011-I03 "Specification for High Speed Gear Units." This standard includes design, rating, lubrication, testing, for helical high-speed gears. 2003. ISBN: 1-55589-819-X.

Dudley, Darle W. *Handbook of Practical Gear Design*. McGraw-Hill, New York, 1984. ISBN: 0-07-017951-4.

MAAG, *MAAG Gear Book*. MAAG Gear Company Ltd., Zurich, Switzerland, 1990.

SECTION 9.3

PUMP COUPLINGS

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COUPLING TYPES USED IN PUMP DRIVE SYSTEMS

A coupling is used wherever there is a need to connect a prime mover to a piece of driven machinery. The principal purpose of a coupling is to transmit rotary motion and torque from one piece of equipment to another. Couplings may perform other secondary functions, such as accommodating misalignment between shafts, compensating for axial shaft movement, and helping to isolate vibration, heat, and electrical eddy currents from one shaft to another.

Rigid Couplings Rigid couplings are used to connect machines where it is desired to maintain shafts in precise alignment. They are also used where the rotor of one machine is used to support and position the other rotor in a drive train. Because a rigid coupling cannot accommodate misalignment between shafts, precise alignment of machinery is necessary when one is used.

TYPES There are two commonly used types of rigid couplings. One type consists of two flanged rigid members, each mounted on one of the connected shafts (Figure 1). The flanges are provided with a number of bolt holes for the purpose of connecting the two half-couplings. Through proper design and installation of the coupling, it is possible to transmit the torque load entirely through friction from one flange to the other, which assures that the flange bolts do not experience a shearing stress. This type of arrangement is especially desirable for driving systems where torque oscillations occur, as it avoids subjecting the flange bolts to a shearing stress.

A second type of rigid coupling, known as the *split rigid*, is split along its horizontal centerline (Figure 2). The two halves are clamped together by a series of bolts arranged axially along the coupling. The rigid coupling and machine shafts may be equipped with conventional keyways, which are in turn fitted with keys to transmit the torque load, or in

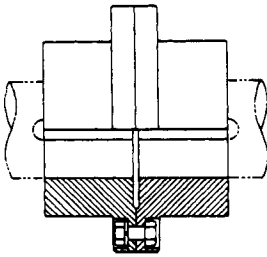


FIGURE 1 Flanged rigid coupling (Kop-Flex EPT)



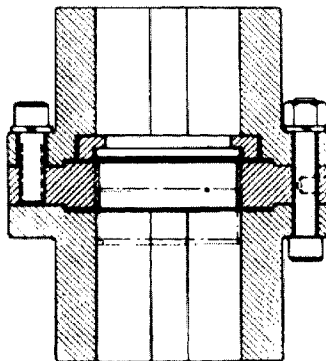
FIGURE 2 Split rigid coupling (Dodge Manufacturing Div. Reliance Electric)

certain cases the frictional clamping force may be sufficient to permit transmitting the torque by means of friction between shaft and rigid coupling. This type of coupling is commonly used to connect sections of line shafting in a drive train.

A variation to the flanged rigid coupling is known as the adjustable rigid coupling (Figure 3). This coupling is designed along the lines of conventional rigid couplings, except that a threaded adjusting ring is placed between the two flanges. This ring engages a threaded extension on one of the shaft ends. By means of this ring, it is possible to position the pump shaft axially with respect to the driver.

APPLICATIONS A common application for rigid couplings in the pump industry is in vertical drives, where the prime mover (generally an electric motor) is positioned above the pump. In such cases, both machines can employ a common thrust bearing, which is generally located in the motor. The coupling flange bolts must be capable of transmitting any down thrust from the pump to the motor. In applications where the thrust from the pump is toward the motor, it is common practice to provide shoulders on the shafts to transmit the axial force.

Many pump drive systems require a rigid coupling that is capable of providing axial adjustment to compensate for wear in the pump impeller or impellers. The adjustable rigid coupling is used for this purpose. The threaded adjusting ring attached to a mating threaded extension of the pump shaft permits vertical positioning of the impeller or impellers. The hub, which is mounted on the pump shaft, is equipped with a clearance fit and feathered key that permit the hub to slide with respect to the shaft. The load capacity



of a coupling of this type is generally limited by the pressure on the pump shaft key because there is no possibility of load being transmitted by interference fit.

A few words of caution should be noted about the use of rigid couplings. First, precise alignment of machine bearings is absolutely necessary because there is no flexibility in the coupling to accommodate misalignment between shafts. Second, accuracy of manufacture is extremely important. The coupling surfaces that interface between driving and driven shafts must be manufactured with high degrees of concentricity and squareness, to avoid the transmittal of eccentric motion from one machine to another.

Flexible Couplings Flexible couplings accomplish the primary purpose of transmitting a driving torque between prime mover and driven machine. In addition, they perform a second important function: they accommodate unavoidable misalignment between shafts. A proliferation of designs exists for flexible couplings, which may be classified into two types: mechanically flexible and materially flexible. Except for some elastomer types, couplings must have two flexible planes to accommodate radial misalignment. The distance between the flexible planes or elements determines the radial (parallel) misalignment capabilities. The individual flexible elements provide angular misalignment. Single flex plane couplings are used in some configurations, as are rigid couplings.

MECHANICALLY FLEXIBLE COUPLINGS Mechanically flexible couplings compensate for misalignment between two connected shafts by means of clearances incorporated in the design of the coupling. The most commonly used type of mechanically flexible coupling is the gear, or dental, coupling (Figure 4). This coupling essentially consists of two pairs of crowned spur gear teeth. In this coupling configuration, the two machine shafts are equipped with hub members having teeth cut integrally on the hubs. The two hubs are connected by a sleeve member having mating internal gear teeth. A backlash and a crown are intentionally built into the gear teeth connection, and it is this configuration that compensates for shaft misalignment. Sliding motion occurs in a coupling of this type, and so a supply of clean lubricant (grease or oil, depending on the design) is necessary to reduce wear of the rubbing surfaces. Straight tooth, loose spline, couplings are seldom used today but were often used on pumps in the past.

If operation cannot be interrupted to lubricate the couplings, constantly lubricated couplings are used, as shown in Figure 5. These were most often used on high-speed services with the oil lubricant coming from the bearing oil circulation system, as it was clean and cool. It utilizes an oil-tight enclosure bolted to the driving and driven piece of equipment. Oil under pressure is brought through the enclosure and impinges upon the meshing gear teeth of the coupling, the excess being collected at the bottom of the enclosure and returned to the oil reservoir.

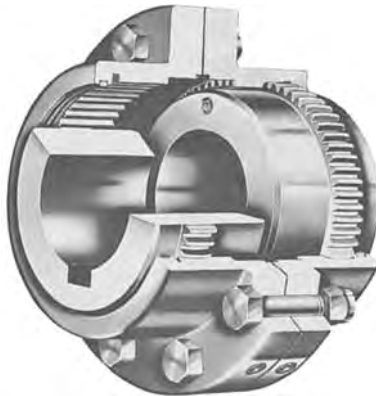


FIGURE 4 Gear-type mechanically flexible coupling (Lovejoy Inc)

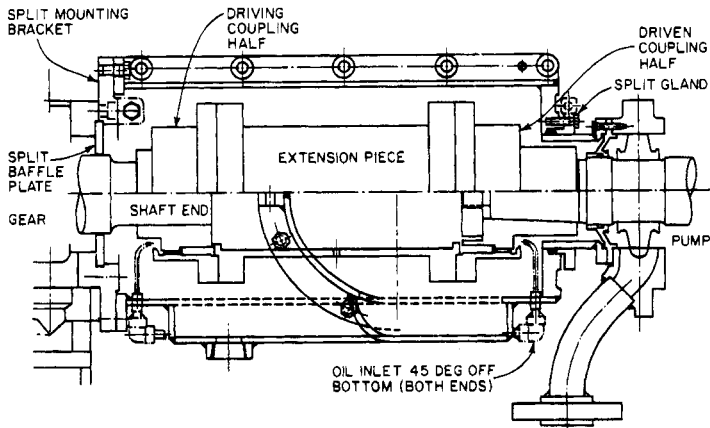


FIGURE 5 Continuously lubricated coupling (Kop-Flex EPT)

A second type of mechanically flexible coupling that is used in low-cost drive systems, is known as the roller-chain flexible coupling (Figure 6). This coupling employs two sprocket-like members mounted one on each of the two machine shafts and connected by an annulus of roller chain. The clearance between sprocket and roller and, in some cases, the crowning of the rollers provide mechanical flexibility for misalignment. This type of coupling is generally limited to low-speed machinery as it wears quickly, but it is very inexpensive.

SPRING-GRID COUPLING There is a type of commercially available flexible coupling that combines the characteristics of mechanically flexible and material-flexible couplings. This is the spring-grid coupling (Figure 7). This design has two hubs, one mounted on each machine shaft. Each hub has a raised portion on which tooth-like slots are cut. A spring steel grid member is fitted, or woven, between the slots on the two hubs. The grid element can slide in the slots to accommodate shaft misalignment and flexes like a leaf spring to transmit torque from one machine to the other. Unlike most material-flexible couplings, this design requires periodic lubrication to prevent excessive wear of the grid member.



FIGURE 6 Roller-chain mechanically flexible coupling (Dodge Manufacturing Div. Reliance Electric)

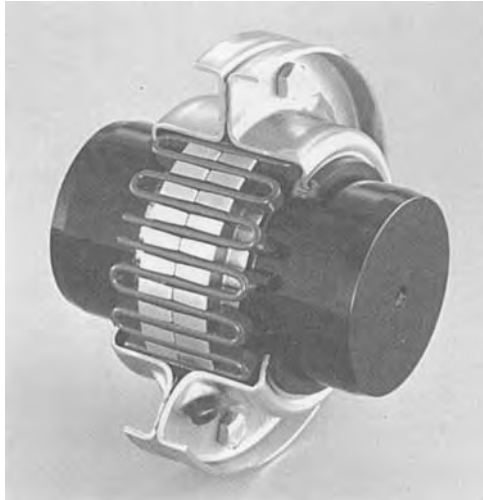


FIGURE 7 Spring-grid coupling (Lovejoy Inc)

MATERIAL-FLEXIBLE COUPLINGS These couplings rely on flexing of the coupling element to compensate for shaft misalignment. The flexing element may be of any suitable material (metal, elastomer, or plastic) that has sufficient resistance to fatigue failure to provide acceptable life. Some materials, such as steel, have a finite fatigue limit. A coupling made of such material must be operated under conditions of load and misalignment that assure that the stress developed in the coupling element is within that fatigue limit. Other materials, such as elastomers, generally do not have a well-defined fatigue limit. In these cases, however, heat developed in the material when the coupling flexes can cause failure if excessive.

Metallic flexible element couplings are tension-loaded or shear-loaded for torque transmission and bend for misalignment accommodation. Metal flexible element couplings have two flex planes separated by a spacer section. The spacer section can be either a drop-out type to facilitate use with back pull-out pumps or an integral-type. Drop-out spacer types have more parts so are more difficult to balance, if necessary, and are more costly.

A popular type of material-flexible coupling is the metal-disc coupling (Figure 8). This coupling consists of two sets of thin sheet-metal discs bolted to the driving and driven hub members. Each set of discs is made up of a number of thin laminations that are individually flexible and compensate for shaft misalignment by means of this flexibility. The discs are stacked together as required to obtain the desired torque transmission capability. This type of coupling requires no lubrication; however, alignment of the equipment must be maintained within acceptable limits so as not to exceed the fatigue limit of the material. Metal-disc couplings are replacing gear couplings as the most popular all metal couplings because they have good power density and do not need lubrication. When properly selected and installed, they have an extremely long life. Disc couplings are often configured with a drop out center section for ease of maintenance.

Another example of an all-metal material-flexible coupling is the flexible diaphragm coupling, shown in Figure 9. This coupling is similar in function to the metal-disc coupling in that the disc flexes to accommodate misalignment. However, the diaphragm type is loaded in shear. It can be a single element with a hyperbolic contour that is designed to produce uniform stress in the member from inner to outer diameter. By more efficiently utilizing the material, the weight is reduced correspondingly, thus making this coupling suitable for high-speed applications. This coupling is used both on high performance and high-speed applications.



FIGURE 8 Spacer metal disk coupling (Lovejoy, Inc)

Elastomer flexible element couplings use a material that has, by definition, a high degree of elasticity and resiliency and will return to its original shape after undergoing large-amplitude deformations. A version of this coupling for medium-speed applications, such as pumps, uses urethane as the diaphragm material (Figure 10). This version of the coupling would be torsional compliant and suitable for back pull out pumps.

Material-flexible couplings employing elastomer materials are numerous and their designs are varied. Elastomers are loaded only in compression or shear. Compression-loaded couplings are stiffer in torsion than shear-loaded elastomers and can carry more load while shear-loaded elastomers are more compliant in torsion and may be needed to torsional tune some pump systems. Rubber elastomers and synthetic elastomers such as urethanes, or nylons are used. Each has its own load carrying capability, resiliency, and hysteresis capability. Often shear-loaded elastomer couplings are used for their fusible link capability while compression-loaded elastomers are used in fail-safe couplings. Elastomer couplings are also used because they are quiet.



FIGURE 9 Diaphragm material-flexible coupling (Fluid Power Division, Goodrich)



FIGURE 10 Elastomer type diaphragm coupling (Kop-flex EPT)

A type of elastomer coupling that is commonly used on low-power drive systems is the rubber jaw coupling (Figures 11a, 11b, 11c). The heart of this coupling is a “spider” member, having three or five or more segments extending in a radial direction from a central section. The hubs, which are mounted on driving and driven shafts, each have a set of jaws corresponding to the number of spiders on the flexible element. The spider fits between the two sets of jaws and provides a flexible “cushion” between them. This cushion transmits the torque load as well as compensating for misalignment. Jaws can be of the straight-sided or curve-sided type. Both types load the elastomer in compression. The curved-jaw type more fully encompasses the elastomer and restrains it for higher torque capability. It is not as resilient and may not be as easy to maintain as the straight-sided jaw in some configurations. A variation of the jaw coupling uses a wrap around “spider” that bridges a wider gap from hub to hub. In this case, the elastomer is shear loaded and has the advantage of easy elastomer replacement when worn. It is a fusible link, not a fail-safe variation.

Another group of elastomer couplings employs a sleeve-like element that is connected to a hub member on each shaft and transmits torque through shearing of the flexible

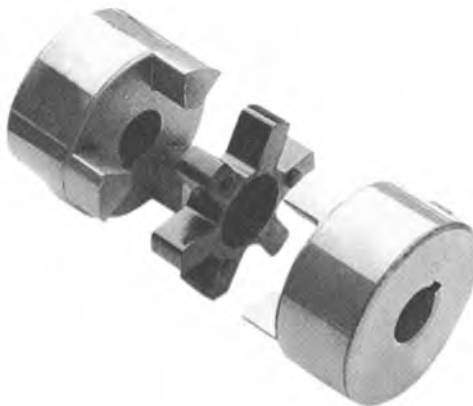


FIGURE 11A Jaw Coupling straight (Lovejoy Inc)

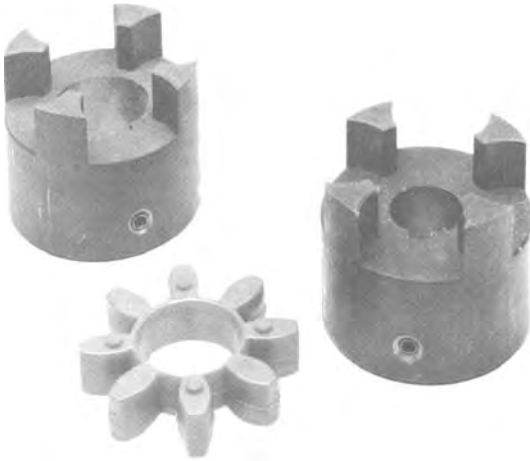


FIGURE 11B Jaw Coupling curved (Lovejoy Inc)

element. The flexible element may be attached to the machine hubs by a number of different means: it may be chemically bonded, it may be mechanically connected by means of loose fitting splines (Figure 12), or it may be clamped to the hubs and held in place by friction (Figure 13).

A third group of couplings utilizes an elastomer member that is loaded in compression to transmit load from one shaft to the other (Figure 14). The elastomer material is placed loosely into cavities formed by members rigidly mounted onto the two shafts or the elastomer



FIGURE 11C Jaw Coupling elastomer in shear (Lovejoy Inc)



FIGURE 12 Sleeve-type elastomer coupling (T. B. Woods)

can be a donut-shaped device attached to rigid hubs on either shaft (Figure 15). Again, the elastomer material deflects to compensate for shaft misalignment, and is usually used when torsional damping is required. The shear and compression type of torsional couplings are often used for connecting centrifugal pumps directly to diesel engines.

Another example of a compression-type elastomer coupling is the pin-and-bushing coupling (Figure 16). This design comprises two flanged hub members, one mounted on each machine shaft. The flange of one hub is fitted with pins that extend axially toward the adjacent hub flange. The other flange is equipped with rubber bushings, which generally have a metal sleeve at the center. The pins fit into these sleeves and provide transmission of torque through the bushings. Because the bushings are made of flexible material, they can accept slight angularity or offset conditions between the two flanges. This coupling has limited misalignment capability as it depends on the distortion of the elastomer in a single plane.

PERMANENT MAGNET COUPLINGS These couplings rely on two mechanically independent coupling components to compensate for shaft misalignment. The design does not depend on a material flexible element or a mechanically flexible connection between driver and load shafts; instead, shaft misalignment is accommodated by an air gap that separates both shafts (Figures 17 and 18). The mechanically independent feature enables the permanent magnet coupling to handle parallel offset, angular misalignment, shock loads, and thermal expansion without the transfer of forces/moments and vibration between equipment shafts.

The permanent magnet coupling is a constant torque device that operates on the slip principle. The device must slip in order to transmit torque; hence the difference in power is called slip loss, and is in the form of generated heat. There are also windage and frictional losses, but they are small enough to be neglected. Common values of slip for a centrifugal pump or fan application ranges from 1 to 4%. Slip heat is dissipated into the surrounding air by convection.

The basic permanent magnet coupling is factory assembled and balanced. They may be installed indoor or outdoor without special provisions and can be installed and maintained by millwrights. They mount directly to the driver and load shafts using keyed or shrink-disc coupling hubs. The permanent magnet coupling is nonenergized, therefore, it is not vulnerable to process shutdown due to voltage sags and power fluctuations. Further, they have no wear parts, eliminating the need for lubrication and hazardous waste disposal.

A permanent magnet coupling has limited axial end float capability—a result of the air gap between rotors. For shaft mount configurations, both pieces of machinery must have bearings that limit total axial end float. For an assembled unit, electromagnetic interference is not a factor and is less than that typically found in an electric motor. However, for

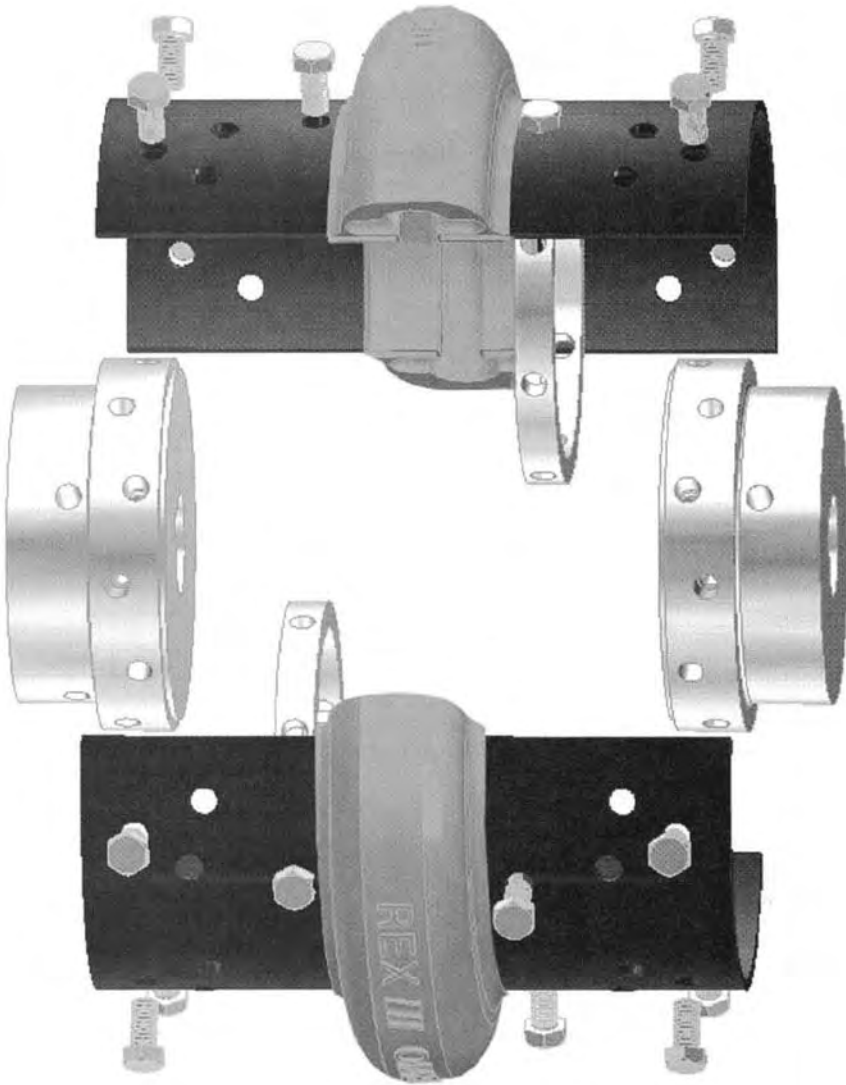


FIGURE 13 Spacer elastomer coupling (Falk Rexnord Couplings)

disassembled units, individual magnets and exposed magnet rotors require strict safety precautions and handling procedures.

A second type of permanent magnet coupling provides equipment protection in the event of an over torque occurrence, such as a load seizure. The torque-limiting feature employs a declutching mechanism that disconnects the rotors and reduces the transmitted torque to about 5% of the couplings-rated torque. If the output torque exceeds the coupling breakdown torque, the load shaft will decelerate, causing the coupling to disengage (increase the air gap). Once disengaged, the driver spins freely unloaded. The torque-limiting

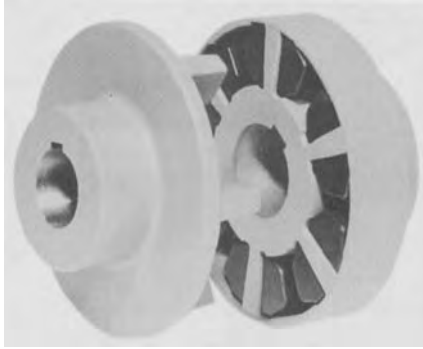


FIGURE 14 Compression loaded loose fitted elastomer coupling (Lovejoy Inc)

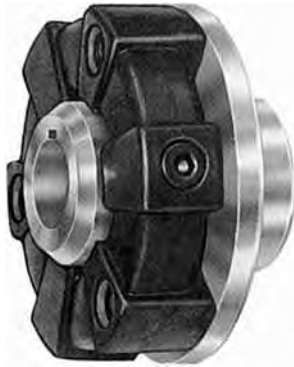


FIGURE 15 Compression loaded donut style elastomer coupling (Lovejoy Inc)



FIGURE 16 Pin-and-bushing elastomer coupling (Ajax Flexible Coupling)

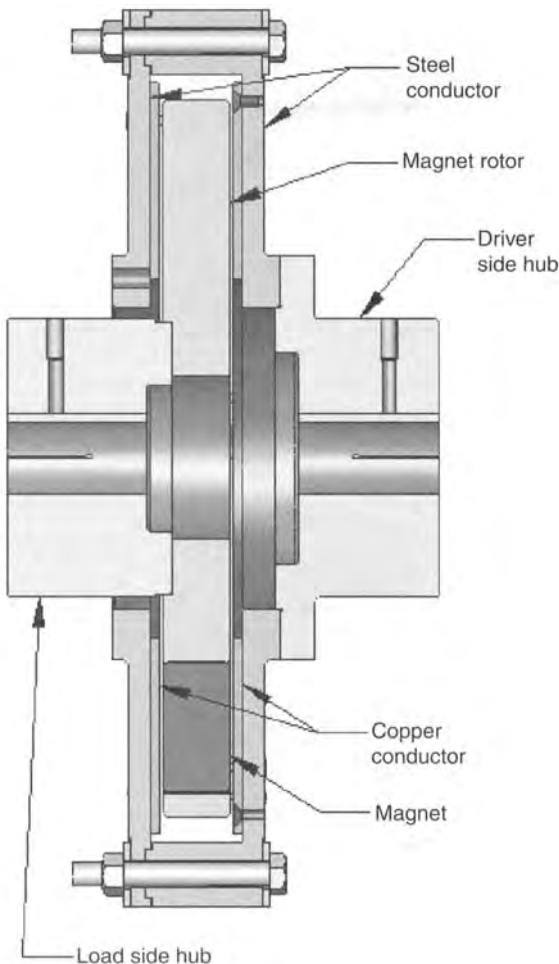


FIGURE 17 Cutaway of a shaft-mount, air-cooled permanent magnet coupling (MagnaDrive Corp)

coupling will remain in this configuration until motor speed and load speed are synchronized, typically by turning off the motor, whereby the coupling will automatically reset, ready for restarting.

APPLICATIONS The type of flexible coupling most suitable for a particular application depends upon a number of factors, including power, speed of rotation, shaft separation, amount of misalignment, quiet operation, torsional tuning, cost, and reliability. In the design of a system, it is the goal of the designer to use the least expensive coupling that will do the job. In low-cost systems, cost alone may be the most important criterion, and the least expensive coupling that transmits the rated power and accepts some small degree of misalignment is generally the choice, albeit at some sacrifice of reliability and durability. On the other hand, high-power, high-speed machinery generally represents a critical piece of equipment for a power station, sewage plant, or other vital process, and in these cases a coupling should be selected that will not compromise the overall reliability of the system.

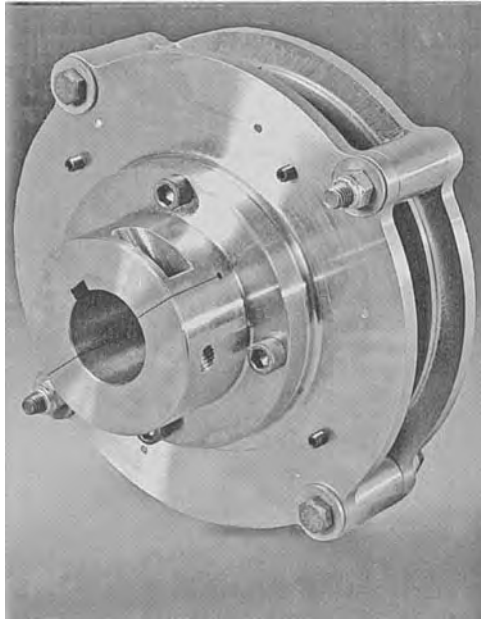


FIGURE 18 Key driven permanent magnet coupling (MagnaDrive Corp.)

Low-power pumps (up to about 200 hp, 150 kW) driven by electric motors can usually be coupled successfully by any of the couplings described here. Selection procedures vary from manufacturer to manufacturer, but generally the following data are required: power rating, speed, anticipated misalignment, type of pump (reciprocating, vane, centrifugal, and shaft diameter of the driver and driven side of the coupling).

Pumps of the same power range are very often driven by reciprocating engines (diesel, gasoline, natural gas). This is quite common in remote areas, such as at pipeline pumping stations, or agricultural watering systems (Figure 19), where a source of electric power is not available or is expensive. Because reciprocating engines of the diesel type produce a cyclic torsional vibration, it is often necessary to perform a torsional vibration analysis of the drive system to ensure that the normal operating speed is well removed from a speed that may produce a torsional resonance. Such an analysis requires that the torsional stiffness of the coupling be known. It is quite often possible to tune the drive system to avoid operating at a resonant condition by selecting the proper coupling stiffness. The selection data required for a system of this type include rotating inertias and connecting shaft stiffness. Most coupling manufacturers will, however, assign a higher service factor to an application involving a reciprocating prime mover, to compensate for fatigue effects due to torque fluctuations, particularly if a torsional analysis is not completed.

If a diesel engine is used, a torsional analysis should be performed, as oversize couplings may be so stiff as to raise the torsional resonance right into the operating or idle speed. In addition, the remote location of many engine-driven pumps indicates a special need to ensure a high degree of reliability of the system.

Another commonly employed prime mover is a steam turbine. These machines, which range from about 100 hp (75 kW) to well over 50,000 hp (37,000 kW) for pump drives, operate very efficiently and economically, providing there is a source of steam available at the installation. Any flexible coupling employed on a machine driven by a steam turbine must be capable of accepting the thermal gradient at the turbine shaft and must also accommodate the axial growths of the turbine shaft as it warms up to operating speed.

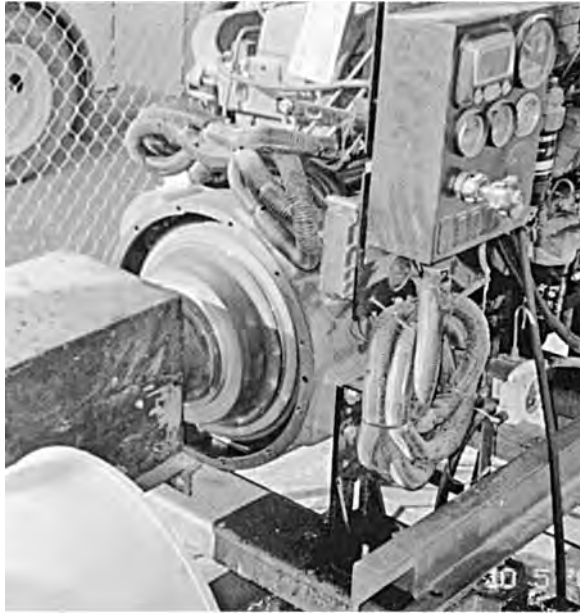


FIGURE 19 Flywheel coupling for vertical pump (Lovejoy Inc)

Steam turbines are generally high-speed machines (4,000 to as high as 10,000 to 15,000 rpm in some cases) and as such require a relatively high degree of system balance to avoid critical vibration. Gas turbines could also be the prime mover, but they are seldom found on pumps drives. Elastomer couplings have occasionally been applied successfully to steam turbine drives, but because of the high speeds involved, all-metal couplings are most commonly employed. Disc or diaphragm coupling designs that are light, flexible, and do not require lubrication are the most popular for this application. High-speed machinery requires that the weight of rotating components be minimized to decrease shaft deflections and hence increase the lateral critical speed of the system. The gear coupling is an efficient design for transmitting large amounts of power at high speeds and with minimum weight and with axial slide capability, but the requirement for a lubrication system makes the gear coupling less desirable, and it is no longer the most common choice.

Where coupling weight and torsional stiffness are critical values to the overall system, special designs may be created that provide the specific values required for satisfactory system operation. Special designs can be disc, diaphragm or, in some cases, gear couplings.

NONSPARKING AND GALVANIC CURRENT Some applications require a coupling that will not cause a spark or a coupling that will isolate a galvanic current. Usually these couplings use elastomeric flexing elements and require that both sides of the coupling be grounded. Coupling manufacturers have their own design features to make the couplings conform. Most nonsparking couplings have shorter maintenance review cycles and elastomer replacement cycles to assure the elastomer never wears out and allows a spark from metal to metal contact. The couplings could be fail safe or fusible-link type. Coupling manufacturers can offer ex-couplings for ATEX applications.

BALANCE For higher speed, higher power applications, coupling balance (or potential unbalance) is an important factor to consider. Elastomeric couplings may have a considerable amount of potential unbalance because of their construction, and they don't lend themselves to balancing. The amount of inherent unbalance in metal couplings is controlled

largely by the level of precision to which they are manufactured. When pumps operate at speeds exceeding four-pole motor speeds and low equipment vibration levels are critical to service life, elastomeric couplings may not be a good choice. Such requirements are important to certain pump types, like those required for API Standard 610.¹ For more information on coupling manufacturing and balance classes and requirements, the reader is referred to the API and AGMA standards listed at the end of this section.

LIMITED END FLOAT Many horizontal motor-driven pump systems utilize motors that are equipped with journal, or sleeve-type, bearings. These bearings are intended only to absorb the transient thrust created by the motor rotor during acceleration and deceleration. The coupling for this type of drive should be equipped with suitable provisions for limiting the axial float of the motor rotor to some fraction of its total float. This may be done by positioning the motor in the center of its axial travel and then employing a coupling having a total float that is less than the float of the motor. Any motor thrust is taken by the pump bearing. Gear coupling total float can be limited by inserting a button between shaft ends, as shown in Figure 20. This type of coupling prevents the motor rotor from ever contacting the thrust shoulders on the shaft bearings. Certain types of elastomer and disc couplings having inherent float-restricting characteristics provide centering without any additional modifications (Figures 21 and 22).

VERTICAL OPERATIONS As previously noted, rigid couplings are commonly used on vertical-drive systems where the system characteristics warrant such a coupling. However, many vertical-drive systems require a flexible coupling to accommodate shaft misalignment. It is generally possible to use a nonlubricated coupling, such as one of the many elastomer designs, in a vertical position without modification, provided the shafts are supported in their own bearings and the coupling does not have to transmit a thrust force. Lubricated designs, such as the gear and spring-grid types, usually require some modification to make certain that lubricant is retained in both halves of the coupling.

PUMP DRIVE SHAFT SYSTEMS

Pump drive system arrangements may be classified in one of two categories: those that are close-coupled, having a shaft separation of a fraction of an inch (not to be confused with integral motor pumps not having a flexible coupling) and those that, for one or more reasons, have the prime mover located a substantial distance from the pump. These latter types of systems require a modification to the basic flexible coupling designs previously described.

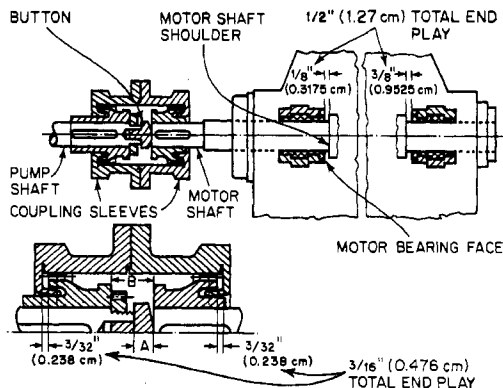


FIGURE 20 Motor-driven coupling end float limited by insertion of button between shaft ends (Flowservice Corp.)

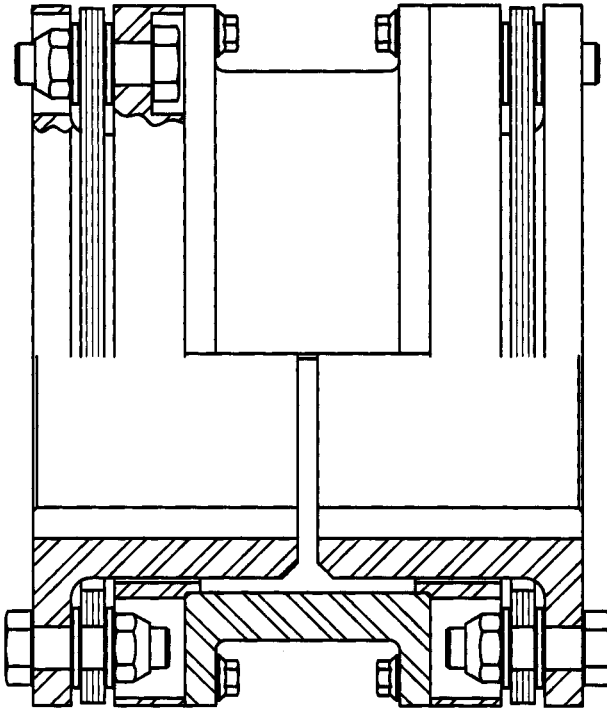


FIGURE 21 Close-coupled metal disk coupling having inherent float-restricting characteristics. (Falk Rexnord Couplings)

Spacers One means of accommodating shaft separation in excess of the normal amount provided in a standard coupling is to employ a flanged tubular spacer between the two coupling halves (Figure 23). These components are lightweight and are commonly used with end-suction pumps, where it is possible to remove the pump impeller while the pump housing and piping, as well as the prime mover, remain in place (Figure 24). For high-speed



FIGURE 22 Spacer type metal disc coupling (Lovejoy Inc)

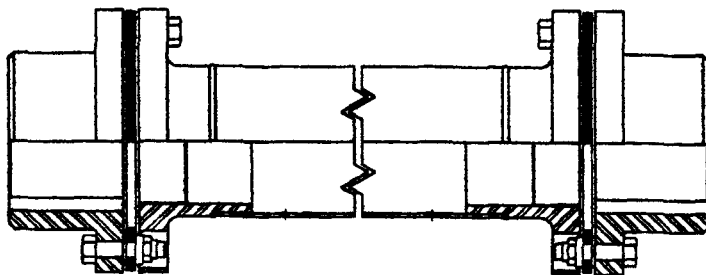


FIGURE 23 Gear coupling with tubular spacer (Lovejoy Inc)

drive systems, a lightweight spacer is usually the only practical means of achieving the goal. This spacer may be used to tune the system torsionally (as previously discussed) by varying the body diameter and wall thickness. Spacers can be manufactured in lengths up to several feet, but the cost generally restricts its usage to shorter lengths, except where no other means is suitable.

Floating Shafts Floating shaft couplings (Figure 25) have a purpose similar to that of spacer couplings; namely, to connect widely separated drivers from the driven equipment shafts. They may be used where a large radial misalignment is required. Whereas spacers are normally made with the body having an integrally formed flange to connect the two coupling halves, a floating shaft is made by connecting two couplings or U-joints with an unsupported, hence floating, piece of shafting. This type of construction is generally less expensive than a one-piece spacer, especially when very large shaft separations must be spanned. Graphite composite center members are also being used. This material has the advantage of being much lighter and stiffer than standard metal components, allowing longer spans between bearing supports without adding mass.

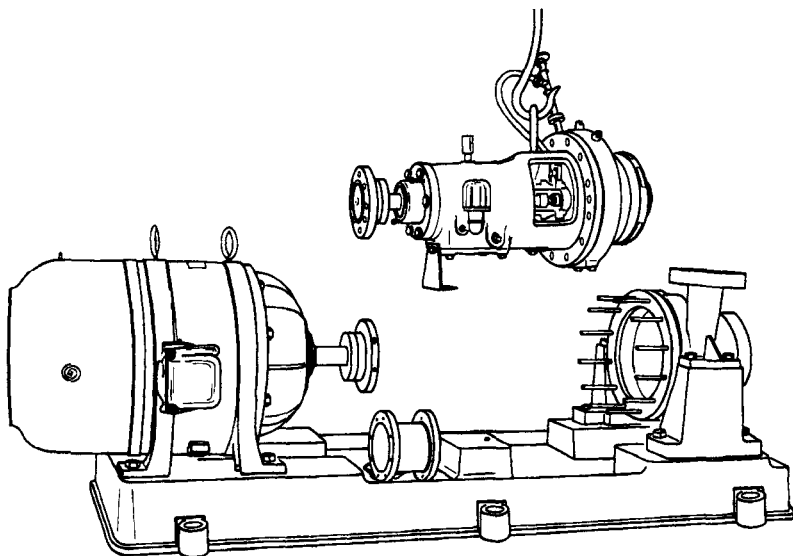


FIGURE 24 Spacer coupling enables end suction pump to be dismantled without disturbing piping, pump casing, or driver (Flowsolve Corp)

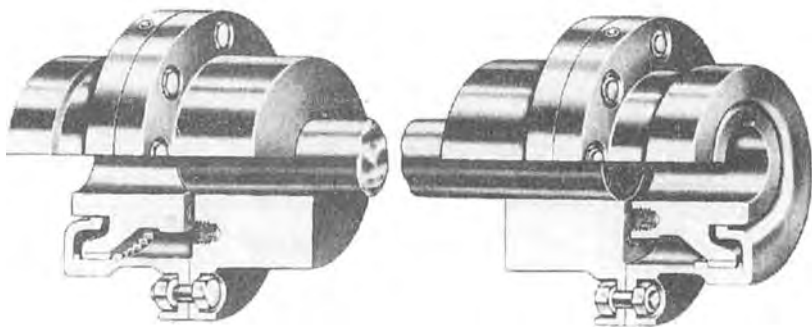


FIGURE 25 Section of floating shaft and coupling (Kop-Flex EPT)

Floating shaft arrangements are widely used on horizontal pump applications of all types and are especially common on vertical pump applications, such as in water pumping and sewage treatment stations. In these latter applications, the pump may be submerged in a dry pit usually 30 to 50 ft (9 to 15 m) below ground level, whereas the motor (to prevent damage during flooding) is mounted at ground level. Settings as deep as 100 ft (30 m) have been constructed. Such systems require long floating shafts, which are generally made in sections and supported by line bearings at intermediate supports or floors.

Rigid Shafts Vertical centrifugal pumps can be designed to contain their own thrust and line bearings. These pumps employ a flexible coupling at the pump shaft. When vertical intermediate shafting is used, the shaft need to be designed only to transmit torque. Some pumps are designed to contain only a single line bearing. These pumps require a rigid coupling at the pump shaft so axial thrust can be carried by the thrust bearing in the driver or gear located above. The constructions of a single oil-lubricated line-bearing pump is shown in Figure 79 of Section 2.2.1. In order to carry thrust, the drive shaft, if made in sections, must use rigid intermediate couplings and the coupling at the driver or gear must also be rigid. Intermediate bearings used with rigid shafting must be designed to provide only lateral support, thereby assuring that all the axial thrust is carried by the driver or gear. An intermediate oil-lubricated sleeve-type guide bearing is shown in Figure 78 of Section 2.2.1. Pump and intermediate shaft sleeve guide bearings can be either grease- or oil-lubricated, and they can also be antifriction if preferred.

A rigid intermediate shaft connected to a pump provided with a rigid coupling must be designed to carry its own weight, the axial thrust from the pump, and a bending load. Volute-type centrifugal pumps produce a radial reaction on the pump shaft that in turn is transmitted to the vertical shafting through the rigid pump coupling. Intermediate bearings then act as line bearings. The bearing supports must also be designed to resist this bending force. The pump shaft and bearing, intermediate shafts and line bearings, and the driver or gear shaft and bearings must be treated as a single shaft supported at each bearing when analyzing the critical speed of the entire rotor system. If the intermediate line bearing supports are assumed to be nodes in the critical speed calculation, these supports must be absolutely rigid. The supplier of the shafting should, however, confirm what flexibility will be allowed at these supports and give the design forces. The support for the guide bearings must also be designed not to have a natural frequency of vibration within the operating speed range of the pump.

Flexible Drive Shafts Universal joints with tubular shafting (Figure 26) can be substituted for flexible couplings whenever it is necessary to (a) eliminate the need for critical alignment, (b) provide wider latitude in placement of pump and driver, and (c) permit large amounts of relative motion between pump and driver. This type of shafting can be used horizontally as well as vertically to provide a short spacer arrangement or a large separation between pump and driver, such as that required for deep settings (Figure 27).



FIGURE 26 Tubular intermediate shafting and universal joints (H. S. Watson)

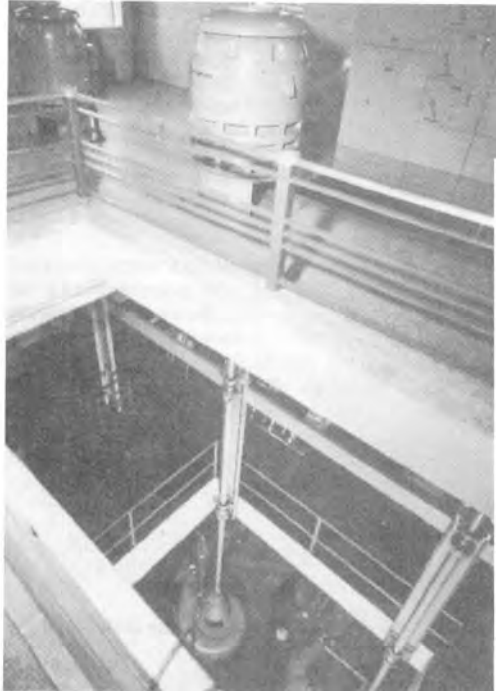


FIGURE 27 Sections of flexible shafts and intermediate guide bearings (Flowserve Corp)

Flanges are furnished to fit pump and driver shafts to the universal joints, which are splined to allow movement of the shaft. An intermediate steady bearing is required at each joint, which must also support the weight of a section of shafting (Figure 28). Because of the spline, pump thrust cannot be taken by the driver, and it is necessary to use a combination pump thrust and line bearing. Because of the universal joint, the intermediate bearing or bearings take no radial load from the pumps and therefore act only as a steady bearing or bearings for the shaft.

The shaft must be selected to transmit the required torque and be of a length and diameter that will have a critical speed well removed from the operating speed range of the pump. The support for the guide bearing must not have a natural frequency of vibration within the speed range. Shafts of this type are often pre-engineered and stocked. Selection charts are available from the manufacturer to make a proper size and length selection. Standard tubular, flexible drive shafts have limited torque-carrying capacity.

Design Criteria

COUPLING BORES Couplings are mounted to the pump and driver shafts using clearance fit or shrink fit. They can be fitted with keys or not and can be cylindrical or tapered style depending on the shaft to which they are attached. Standards have been developed to assure communication between the equipment manufacturer and the coupling manufacturer. See the ANSI/AGMA references at the end of this chapter. Shaft standards exist and the coupling bore standards accommodate those whenever possible. It is important that the coupling bore is suitably concentric with the hub OD and that the shaft have little run-out so that extra misalignment or unbalance is not introduced to the system. It is

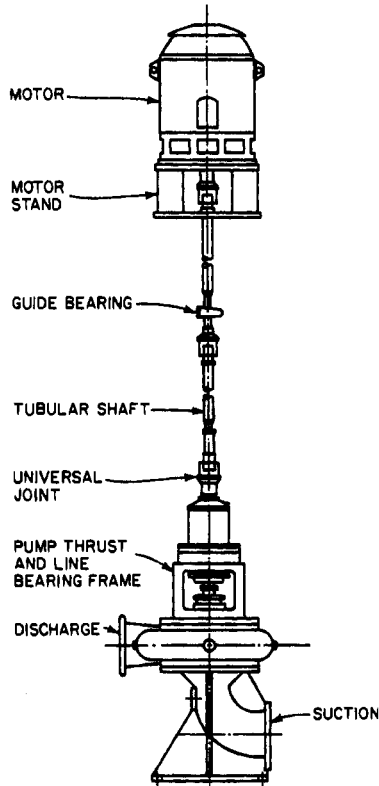


FIGURE 28 Vertical pump with flexible shafting (Flowservice Corp)

also important that the shaft size, coupling bore, and torque transmission method are suitable for the amount of torque to be transmitted.

Clearance fits with keys are used for low power transmission and easy removal of the coupling hub when maintenance is needed. Tapered shafts with keys can be used for easy removal also. Shrink fits with keys and heavy shrink tapered fits without keys are used for high-power, high-speed applications.

TORSIONAL STRESS The torsional stress in the shafting may be calculated by the following equations:

$$S_s = \frac{16T}{\pi D_o^3} \quad \text{for solid shafting}$$

$$S_s = \frac{16T}{\pi D_o^3 \left(1 - \frac{D^4}{D_o^4}\right)} \quad \text{for tubular shafting}$$

where S_s = torsional shear stress, lb/in² (N/m²)

T = transmitted torque, in · lb (N · m)

D_o = shaft outer diameter, in (m)

D = shaft inner diameter (tubular shaft only), in (m)

The allowable shear stress depends upon the material being used and whether it is subjected to other loads, such as bending or compression. The design safety factor on the shafting should be equal to or greater than those of the other components in the drive train.

CRITICAL SPEED The lateral critical speed of a drive shaft is determined by the deflection, or "sag," of the shaft in a horizontal position under its own weight. The less the sag, the higher the critical speed. In practical terms, a long, slender shaft will have a low critical speed and a short, large-diameter shaft will have a very high critical speed. The deflection of a simply supported shaft is calculated as follows:

$$y = \frac{5wL^4}{384EI} = \text{shaft deflection, in (m)}$$

noting that

$$I = \frac{\pi D_o^4}{64} \quad \text{for solid shafting}$$

$$I = \frac{\pi(D_o^4 - D^4)}{64} \quad \text{for tubular shafting}$$

where w = weight (force) of shaft per unit length, lb/in (N/m)

L = length between bearing supports, in (m)

E = Young's modulus, lb/in² (N/m²)

I = moment of inertia, in⁴ (m⁴)

Knowing the natural deflection of the shaft, it is possible to calculate the first critical speed from the equation:

$$N_{\text{crit}} = 187.7 \sqrt{\frac{1}{y \text{ (in inches)}}} = 946 \sqrt{\frac{1}{y \text{ (in millimeters)}}}$$

which expresses critical speed directly in revolutions per minute of the rotating shaft.

In practice, the critical speed should be placed well away from the operating speed of the shaft. Actual operating conditions, such as imbalance and clearance in bearings and couplings, tend to reduce the critical speed from its theoretical value. The designer has several options from which to choose to attain the desired critical speed: (1) vary the shaft diameter, (2) use tubular shafting to increase stiffness and reduce deflection, and (3) vary the bearing span.

Varying the diameter of the shaft has a dramatic effect on shaft deflection because the deflection decreases inversely with the cube of the diameter. The weight of the shaft will increase proportionately to the square of the diameter and may consequently impose excessive loads on bearings. When this is the case, tubular shafting may be substituted for solid shafting. The ratio I/w is much higher for a tubular shaft than for a solid shaft of the same diameter, so the shaft deflection is again reduced and critical speed increased.

The bearing span may be varied when the size and weight of a single length is broken into sections, with a single engagement coupling used to connect each section and acting as a hinge joint. A self-aligning bearing is placed on each section immediately adjacent to the coupling. Each section may then be treated as an individual shaft, which will result in a substantial reduction in shaft size. The rigidity of the support at each bearing must be taken into consideration when calculating the critical speed of the shaft.

DYNAMIC BALANCE Drive systems that operate at high speeds and connect two widely separated shafts must be given special consideration with regard to dynamic imbalance. Care must be taken to ensure that the long, slender shaft used in such systems is not bent, either in manufacture or in installation. These systems quite often require shafting that has been dynamically balanced after manufacture to compensate for these normal errors.

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PUMP NOISE

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Cecil R. Sparks

J. C. Wachel

Donald P. Sloteman

The major concern regarding pump noise falls into two categories:

1. Noise levels that do not meet applicable environmental criteria. Examples range from personnel noise exposure criteria to overside noise criteria for submarines.
2. Noise signatures that can be used to diagnose faulty pump operation or incipient failure.

The proliferation of industrial noise regulations in recent years has taken much of the guesswork out of allowable noise levels insofar as personnel and community exposure is concerned, and various noise standards have specified noise measurement techniques. Several organizations have developed test procedures and codes for machinery-generated noise levels.^{1,2} The Hydraulic Institute code was specifically developed for the measurement of airborne sound generated by pumps (Reference 3).

The most common approach for controlling airborne noise levels from pumps is to interrupt the paths by which noise reaches the listener. When noise is used as an indicator of abnormal pump operation, modification of pump internals or operating conditions is normally required.

The measurement of noise for diagnostic purposes is not well prescribed, either for instrumentation or for interpretation. Even a well-designed and properly operated pump will of course produce noise. Variations in noise amplitude and frequency that result from malfunction or improper operating conditions will depend upon the type and design of the pump and the type of problem causing the noise. Measurement and analysis techniques for interpreting these signatures will depend upon whether the noise is solid-, liquid-, or airborne and upon the nature of coexisting noise from other sources.

Determining the source and cause of noise is the first step in evaluating whether noise is normal or an indicator of possible problems. Noise in pumping systems can be generated both by the mechanical motion of pump components and by the liquid motion in the pump and piping systems. Liquid noise sources can result from vortex formation in high-velocity (shear) flow, from pulsating flow, and from cavitation and flashing.

Noise from internal mechanical and liquid sources can be propagated to the environment by several paths, including the pump and support structure, attached piping, the liquid in the piping, and ultimately the surrounding air itself.

This section discusses various pump noise-generating mechanisms (sources) and common noise conduction paths as a basis for both effective diagnostics and treatment. In addition, a method to estimate pump noise levels for conventionally designed pumps will be presented.

NOISE MEASUREMENT

In very general terms, sound consists of cyclic modulations of ambient pressure at frequencies that are detectable to the human ear, normally between 30 Hz and 17 kHz. For our purposes, noise is simply unwanted sound. The complications in measuring noise or in assessing its effects on personnel arise chiefly because of the remarkable complexity of the human ear and its very nonlinear acoustic response with both frequency and amplitude. The dynamic range of the human ear is 160 dB, from a minimum detectable level of about 3.7×10^{-9} lb/in² to about 0.37 lb/in² (2.6×10^{-10} to 0.026 bar*). In order to compensate for the ear's dynamic range, the decibel scale was adapted to roughly simulate the ear's nonlinear sensitivity and reduce numerical sound level values to a manageable range. The strength of familiar sounds relative to their typical decibel level is given in Figure 1.

The decibel is a unit of sound measurement relating the dynamic pressure variations produced by the source of interest to a reference sound pressure (0.0002 μ bar). The sound pressure level in decibels is equal to 20 times the common logarithm of this ratio:

$$SPL = 20 \log \left(\frac{P}{0.0002} \right) \quad (1)$$

where P is the rms sound pressure in microbars.

*1 bar = 10^5 Pa.

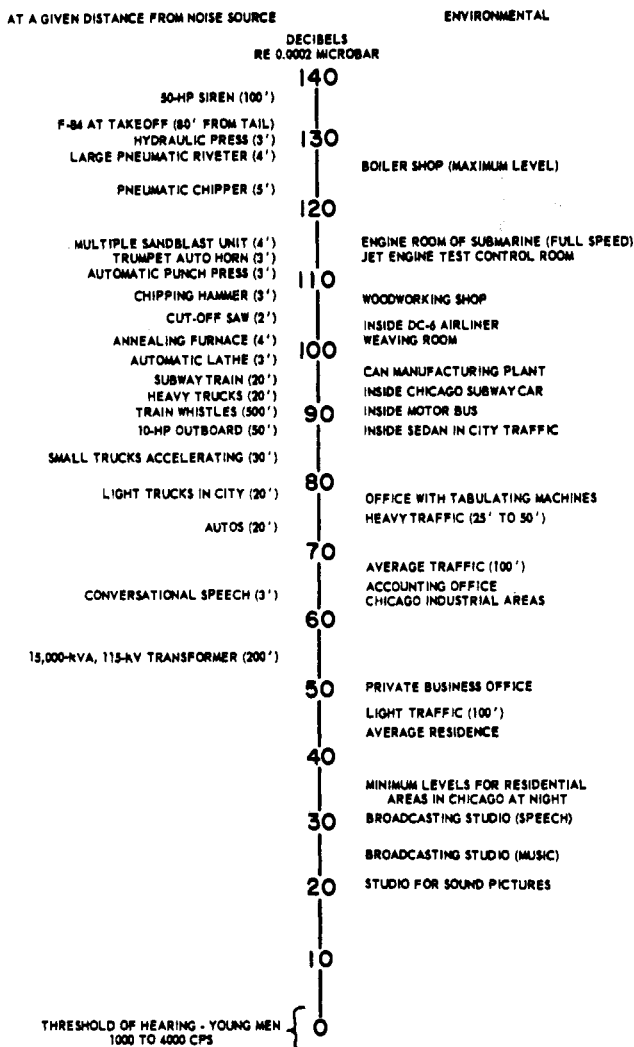


FIGURE 1 Typical overall sound levels (1 ft = 0.3048 m) (Reference 15)

For example, if one sound pressure level is twice another, the measured noise level will be 6 dB greater.

The basic instrument used to measure sound intensity is the sound level meter, which provides a numerical reading of decibel values by integrating sound throughout the audible frequency spectrum. Because the frequency response of the ear varies with sound intensity, the sound level meter has three internal filters that can be used to approximate the ear's frequency response at 45, 75, and 95 dB. These are called the A, B, and C filters, respectively. In recent years, use of the C weighing filter has become popular in prescribing noise criteria. The dBC scale has been thus adapted, partially because it is easy to use and partially because it attenuates very-low-frequency noise, which is not as potentially injurious to the ear.

SOURCES OF PUMP NOISE

Effective control of pump noise requires knowledge of the liquid and mechanical noise-generation mechanisms and the paths by which noise can be transmitted to a listener.

Mechanical Noise Sources Mechanical sources are vibrating components or surfaces which produce acoustic pressure fluctuations in an adjacent medium. Examples are pistons, rotating unbalance vibrations, and vibrating pipe walls.

In positive displacement pumps, noise is generally associated with the speed of the pump and the number of pump plungers. Liquid pulsations are the primary mechanically induced noise, and these in turn can excite mechanical vibrations in components of both pump and piping system. Incorrect crankshaft counterweights will also cause shaking at running speed, which may loosen anchor bolts and produce rattling of the foundation or skid. Other mechanical noises are associated with worn hearings on the connecting rods, worn wrist pins, or slapping of the pistons or plungers.

In centrifugal machines, improper installation of couplings often causes mechanical noise at twice pump speed (misalignment). If pump speed is near or passes through the lateral critical speed, noise can be generated by high vibrations resulting from imbalance or by the rubbing of bearings, seals, or impellers. If rubbing occurs, it may be characterized by a high-pitched squeal. Windage noises may be generated by motor fans, shaft keys, and coupling bolts.

Liquid Noise Sources When pressure fluctuations are produced directly by liquid motion, the sources are fluid dynamic in character. Potential fluid dynamic sources include turbulence, flow separation (vorticity), cavitation, waterhammer, flashing, and impeller interaction with the pump cutwater (the interaction of a rotating and a stationary pressure field). The resulting pressure and flow pulsations may be either periodic or broadband in frequency and generally excite either the piping or the pump itself into mechanical vibration. These mechanical vibrations can then radiate acoustic noise into their environment.

In general, pulsation sources are of four types in liquid pumps:

1. Discrete-frequency components generated by the pump impeller or plungers
2. Broad-band turbulent energy resulting from high flow velocities
3. Impact noise consisting of intermittent bursts of broad-band noise caused by cavitation, flashing, and waterhammer
4. Flow-induced pulsations caused by periodic vortex formation when flow is past obstructions and side branches in the piping system

A variety of secondary flow patterns that produce pressure fluctuations are possible in centrifugal pumps, as shown in Figure 2, particularly for operation at off-design flow. The numbers shown in the flow stream are the locations of the following flow mechanisms:

1. Stall
2. Recirculation (secondary flow)
3. Circulation
4. Leakage
5. Unsteady flow fluctuations
6. Wake (vortices)
7. Turbulence
8. Cavitation

Most of these unstable flow patterns produce vortices by boundary layer interaction between a high-velocity and low-velocity region in a fluid field, for example, by flow around obstructions or past deadwater regions or by bidirectional flow. The vortices, or eddies, are

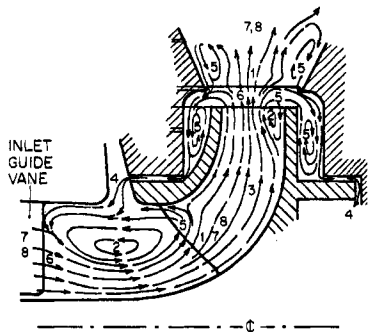


FIGURE 2 Secondary flows in and around a pump impeller stage (Reference 12)

converted to pressure perturbations as they impinge on the sidewall and may result in localized vibration excitation of the piping or pump components. The acoustic response of the piping system can strongly influence the frequency and amplitude of this vortex shedding. Experimental work has shown that vortex flow is most severe when a system acoustic resonance coincides with the natural or preferred generation frequency of the source. This source frequency has been found to correlate to a Strouhal number (S_n) ranging from 0.2 to 0.5, where

$$S_n = \frac{f_e D}{V} \tag{2}$$

- where f_e = vortex frequency, Hz
 D = a characteristic dimension of the generation source, ft (m)
 V = flow velocity in the pipe, ft/s (m/s)

For flow past tubes, D is the tube diameter, and for branch piping excitation, D is the diameter of the branch pipe. The basic Strouhal equation is further defined in Table 1,

TABLE 1 Piping vibration excitation forces

Generation mechanism	Excitation frequency f_e , Hz
1. Reciprocating compressors	nf
2. Reciprocating pumps	nf, nPf
3. Centrifugal compressors and pumps	nf, nBf, nvf
4. Flow excitation	
A. Flow through restrictions	$\frac{0.2V}{D}$ to $\frac{0.5V}{D}$
B. Flow past stubs	$\frac{0.5V}{D}$
C. Flow turbulence due to quasi-steady flow	0–30 Hz (typically)
D. Cavitation and flashing	Broad band

- $n = 1, 2, 3, \dots$
 f = running speed, Hz
 P = number of pump plungers
 B = number of blades
 v = number of volutes or diffuser vanes
 V = flow velocity, ft/s (m/s)
 D = restriction diameter, ft (m)

items 4A and B. As an example, flow at 100 ft/s (30 m/s) past a 12-in diameter (0.3-m) stub line would exhibit instability tendencies at a frequency of approximately 50 Hz. If the stub is acoustically resonant at a frequency near 50 Hz, rather large pulsation amplitudes can result.

When a centrifugal pump is operated at flows less than or greater than best efficiency capacity, noise is usually heard around the pump casing. The magnitude and frequency of this noise vary from pump to pump and are dependent on the magnitude of the pump head being generated, the ratio of NPSH required to NPSH available, and the amount by which pump flow deviates from ideal flow. Noise is often generated when the vane angles of the inlet guide, impeller, and casing (or diffuser) are incorrect for the actual flow rate. Another major source contributing to this noise is referred to as *recirculation*.^{4,5}

Before the pressure of the liquid flowing through a centrifugal pump is increased, the liquid must pass through a region where its pressure is less than that existing in the suction pipe. This is due in part to acceleration of the liquid into the eye of the impeller. It is also due to flow separation from the impeller inlet vanes. If flow is in excess of design and the incident vane angle is incorrect, high-velocity, low-pressure eddies will form. If the liquid pressure is reduced to the vaporization pressure, the liquid will flash. Later in the flow path the pressure will increase. The implosion that follows causes what is usually referred to as *cavitation* noise. The collapse of the vapor pockets, usually on the nonpressure side of the impeller blades, causes severe damage (blade erosion) in addition to noise.

Sound levels measured at the casing of an 8000-hp (5970-kW) pump and near the suction piping during cavitation are shown in Figure 3. The cavitation produced a wide-band shock that excited many frequencies; however, in this case, the vane passing frequency (number of impeller blades times revolutions per second) and multiples of it predominated. Cavitation noise of this type usually produces very-high-frequency noise, best described as "crackling."

Cavitation-like noise can also be heard at flows less than design, even when available inlet NPSH is in excess of pump required NPSH, and this has been a puzzling problem. An explanation offered by Fraser^{4,5} suggests that noise of a very low random frequency but very high intensity results from backflow at the impeller eye or at the impeller discharge, or both, and every centrifugal pump has this recirculation to some degree under certain conditions of reduced flow. Operation at low suction pressure and in a recirculating condition can be damaging to the pressure side of the inlet or discharge impeller blades (and also to casing vanes) if cavitation is produced. Recirculation is evidenced by an increase in loudness of a banging type, random noise, and an increase in suction or discharge pressure pulsations as flow is decreased. Refer to Sections 2.1.3 and 2.1.4 for further information.

Pressure regulators or flow control valves may produce noise associated with both turbulence and flow separation. These valves, when operating with a severe pressure drop, have high flow velocities that generate significant turbulence. Although the generated noise spectrum is very broad-band, it is characteristically centered around a frequency corresponding to a Strouhal number of approximately 0.2.

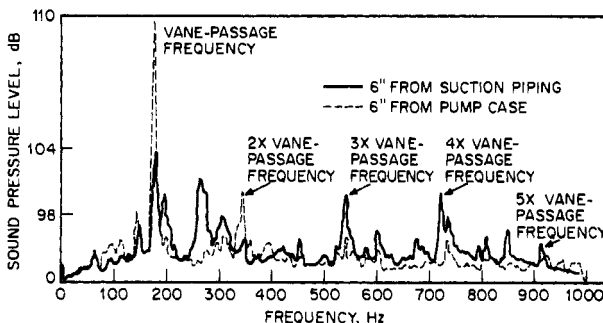


FIGURE 3 Noise spectra of cavitation and vane passage on a centrifugal pump

CAVITATION AND FLASHING For many liquid pump piping systems, it is common to have some degree of flashing and cavitation associated with the pump or with the pressure control valves in the piping system. High flow rates produce more severe cavitation because of greater flow losses through restrictions.

In the suction piping of positive displacement pumps, high-amplitude pulsations can be generated by the plungers and amplified by the system acoustics and cause the dynamic pressure to periodically reach the vapor pressure of the liquid even though the suction static pressure may be above this pressure. As the cyclic pressure increases, the vapor bubbles collapse, producing noise and shock to the system, and this can result in erosion as well as undesirable noise. (See Figure 9, Section 3.4.)

Flashing is particularly common in hot water systems (feedwater pump systems) when the hot, pressurized water experiences a decrease in pressure through a restriction (for example, flow control valve). This reduction of pressure allows the liquid to suddenly vaporize, or flash, which results in a noise similar to cavitation. To avoid flashing after a restriction, sufficient back pressure should be provided. Alternately, the restriction could be located at the end of the line so the flashing energy can dissipate into a larger volume.

ESTIMATING PUMP NOISE LEVEL

Pump noise is only one component of pumping system noise. Usually, the pump and driver are included in any specified noise level. The noise level from the driver, whether electric motor, engine, or steam turbine must be acquired from the supplier of that particular driver. The noise from these components can be added together (described later in this section) to arrive at the total noise for the pump/driver system. As described in the previous section, the pump generates noise from a variety of sources, over a wide range of frequencies and radiated from not only the pump casing but also from surfaces directly connected to the pump (e.g., piping). Due to the complex nature of pump noise generation it is best to acquire specific test data for the pump in conditions similar to the application conditions. One alternative to testing is to perform a detailed transient analysis of the internal flow using computational fluid dynamics as well as a transient mechanical analysis (considering the unbalance, misalignment, bearing forces, structural displacement, etc.) and use these results to predict displacement of the different surfaces. The displacement of the surfaces is translated into the movement of air which in turn generates the sound pressure level that is measured as airborne noise. The software, computational power and effort to perform such an analysis is complex and costly and is reserved for the most critical and high-value applications. The result is detailed enough to produce the type of spectrum of amplitude versus frequency seen in Figure 3. This is often referred to as a narrow-band spectrum. A second alternative involves using an empirical correlation to predict noise levels for the pump. The correlation must be robust enough to account for pump configuration, energy level, and flow rate relative to design flow.

One such correlation to predict pump noise levels was developed by a major pump manufacturer and is presented here to enable an estimate of approximate airborne noise level for the conventional style of pumps that were included in the correlation. Airborne noise data were collected for a variety of centrifugal pumps during shop floor testing. During this testing, the driver was effectively isolated so that the noise measured was dominated by the pump and valve used to control flow. This work produced a methodology for predicting pump noise (minus driver) as an overall level (overall dB weighted by the 'A' scale, dBA) and as levels associated with octave band frequencies. Octave bands cover a broad range of frequencies and combine the amplitude levels occurring over that frequency range into a single value. This is different from the narrow-band spectrum (seen in Figure 3) where the energy levels are measured over a very narrow frequency range. Reference 6 defines the basis of these octave bands and how a signal is measured and how it is distributed within each octave. Each frequency range or octave is referenced by its mid-band frequency.

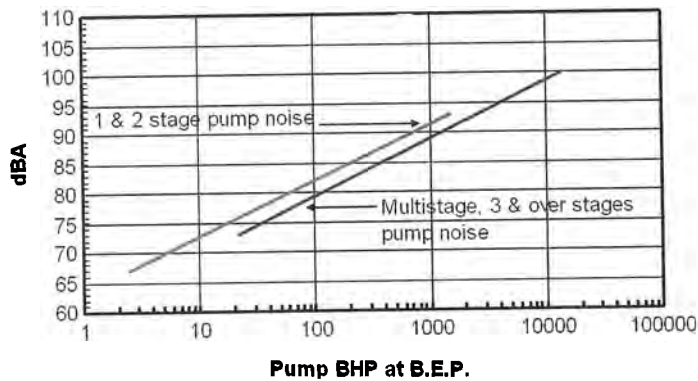


FIGURE 4 Estimate of dBA at BEP using pump BHP ($BHP = 1.34 \times kW$)

Methodology The correlation is derived from tests conducted on radial-type centrifugal pumps. It is not applicable for solids-handling, vertical, mixed- or axial-flow pumps. The measuring point is located in a free field above a reflecting plane (e.g., the baseplate) in which the reduction in noise level in all directions is 6 dB in each octave band for each doubling of distance. Background noise is at least 10 dB below the noise levels associated with each octave band. The pumps were not operating in a cavitating condition. The values are for measurement points 1 m (horizontally) from major pump surfaces and 1.5 m above the floor. The standard pressure reference for converting to decibels is 0.00002 Pa.

Overall Level at BEP An estimate of the overall level of pump noise at best efficiency point (BEP) is found from Figure 4. The dBA level correlated with brake horsepower (BHP). Two different pump configurations are considered, 1 and 2 stage pumps and multistage pumps of 3 stages or greater.

If pumping fluid with specific gravity (SG) of less than 1.0, use the BHP associated with 1.0 SG fluid. For SG greater than 1.0, use the BHP associated with the actual SG.

Part Load Correction Once the overall dBA level is found at the BEP a correction for operation of the pump at lower or higher flow rates is made using Figure 5. This correction

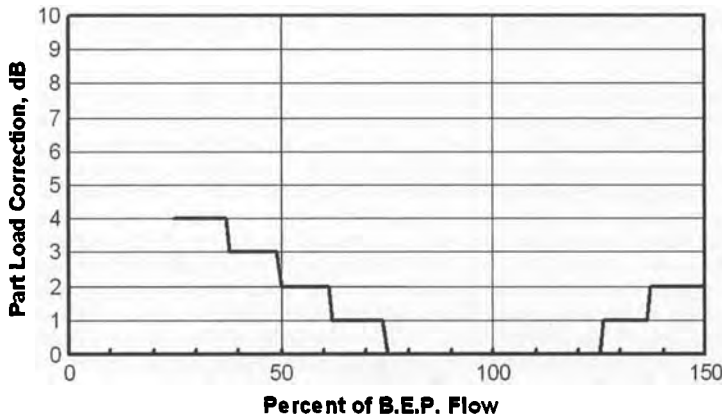


FIGURE 5 Part load correction to overall level at BEP

TABLE 2 Octave, Mid-Band Corrections to Overall Level

		Octave Mid-Band Frequency, Hz								
		31.5	63	125	250	500	1K	2K	4K	8K
Deduct from dBA level, dB	Single & 2 stage	4	6	4	3	4	5	6	9	12
	Multistage (3 and over)	11	9	7	5	3	3	3	7	12

accounts for the levels associated with noise caused by greater radial loads, discharge and suction recirculation, and the increased turbulence levels associated with this behavior. The dB value associated with the percent of BEP flow is added to the overall level at BEP from Figure 4.

Noise Levels in Octave Bands Understanding how pump noise level translates to octave bands is helpful in understanding effects an observers or how attenuation methods should be directed to reduce noise in a specific frequency range. Table 2 is used to predict how the overall level would be decomposed into the specific octave band frequency ranges. This is a generalized approximation and will vary depending on several factors. Rotating speed and vane number combination are two variables that affect how levels may differ in bands that contain frequency content associated with rotation and vane pass signals. So, Table 2 is useful as an approximation, based on the variety of pumps tested.

Noise Reduction over Distance It is often useful to be able to predict the noise level at distances other than 1 m from the noise source. The attenuation of level can be predicted using the simple relationship shown in Eq. 3. This relationship is valid when the surroundings can be characterized as a “free field” (no obstructions) above a hard reflecting plane (e.g., the plant floor). If these conditions are not met, then determining the attenuation amount requires more sophisticated means.

$$\text{dB Reduction} = 20 \log_{10} (\text{Distance in meters}) \quad (3)$$

This relationship is shown in Table 3 for various select distances.

Combining Noise Levels Pumps require some form of driver and sometimes gearboxes to change the driver speed to the desired pump speed. These additional components generate noise based on their design. The supplier of these components must be able to provide overall levels and octave band levels so that the overall noise level of the system can be predicted. Since sound pressure level is referred to as a decibel and is based on the log of a pressure ratio (see Eq. 1), the decibel values cannot simply be added together. Eq. 4 shows how various signal levels, L_a in decibels, are added together to get a total level.

$$\text{dB Total} = 10 \log \sum 10^{L_a/10} \quad (4)$$

TABLE 3 Attenuation of noise level for distance from source

		Noise Level Reductions for Distances over 1 m									
	Feet	3.28	5	10	20	30	40	50	100	150	200
Distance from source	Meters	1	1.52	3.05	6.1	9.15	12.2	15.2	30.5	45.7	61
dB Reduction	dB	0	3	9	15	19	21	23	29	33	35

TABLE 4 Combining two noise levels

	dB Correction for Combining Noise Levels						
Difference between two levels to be combined, dB	0	1	2	4	6	9	10
Add to higher level to obtain combined level, dB	3	2.5	2	1.5	1	0.5	0

An alternative to calculating the combined level from Eq. 4 is to use Table 4. This table is based on Eq. 4 and shows how two levels are combined without resorting to the required mathematics.

Noise Level Example Problem The following example shows how to use this correlation to predict noise levels for a conventionally designed pump. The pump in this example is a horizontally split, double-suction, single-stage pump. It is driven by a steam turbine with a gearbox between the driver and the pump. Specifics regarding this example pump are:

Condition point flow = 7500 gpm

BEP flow = 11,060 gpm

BEP head = 103 ft

BEP efficiency = 87.5%

SG = 0.94

Pump speed = 880 rpm

The shaft horsepower at BEP at SG of 1.0 (for SG <1.0 use the value of 1.0 to determine BHP) is calculated to be 329 BHP. From Figure 4 the estimated overall noise level at BEP is 87 dBA. Since the pump is being operated below the BEP flow (7500 gpm is 68% of the BEP flow of 11,060 gpm) a part load correction to noise level from Figure 5 is needed. From Figure 5, for 68% of BEP, 1 dB must be added to the overall noise level so the overall noise level at condition point is 88 dBA.

An approximation of the octave band noise distribution can be made using Table 2. Table 5 shows how the noise level at BEP is used to develop the levels at mid-octave band and how these levels are modified with the part load correction to provide the octave band prediction at the condition point.

The noise levels of the supporting equipment and pump are:

Turbine 82 dBA
Gear 91 dBA
Pump 88 dBA (at part load)

TABLE 5 Combining two noise levels

		Mid-Band Frequency, Hz								
	dBA	31.5	63	125	250	500	1K	2K	4K	8K
Deduct for mid-band	dB	-4	-6	-4	-3	-4	-5	-6	-9	-12
Noise level, BEP	87	83	81	83	84	83	82	81	78	75
Part load correction	+1	+1	+1	+1	+1	+1	+1	+1	+1	+1
Noise at condition pt.	88	84	82	84	85	84	83	82	79	76

$$\text{dB Total} = 10\text{Log}(10^{82/10} + 10^{91/10} + 10^{88/10}) = 93.1 \text{ dBA}$$

Or using Table 4:

Step 1: 88 – 82 = 6 Add 1 to 88 = 89 dBA

Step 2: 91 – 89 = 2 Add 2 to 91 = 93 dBA

93 dBA total system noise @ 1 meter from equipment

If it is desirable to predict what the total system noise level would be farther from the equipment, Eq. 3 or Table 3 can be used to determine the amount to subtract from the 1 m distance value.

This empirically based correlation was developed from conventional centrifugal pumps without the addition of any special noise attenuating features (either within the pump or to the mated components). It is useful, however, in demonstrating factors to consider when evaluating the noise generated by the pumping system. Later sections will discuss control techniques and how to test the combined system for determining the actual noise levels produced.

NOISE CONTROL TECHNIQUES

Environmental noise usually does not emanate directly from the energy source; rather, it is transmitted along mechanical or liquid paths before it finally radiates from some vibrating surface into the surrounding environment.

The approaches to treating pump noise generally include the following:

1. **Source modification** Modify the basic pump design or operating condition to minimize the generation of acoustic energy.
2. **Interruption of transmission** Prevent sources from generating airborne (or over-side) noise by interrupting the path between the energy source and the listener. This approach may range from isolation mounts at the source to physically removing the listener.

Source Modification Equipment modification to eliminate the noise source is usually quite specific to each particular situation; therefore, only general guidelines can be given. Many technical papers relate pump configurational modifications to the degree of noise reduction achieved; however, noise reduction depends on many parameters, and hence a particular modification may or may not help in a specific case. On the other hand, if the noise results from a resonant condition in the pump system, almost any reasonably conceived modification can destroy the resonance with varying degrees of improvement.

Some of the source modification approaches for pump applications are:

1. Increase or decrease pump speed to avoid system resonances of the mechanical or liquid systems.
2. Increase liquid pressures (NPSH, and so on) to avoid cavitation or flashing; decrease suction lift.
3. Balance rotating or oscillating components.
4. Change drive system to eliminate noisy components.
5. Correct acoustic resonance to minimize liquid-borne energy.
6. Modify centrifugal pump casing vanes so clearance between impeller diameter and casing cutwater (tongue) or diffuser vanes is increased.
7. Modify centrifugal pump impeller discharge blade configuration by pointing, slanting, grooving, adding holes, or staggering one-half pitch (if double suction).
8. Modify centrifugal pump casing cutwater (tongue) by slanting or adding holes.

9. Replace pump with different model or type to permit operation at reduced speed and the least number required.
10. If noise is due to operation of a centrifugal pump at flows less than design and recirculation is the problem, install minimum flow recirculation system bypass to increase total pump flow; if several pumps are operating in parallel, operate all pumps at the same speed and the least number required.
11. Use heavier bearing lubricant or increase number of bearing rolling elements.
12. Inject small quantity of air into the suction of a centrifugal pump to reduce cavitation noises.

The degree of improvement that can be achieved by any of the source modification approaches obviously depends upon the particulars of each installation, that is, the basic causes of excess noise. Modification of the pump internals, for example, is extremely difficult in existing installations and can produce undesirable side effects unless the pump was poorly designed or selected initially.

As described earlier, the major sources of internal noise in reciprocating pumps are usually associated with piston-induced pulsations, piston mechanical reactions, turbulence, vortex formation from separated flow around obstructions, and cavitation. In centrifugal pumps, in addition to recirculation noise, interaction of the impeller flow with the pump case (especially the cutwater), high-velocity and pressure gradients at the impeller blade tip, and flow separation can make significant contributions to pulsation levels and noise. Internal modifications to the pump can ameliorate any or all of these conditions if they are severe initially. The techniques are well known to most pump designers: Use adequate valve sizes, avoid high velocities and obstructed flows, keep pressures above the vapor pressure of the fluid being pumped, degasify the fluid, provide adequate pulsation control equipment, and maintain proper angles of attack in centrifugal machines.

Many references give examples of how changing pump design parameters affects noise.⁷⁻¹¹ Although such examples are valuable, they are of interest more in suggesting approaches than in predicting the degree of noise reduction that can be achieved in other pump applications.

Sudo, Komatsu, and Kondo¹⁰ investigated pressure pulsations (and noise) generated in a centrifugal pump as a result of interference between the impeller discharge vanes and the receiving spiral casing single cutwater vane. The geometry of the pump (Figure 6) defines the gap G between the impeller outer diameter D_2 and the casing cutwater. How varying the gap G/D_2 and the skew ratio (inclination of the cutwater or impeller vanes) affected discharge pressure pulsations is shown in Figure 7. Increasing G and the skew ratio decreases pulsation and noise amplitudes. Some investigators suggest that the ratio of cutwater diameter to impeller outer diameter be as large as 2:1 for optimum operation. Using this ratio would lead to a larger than practical pump casing. When the cutwater diameter (or gap) is

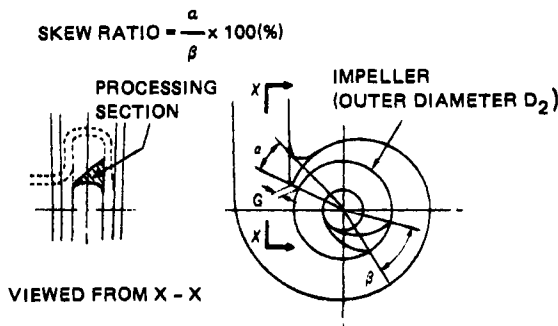


FIGURE 6 Cutwater of a spiral casing and impeller (Reference 10)

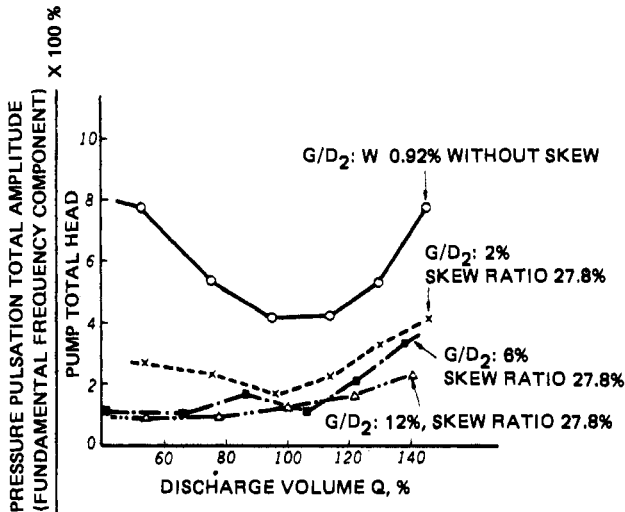


FIGURE 7 Effects of gap width between cutwater of a spiral casing and trailing edge of the impeller blade (Reference 10)

unknown, a rule of thumb suggested to optimize impeller diameter selection is not to use an impeller larger than 85% of maximum diameter. This percentage can be increased when using a multivaned diffuser with suitable impeller and diffuser vane number combination.¹³

To reduce the effect of impeller/casing vane passing pulsations, and consequently noise, double-suction impellers should have staggered vanes; i.e., the discharge vane tips should be shifted one-half pitch. This allows the fluctuation of the flow from each half of the impeller to interfere and thus reduces pressure pulsations at the pump discharge.

Florjancic, Schöffler, and Zogg¹¹ have reported that centrifugal pump impeller blade and casing tongue configuration can affect sound pressure levels and alter pump head and efficiency (Figure 8).

Control of Noise Paths These approaches consist of system modifications and treatments to disrupt the conduction of sound, whether borne by the structure, liquid, or air. They include approaches other than those that directly affect the originating source of oscillating energy (Figure 9).

LIQUID PATH Noise generated in the region of the pump is conducted both upstream and downstream by the liquid in the piping system. Such paths are most effectively reduced by acoustic (pulsation) filters or other pulsation control equipment, such as side branch accumulators (see Section 3.6).

STRUCTURE-BORNE NOISE Obvious paths for solid-borne noise are the attached piping and pump support systems. Oscillatory energy generated near the pump by pulsation, cavitation, turbulence, and so on, can be conducted as solid-borne noise for substantial distances before it is radiated as acoustic noise into the atmosphere. Some success in controlling solid-borne noise propagation can be achieved by the use of flexible couplings in the piping systems, mechanical isolation (vibration mounts and so on) for the pump and drive systems, and resilient pipe hangers and supports.

Techniques for the vibration isolation of mechanical equipment are well known and available from many suppliers of vibration isolators or mounts. These may consist of resilient supports at each mounting point of the machine, although it is often necessary to mount the pump and drive system on a single rigid skid (to assure alignment) and then

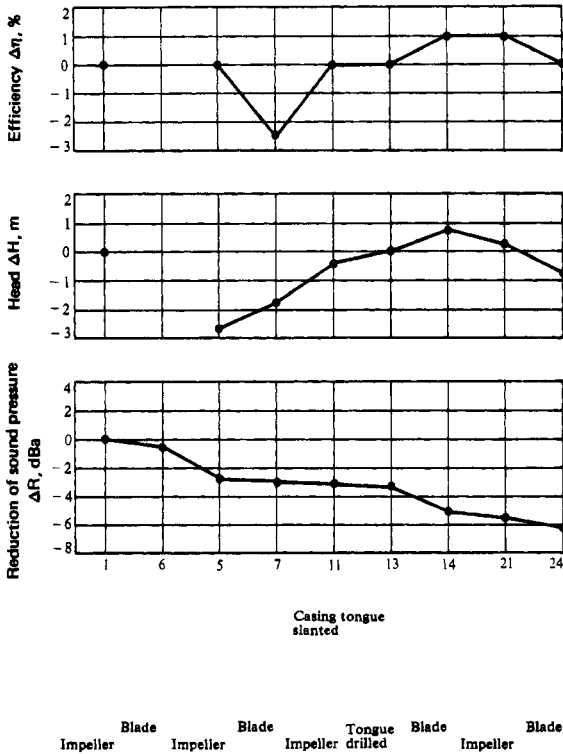


FIGURE 8 Reduction in sound pressure level with influences on head and efficiency at 100% flow for various impeller blade and casing tongue configurations (Reference 11)

isolate the skid from its support system. For best isolation, the lowest resonant frequency of the supported system should be well below the minimum operating frequency, and none of the higher resonant modes should be coincident with running speed or multiples thereof.

The application of elastomeric coatings to the exterior surface of the pump or piping to damp pipe wall vibrations is normally ineffective except on very thin conduit. However, such coatings may have a small acoustic effect in confining or absorbing high-frequency noise that would otherwise be radiated by the pipe wall vibrations. (See the following discussion on pipe wraps.)

Airborne Noise Although acoustic energy can be generated in the pumped liquid by purely fluid processes (turbulences and so on), most noise radiated to the surrounding air is the direct result of mechanical vibrations in the pump case, the pipe wall, or other structures to which the pump system is coupled by liquid or mechanical attachment. (The exceptions to purely mechanical sources are windage noise produced by the rotating coupling, by cooling air in the drive motors, and so on).

When excessive noise is encountered, the source or sources can sometimes be identified by making sound measurements at points in a grid around the suspect equipment and plotting sound level contours. A sound level contour of a typical flow control valve installation is shown in Figure 10. Octave band analyses or spectral analyses and contours can also provide clues by identifying the source of various frequency components of the noise. These components can also be compared with the information in Table 1 to aid in determining the source and location of the noise-generating mechanism.

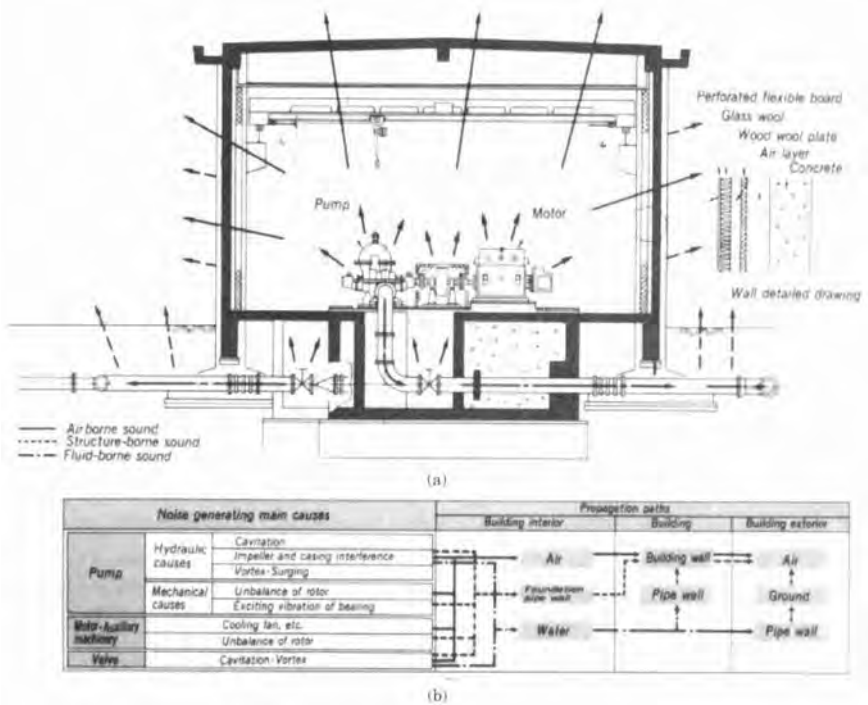


FIGURE 9 Noise propagation paths in a pumping plant (Reference 10)

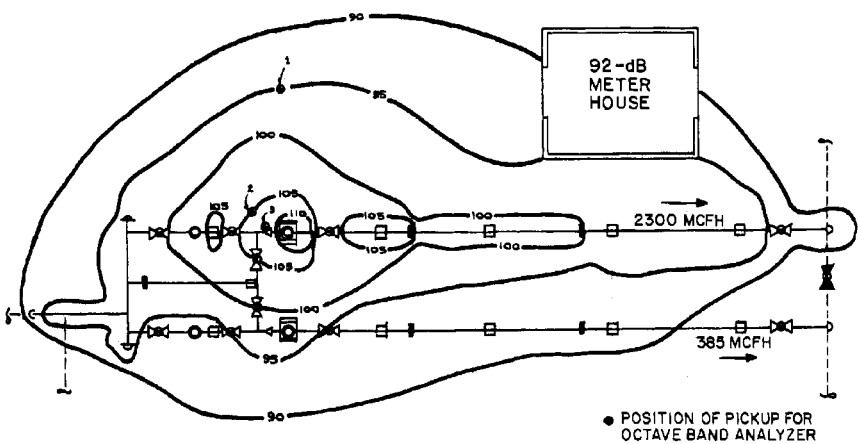


FIGURE 10 Contours of equal sound level (decibels) (Reference 14)

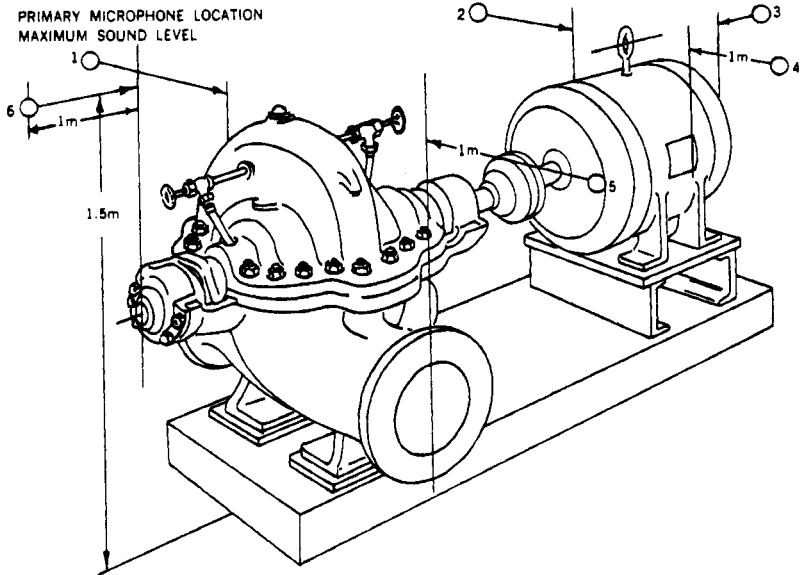


FIGURE 11 Placement of microphones on a horizontally split centrifugal pump (Hydraulic Institute ANSI/HI 2000 Edition Pump Standards, Reference 3)

The Hydraulic Institute standard³ gives specific details of the procedures for noise measurement around pumps, including microphone locations (Figure 11), measurement procedures, and a data sheet (Figure 12).

The approaches for reducing noise from pumps and piping systems after it is airborne generally consist of either interrupting the transmission path (barriers) or controlling the reverberation characteristics of the pump room.

A highly reflective (reverberant) pump room or enclosure can increase pump noise levels several decibels by reflections of the noise back and forth in the enclosure. The maximum reduction that can be achieved by the application of acoustic absorption material to the interior surfaces is normally about 10 dB for a highly reverberant enclosure. At most, such a treatment can reduce noise levels to those that would exist if the pump were operating in the open, totally free of reflecting surfaces. Quantitatively, the noise reduction NR in decibels that can be achieved is

$$NR = 10 \log \frac{\bar{\alpha}_a}{\bar{\alpha}_b} \quad (5)$$

where $\bar{\alpha}_a$ = average absorption coefficient of the surfaces after treatment

$\bar{\alpha}_b$ = average absorption coefficient of the surfaces before treatment

The average absorption coefficient is defined as follows:

$$\bar{\alpha} = \frac{\alpha_1 A_1 + \alpha_2 A_2 + \alpha_3 A_3 + \cdots + \alpha_n A_n}{A_1 + A_2 + A_3 + \cdots + A_n} \quad (6)$$

where $\alpha_1, \alpha_2, \alpha_3, \dots$ are absorption coefficients of various surface areas within the enclosure and $A_1, A_2, A_3, \dots$ are the corresponding surface areas.

Absorption coefficients for typical building materials are given in Table 6. Note that the absorption coefficients vary with frequency, and hence calculated values of NR can be quite frequency-sensitive. Absorption data for various absorbing materials can best be

AIRBORNE SOUND LEVEL TEST REPORT FOR PUMPING EQUIPMENT											REPORT FORM		
SUBJECT:													
Model: _____			Manufacturer: _____			Serial: _____							
Rated Pump Speed: _____			Capacity: _____			Total Head: _____							
Type of Driver: _____						Speed: _____							
Auxiliaries such as Gears: _____													
Applicable Figure No: _____													
Description: _____													
TEST CONDITIONS:													
Distance from Subject to Microphones: _____										Meters			
Operating Speed as Tested: _____													
Height of Microphone Above Reflecting Plane: _____										Meters			
Reflecting Plane Composition: _____													
Primary Microphone Location No: _____													
Remarks: _____													
INSTRUMENTATION:													
Microphone: _____										No. _____			
Sound Level Meter: _____										No. _____			
Octave Band Analyzer: _____										No. _____			
Calibrator: _____										No. _____			
Other: _____										No. _____			
DATA:													
db Ref. 2 x 10 ⁻¹⁰ N/m ²		BACK- GROUND	LOCATION *										AV.
			1	2	3	4	5	6	7	8	9	10	
dB A													
63													
125													
250													
500													
1k													
2k													
4k													
8k													
*Corrected for background sound. Readings having 3 dB corrections must be reported in brackets. Only octave bands of interest as defined on page 299 need be reported.													
TESTED BY: _____												DATE: _____	
REPORTED BY: _____												DATE: _____	

FIGURE 12 Hydraulic Institute data sheet for measurement of airborne sound from pumping equipment (Hydraulic Institute ANSI/HI 2000 Edition Pump Standards, Reference 3)

obtained from the suppliers. One of the most common and most effective of such materials is fiberglass matting or the more rigid fiberglass board.

The noise reduction that can typically be attained with fiberglass piping insulation is shown in Figure 13. Note that noise reduction varies with noise frequency and with the thickness and density of the fiberglass matting. Densities of 5 to 6 lb/ft³ (80 to 96 kg/m³), found in the rigid fiberglass board, are normally more effective than the 1 to 2 lb/ft³ (16 to 32 kg/m³) found in rolled matting.

Normally, the average absorption in a room must be increased by a factor of at least three before noise improvement is discernible to the ear. Unless the absorption coefficient of the untreated room is less than about 0.3 in the frequency range of maximum noise, the addition of absorbing material to the walls or ceiling will likely be ineffective. Nevertheless, the noise reduction could amount to 5 dB and thus could mean the difference between compliance and noncompliance with existing criteria.

The use of barriers near or enclosing the pump can be much more effective than environmental absorption treatments, particularly if the barriers totally confine the noise-producing components (Figure 14). Acoustic barriers simply interrupt the airborne path from the source to the listener. Such barriers work best (have the highest transmission loss) when

1. Their area density is high.
2. They are total enclosures and all joints are well sealed against air leaks.
3. They are not mechanically tied to the vibrating surfaces of the pump (that is, they are mechanically isolated).

TABLE 6 Sound absorption coefficients of common construction materials

Material	Frequency, Hz					
	125	250	500	1000	2000	4000
Brick						
Unglazed	0.03	0.03	0.03	0.04	0.04	0.05
Painted	0.01	0.01	0.02	0.02	0.02	0.02
Concrete block, painted	0.10	0.05	0.06	0.07	0.09	0.08
Concrete	0.01	0.01	0.015	0.02	0.02	0.02
Wood	0.15	0.11	0.10	0.07	0.06	0.07
Glass	0.35	0.25	0.18	0.12	0.08	0.04
Gypsum board	0.29	0.10	0.05	0.04	0.07	0.09
Plywood	0.28	0.22	0.17	0.09	0.10	0.11
Soundblox concrete block						
Type A (slotted), 6 in (15 cm)	0.62	0.84	0.36	0.43	0.27	0.50
Type B, 6 in (15 cm)	0.31	0.97	0.56	0.47	0.51	0.53
Carpet	0.02	0.06	0.14	0.37	0.60	0.65
Fiberglass typically 4 lb/ft ³ , (64 kg/m ³), hard backing						
1 in (2.5 cm) thick	0.07	0.23	0.48	0.83	0.88	0.80
2 in (5 cm) thick	0.20	0.55	0.89	0.97	0.83	0.79
4 in (10 cm) thick	0.39	0.91	0.99	0.97	0.94	0.89
Polyurethane foam, open-cell						
$\frac{1}{4}$ in (0.6 cm) thick	0.05	0.07	0.10	0.20	0.45	0.81
$\frac{1}{2}$ in (1.3 cm) thick	0.05	0.12	0.25	0.57	0.89	0.98
1 in (2.5 cm) thick	0.14	0.30	0.63	0.91	0.98	0.91
2 in (5 cm) thick	0.35	0.51	0.82	0.98	0.97	0.95
Hairfelt						
$\frac{1}{2}$ in (1.3 cm) thick	0.05	0.07	0.29	0.63	0.83	0.87
1 in (2.5 cm) thick	0.06	0.31	0.80	0.88	0.87	0.87

*For specific grades, see manufacturer's data; note that the term *NCR*, when used, is a single-term rating that is the arithmetic average of the absorption coefficients at 250, 500, 1000, and 2000 Hz.

Source: Reference 8.

For total enclosures, the barrier area density controls transmission loss for noise frequencies above the lowest mechanical flexural frequency of the barrier panels. Loosely sprung but well-sealed and heavy panels therefore serve this function well. Treating the interior of the enclosure with sound-absorbing material tends to reduce the apparent source strength by preventing reverberant buildups in the enclosure. Note that the use of a total enclosure around a pump may cause other operational problems that must be dealt with, for example, overheating, formation of explosive mixtures, or difficulty in getting to the pump for maintenance.

Although partial barriers around a pump usually cannot provide the degree of noise reduction afforded by total enclosures, they may be quite useful. Such barriers normally redirect the noise away from the listener and work best when they are located very near either the source or the listener. They also work better in an open or acoustically dead room than in a highly reverberant one. Their performance can often be enhanced by treating the interior (pump-side) surface with absorbing material or by reflecting the noise into an absorbing surface. The transmission loss of partial barriers is usually not critical as long as it is 20 dB or greater, as reverberation and flanking noise paths will normally limit overall reduction to less than 20 dB.

Exterior control techniques for pipe noise usually consist of pipe wrapping materials in addition to the resilient pipe supports and hangers already mentioned. The function of such materials is identical to that of total enclosures; hence, pipe wraps should be

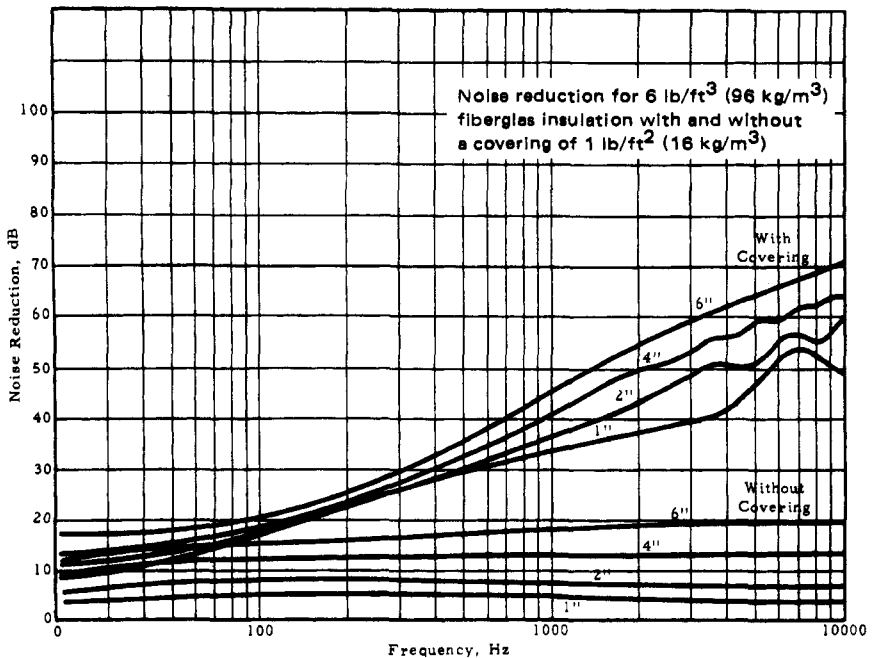


FIGURE 13 Noise reduction for piping with fiberglass insulation (1 in = 2.54 cm)

1. Massive (high area density)
2. Mechanically decoupled from the pipe (loosely sprung)
3. Airtight

For all the very-low-frequency noise, decoupling between a heavy, airtight outer coating and the vibrating pipe wall can be achieved by a compliant acoustic absorbing material, such as preformed fiberglass pipe insulation. This material provides some acoustic absorption

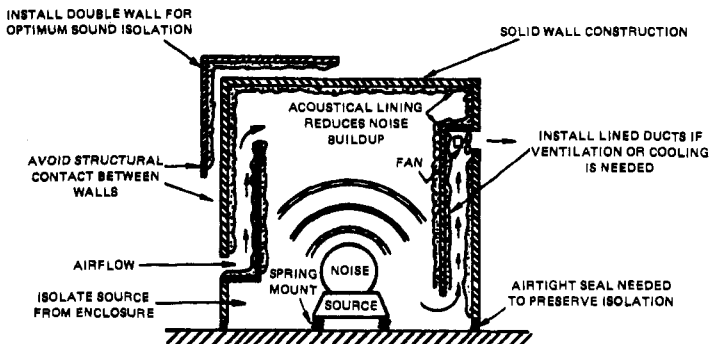


FIGURE 14 Design of an effective sound-isolating enclosure requiring air circulation for cooling (Reference 9)

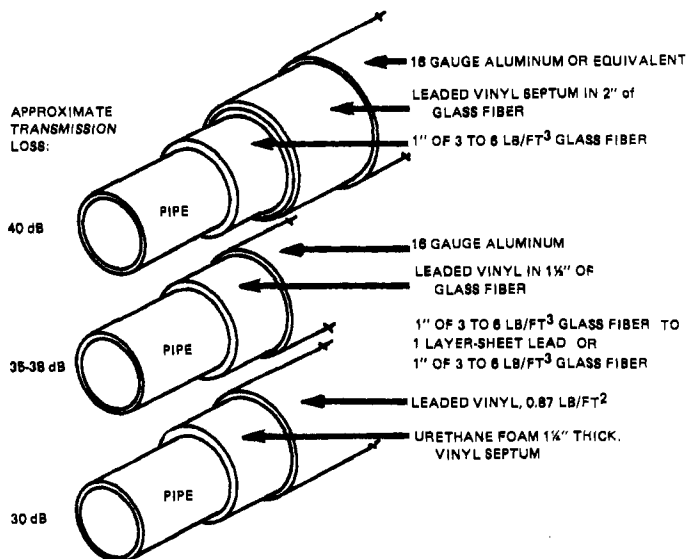


FIGURE 15 Three possible pipe wrap configurations (1 to = 2.54 cm; 1 lb/ft³ = 16 kg/m³) (Reference 14)

of noise as it "passes through," but, more important, it provides vibratory decoupling between the vibrating pipe wall and the external pipe wrap shell or layer.

Multiple-layered panels or pipe wrap normally provide more attenuation than single homogenous layers, as long as the layers are mechanically decoupled. Examples of several pipe wrap configurations are given in Figure 15. Alternative lead-free materials are replacing leaded vinyl.

CRITERIA

In order to decide whether noise treatment is necessary, the sound levels must be measured and compared with applicable criteria. Most protective noise criteria have been developed to protect a specific statistical sample of people from prescribed typical noise conditions and exposure patterns. Although such criteria are admittedly approximate, they are usually sufficiently conservative to protect a large majority of those who may be exposed, whether they are written to prevent hearing loss, community annoyance, speech and communication masking, or any of the other physiological or psychological effects of noise.

When noise criteria are legislated into noise codes or regulations, they normally specify noise measurement locations, instruments, and sampling times as well as allowable levels and exposure durations. Thus they have taken much of the guesswork out of evaluating machinery noise. The main criterion for pump and other machinery installations can be obtained from the U.S. Department of Labor, OSHA Bulletin 334. OSHA guidelines for allowable daily noise exposure as a function of time and sound pressure level for personnel near the installations are given in Table 7. The allowable exposure time depends upon the maximum sound level in the work space. When the noise exposure for personnel is intermittent, the allowable exposure may be calculated using Table 7 by computing the fractions of actual exposure time at a given noise level to the allowable exposure time at that level. If the sum of these fractions is less than 1, the noise level can be considered safe. It is important to know one underlying implication in this criterion. If a reduction of the

TABLE 7 OSHA noise exposure limits

Sound level, dBA	Allowable exposure time	
	Minutes	Hours
90	480	8.00
91	420	7.00
92	360	6.00
93	320	5.33
94	280	4.67
95	240	4.00
96	210	3.50
97	180	3.00
98	160	2.67
99	140	2.33
100	120	2.00
101	105	1.75
102	90	1.50
103	80	1.33
104	70	1.16
105	60	1.00
106	54	0.90
107	48	0.80
108	42	0.70
109	36	0.60
110	30	0.50
111	27	0.45
112	24	0.40
113	21	0.35
114	18	0.30
115 (maximum allowable level)	15	0.25

noise level is not economically feasible, it may be reasonable to schedule operators' work so the criterion levels will not be exceeded. This rescheduling may permit a wider latitude in the final solution of a plant noise problem.

MODEL TESTING

It is possible to conduct tests on reduced-size pumps and pumping systems in order to predict and adjust pressure pulsations and noise from the pump and piping. This has been successfully done by Sudo, Komatsu, and Kondo¹⁰ using a model variable-speed, single-stage, double-suction pump in addition to mathematical computer confirmation. A comparison of performance with nonstaggered and staggered impeller vanes revealed the good pulsation suppression characteristics of the latter impeller configuration (Figure 16).

These investigators, in their modeling, used the following relationships to provide the required similarity between model and prototype:

1. Equal ratios of pipe length to wavelength
2. Equal ratios of pipe cross-sectional area
3. Model pump speed adjusted to make the product of pulsation wavelength and frequency equal to the speed of sound in water

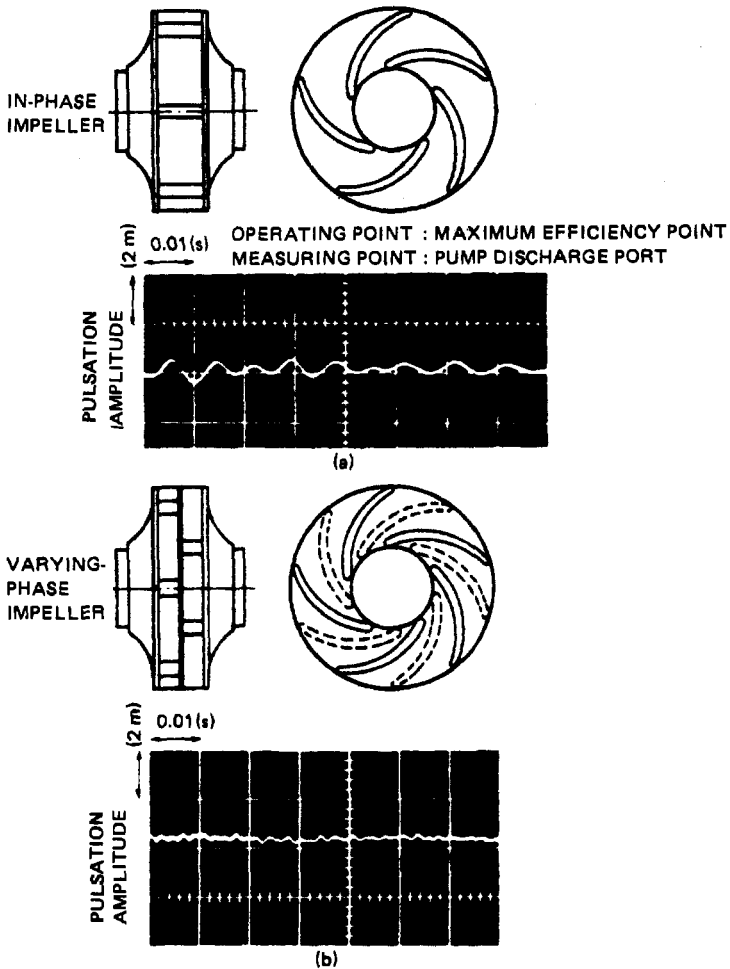


FIGURE 16 Comparison of pressure pulsation waveforms: (a) a double-suction in-phase centrifugal pump impeller and (b) a varying-phase (staggered) centrifugal pump impeller (Reference 10)

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C • H • A • P • T • E • R • 11

PUMP SYSTEMS

SECTION 11.1

GENERAL CHARACTERISTICS OF PUMPING SYSTEMS AND SYSTEM-HEAD CURVES

J. P. MESSINA

SYSTEM CHARACTERISTICS AND PUMP HEAD

A pump is used to deliver a specified rate of flow through a particular system. When a pump is to be purchased, this required capacity must be specified along with the total head necessary to overcome resistance flow and to meet the pressure requirements of the system components. The total head rating of a centrifugal pump is usually measured in feet (meters), and the differential pressure rating of a positive displacement pump is usually measured in pounds per square inch (kilopascals or bar¹). Both express, in equivalent terms, the work in foot-pounds (newton-meters) the pump is capable of doing on each unit weight (force) of liquid pumped at the rated flow.² It is the responsibility of the purchaser to determine the required pump total head so the supplier can make a proper pump selection. Underestimating the total head required will result in a centrifugal pump's delivering less than the desired flow through the system. An underestimate of the differential pressure required will result in a positive displacement pump's using more power than estimated, and the design pressure limit of the pump could be exceeded. Therefore, system pressure and resistance to flow, which are dependent on system characteristics, dictate the required pump head rating.

THE PUMPING SYSTEM

The piping and equipment through which the liquid flows to and from the pump constitute the pumping system. Only the length of the piping containing liquid controlled by the

¹1 bar = 10⁵ Pa.

²Work per unit weight mass, rather than weight force, is sometimes called *specific delivery work*; it has the units of newton-meters per kilogram and is equal to total head multiplied by *g*, the gravitation constant.

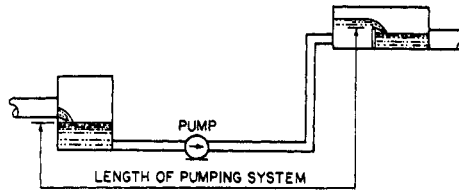


FIGURE 1 Length of system controlled by pump

action of the pump is considered part of the system. A pump and the limit of its system length are shown in Figure 1.

The pump suction and discharge piping can consist of branch lines, as shown in Figure 2. There can be more than one pump in a pumping system. Several pumps can be piped together in series or in parallel or both, as shown in Figure 3. When there is more than one pump, the flow through the system is determined by the combined performance of all the pumps.

The system through which the liquid is pumped offers resistance to flow for several reasons. Flow through pipes is impeded by friction. If the liquid discharges to a higher elevation or a higher pressure, additional resistance is encountered. The pump must therefore overcome the total system resistance due to friction and, as required, produce an increase in elevation or pressure at the desired rate of flow. System requirements may be such that the pump discharges to a lower elevation or pressure but additional pump head is still required to overcome pipe friction and obtain the desired rate of flow.

ENERGY IN AN INCOMPRESSIBLE LIQUID

The work done by a pump is the difference between the energy level at the point where the liquid leaves the pump and the energy level at the point where the liquid enters the pump. Work is also the amount of energy added to the liquid in the system. The total energy at any point in a pumping system is a relative term and is measured relative to some arbitrarily selected datum plane.

An incompressible liquid can have energy in the form of velocity, pressure, or elevation. Energy in various forms is added to the liquid as it passes through the pump, and the total energy added is constantly increasing with flow. It is appropriate then to speak of the energy added by a pump as the energy added per unit of weight (force) of the liquid pumped, and the units of energy expressed this way are foot-pounds per pound (newton-meters per newton) or just feet (meters). Therefore, when adding together the energies in their various forms at some point, it is necessary to express each quantity in common equivalent units of feet (meters) of *head*.

Liquid flowing in a conduit can undergo changes in energy form. Bernoulli's theorem for an incompressible liquid states that in steady flow, without losses, the energy at any point in the conduit is the sum of the *velocity head*, *pressure head*, and *elevation head* and

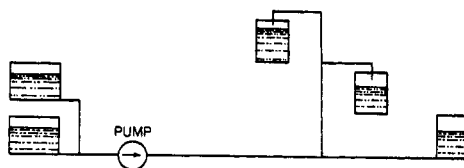


FIGURE 2 Branch-line pumping system

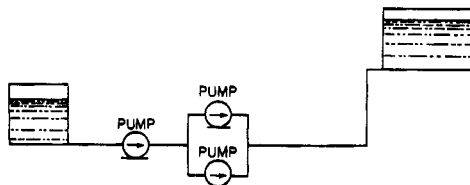


FIGURE 3 Pumps in series and in parallel

that this sum is constant along a streamline in the conduit. Therefore, the energy at any point in the system relative to a selected datum plane is

$$H = \frac{V^2}{2g} + \frac{p}{\gamma} + Z \quad (1)$$

where H = energy (total head) of system, ft · lb/lb or ft (N · m/N or m)

V = velocity, ft/s (m/s)

g = acceleration of gravity, 32.17 ft/s² (9.807 m/s²)

p = pressure, lb/ft² (N/m²)

γ = specific weight (force) of liquid, lb/ft³ (N/m³)

Z = elevation above (+) or below (–) datum plane, ft (m)

The velocity and pressure at the point of energy measurement are expressed in units of *equivalent head* and are added to the distance Z that this point is above or below the selected datum plane. If pressure is measured as gage (relative to atmosphere), total head H is gage; if pressure is measured as absolute, total head H is absolute. Equation 1 can also be applied to liquid at rest in a vertical column or in a large vessel (or to liquids of various densities) to account for changes in pressure with changes in elevation or vice versa.

The equivalent of velocity and pressure energy heads in feet (meters) can be thought of as the height to which a vessel of liquid of constant density has to be filled, above the point of measurement, to create this same velocity or pressure. This is illustrated in Figure 4 and further explained in the following text.

Velocity Head The kinetic energy in a mass of flowing liquid is $\frac{1}{2} mV^2$ or $\frac{1}{2} (W/g)V^2$. The kinetic energy per unit weight (force) of this liquid is $\frac{1}{2} (WV^2/Wg)$, or $V^2/2g$, measured in feet (meters). This quantity is theoretically equal to the equivalent static head of liquid that is required in a vessel above an opening if the discharge is to have a velocity equal to V . This is also the theoretical height the jet of liquid would rise if it were discharging vertically upward from an orifice.

A free-falling particle in a vacuum acquires the velocity V starting from rest after falling a distance H . Also

$$V = \sqrt{2gH}$$

All liquid particles moving with the same velocity have the same velocity head, regardless of specific weight. The velocity of liquid in pipes and open channels invariably varies across any one section of the conduit. However, when calculating system resistance it is sufficiently accurate to use the average velocity, computed by dividing the flow rate by the cross-sectional area of the conduit, when substituting in the term $V^2/2g$.

Pressure Head The pressure head, or flow work, in a liquid is p/γ , with units in feet (meters). A liquid, having pressure, is capable of doing work, for example, on a piston having an area A and stroke L . The quantity of liquid required to complete one stroke is γAL . The work (force × stroke) per unit weight (force) is $pAL/\gamma AL$, or p/γ . The work a pump

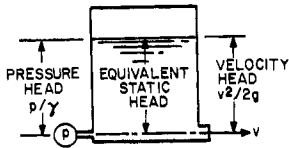


FIGURE 4 Equivalent static head

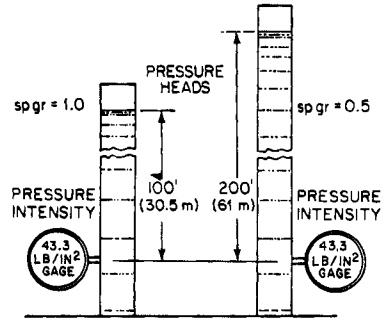


FIGURE 5 Liquids of different specific weights (also specific gravities) require different column heights or pressure heads to produce the same pressure intensity (43.3 lb/in² = 298.5 kPa)

must do to produce pressure intensity in liquids having different specific weights varies inversely with the specific weight or specific gravity of the liquid. Figure 5 illustrates this point for liquids having specific gravities of 1.0 and 0.5. The less dense liquid must be raised to a higher column height to produce the same pressure as the denser liquid. The pressure at the bottom of each liquid column H is the weight of the liquid above the point of pressure measurement divided by the cross-sectional area A at the same point, $AH\gamma/A$, which is simply $H\gamma$. Note that in this discussion A is in square feet (square meters) and L and H are in feet (meters).

The height of the liquid column in feet (meters) above the point of pressure measurement, if the column is of constant density, is equivalent pressure static head. When pressure intensity $AH\gamma/A$ is substituted for p in p/γ , it can be seen that the pressure head H is the liquid column height. Therefore, at the base of equal columns containing different liquids (with equal surface pressures), the pressure heads in feet (meters) are the same but the intensities in pounds per square foot (newtons per square meter) are different. For this reason, it is necessary to identify the liquids when comparing pressure heads.

Elevation Head The elevation energy, or potential energy, in a liquid is the distance Z in feet (meters) measured vertically above or below an arbitrarily selected horizontal datum plane. Liquid above a reference datum plane has positive potential energy because it can fall a distance Z and acquire kinetic energy or vertical head equal to Z . Also, it requires WZ ft · lb (N · m) of work to raise W lb (N) of liquid above the datum plane. The work per unit weight (force) of liquid is therefore WZ/W , or Z ft (m). In a pumping system, the energy required to raise a liquid above a reference datum plane can be thought of as being provided by a pump located at the datum elevation and producing a pressure that will support the total weight of the liquid in a pipe between the pump discharge and the point in the pipe to which the liquid is to be raised. This pressure is $AZ\gamma/A$, or simply $Z\gamma$ lb/ft² (N/m²) or $Z\gamma/144$ lb/in². Because head is equal to pressure divided by specific weight, elevation head is $Z\gamma/\gamma$, or Z ft.

Liquid below the reference datum plane has negative elevation head.

Total Head Figure 6 illustrates a liquid under pressure in a pipe. To determine the total head at the pressure gage connection and relative to the datum plane, Eq. 1 may be used (assume that the gage is at the pipe centerline):

$$H = \frac{V^2}{2g} + \frac{p}{\gamma} + Z$$

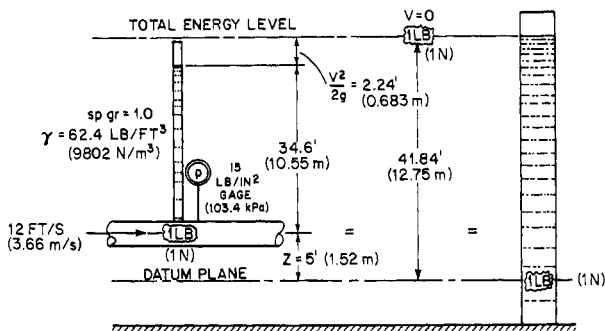


FIGURE 6 The total head, or energy, in foot-pounds per pound (newtonmeters per newton) is equal to the sum of the velocity, pressure, and elevation heads relative to a datum place. A unit weight (force) of the same liquid or any liquid raised to rest at the height shown, or under a column of liquid of this same height, has the same head as the unit weight (force) of liquid shown flowing in the pipe.

In USCS units

$$H = \frac{12^2}{2 \times 32.17} + \frac{144 \times 15}{62.4} + 5$$

$$= 2.24 + 34.6 + 5$$

$$= 41.84 \text{ ft} \cdot \text{lb/lb, or ft}$$

In SI units

$$H = \frac{3.66^2}{2 \times 9.807} + \frac{103.4 \times 1000}{9802} + 1.52$$

$$= 0.683 + 10.55 + 1.52$$

$$= 12.75 \text{ N} \cdot \text{m/N, or m}$$

The total head may also be calculated using the expression

$$H = \frac{V^2}{2g} + \text{manometer height} + Z$$

In USCS units

$$H = 2.24 + 34.6 + 5$$

$$= 41.84 \text{ ft} \cdot \text{lb/lb, or ft}$$

In SI units

$$H = 0.683 + 10.55 + 1.52$$

$$= 12.75 \text{ N} \cdot \text{m/N, or m}$$

The total head of 41.84 ft (12.75 m) is equivalent to 1 lb (N) of the liquid raised 41.84 ft (12.75 m) above the datum plane (zero velocity) or the pressure head of 1 lb (N) of the liquid under a column height of 41.84 ft (12.75 m) measured at the datum plane.

The gage pressure p , and consequently the pressure head, are measured relative to atmospheric pressure. Gage pressure head can therefore be a positive or a negative quantity. The pressure may also be expressed as an absolute pressure (measured from complete vacuum). Therefore, when velocity, pressure, and elevation heads are combined to obtain the total energy at a point, it should be clearly stated that the total head is either feet (meters) gage or feet (meters) absolute with respect to the datum plane.

The pressure or velocity of a liquid may at times be given as a pressure head of a liquid having a density different from the density of the liquid being pumped. In the total head, that is, the sum of the pressure, velocity, and elevation heads, the components must be corrected to be equal to the head of the liquid being pumped. For example, if the pressure is measured by a manometer to be 24 in (61 cm) of mercury (sp. gr. = 13.6) absolute,

the pressure head, or energy, in foot-pounds per pound (newton-meters per newton) of water pumped at 60°F (15.6°C) is found as follows.

Let subscripts 1 and 2 denote different liquids or, in this example, mercury and water, respectively:

$$\begin{aligned} h_1 &= \frac{p_1}{\gamma_1} \\ p_1 &= h_1 \gamma_1 = p_2 \\ h_2 &= \frac{p_2}{\gamma_2} = \frac{h_1 \gamma_1}{\gamma_2} \\ &= \frac{\gamma_1}{\gamma_2} h_1 \end{aligned} \quad (2)$$

$$= \frac{\text{sp. gr.}_1}{\text{sp. gr.}_2} h_1 \quad (3)$$

Therefore

$$\text{in USCS units} \quad h_2 = \frac{13.6}{1} \times \frac{24}{12} = 27.2 \text{ ft abs}$$

$$\text{in SI units} \quad h_2 = \frac{13.6}{1} \times \frac{61}{100} = 8.3 \text{ m abs}$$

Also, $h_2 = (27.2 - 13.6)\left(\frac{30}{12}\right) = -6.8 \text{ ft}$ [$(8.3 - 13.6)\left(\frac{76}{100}\right) = -2.04 \text{ m}$] gage if corrected to a standard barometer of 30 in (76 cm) of mercury.

PUMP TOTAL HEAD

The total head of a pump is the difference between the energy level at the pump discharge (point 2) and that at the pump suction (point 1), as shown in Figures 7 and 8. Applying Bernoulli's equation (Eq. 1) at each point, the pump total head TH in feet (meters) becomes

$$TH = H_d - H_s = \left(\frac{V_d^2}{2g} + \frac{p_d}{\gamma_d} + Z_d \right) - \left(\frac{V_s^2}{2g} + \frac{p_s}{\gamma_s} + Z_s \right) \quad (4)$$

The equation for pump differential pressure P_Δ in pounds per square foot (newtons per square meter) is

$$P_\Delta = P_d - P_s = \left[p_d + \gamma_d \left(Z_d + \frac{V_d^2}{2g} \right) \right] - \left[p_s + \gamma_s \left(Z_s + \frac{V_s^2}{2g} \right) \right] \quad (5)$$

$$P_\Delta \text{ in lb/in}^2 = P_\Delta \text{ in lb/ft}^2 \div 144$$

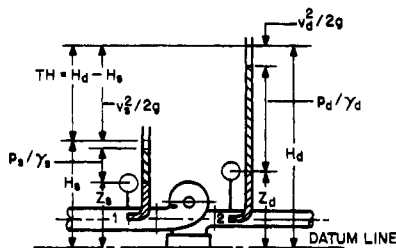


FIGURE 7 Centrifugal pump total head

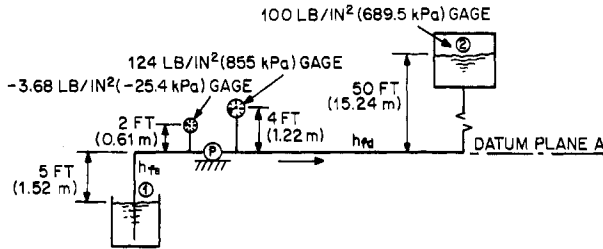


FIGURE 8 Example 1

where the subscripts d and s denote discharge and suction, respectively, and

H = total head of system, (+) or (-) ft (m) gage or (+) ft (m) abs

P = total pressure of system, (+) or (-) lb/ft² (N/m²) gage or (+) lb/ft² (N/m²) abs

V = velocity, ft/s (m/s)

p = pressure, (+) or (-) lb/ft² (N/m²) gage or (+) lb/ft² (N/m²) abs

Z = elevation above (+) or below (-) datum plane, ft (m)

γ = specific weight (force) of liquid, lb/ft³ (N/m³)

g = acceleration of gravity, 32.17 ft/s² (9.807 m/s²)

Pump total head TH and pump differential pressure P_d are always absolute quantities because either gage pressures or absolute pressures but not both are used at the discharge and suction connections of the pump and a common datum plane is selected.

Pump total head in feet (meters) and pump differential pressure in pounds per square foot (newtons per square meter) are related to each other as

$$TH = \frac{P_d}{\gamma} \quad (6)$$

It is very important to note that, if the rated total head of a centrifugal pump is given in feet (meters), this head can be imparted to *all* individual liquids pumped at the rated capacity and speed, regardless of the specific weight (force) of the liquids as long as their viscosities are approximately the same. A pump handling different liquids of approximately the same viscosity will generate the same total head but will not produce the same differential pressure, nor will the power required to drive the pump be the same. On the other hand, a centrifugal pump rated in pressure units would have to have a different pressure rating for each liquid of different specific weight (force). In this section, pump total head will be expressed in feet (meters), the usual way of rating centrifugal pumps. For an explanation of positive displacement pump differential pressure, its use and relationship to pump total head, see Chap. 3.

Pump total head can be measured by installing gages at the pump suction and discharge connections and then substituting these gage readings into Eq. 4. Pump total head may also be found by measuring the energy difference between any two points in the pumping system, one on each side of the pump, providing all losses (other than pump losses) between these points are credited to the pump and added to the energy head difference. Therefore, between any two points in a pumping system where the energy is added only by the pump and the specific weight (force) of the liquid does not change (for example, as a result of temperature), the following general equation for determining pump total head applies:

$$\begin{aligned} TH &= (H_2 - H_1) + \Sigma h_{f(1-2)} \\ &= \left(\frac{V_2^2}{2g} + \frac{p_2}{\gamma} + Z_2 \right) - \left(\frac{V_1^2}{2g} + \frac{p_1}{\gamma} + Z_1 \right) + \Sigma h_{f(1-2)} \end{aligned} \quad (7)$$

where the subscripts 1 and 2 denote points in the pumping system anyplace upstream and downstream from the pump, respectively, and

H = total head of system, (+) or (-) ft (m) gage or (+) ft (m) abs

V = velocity, ft/s (m/s)

p = pressure, (+) or (-) lb/in² (N/m²) gage or (+) lb/in² (N/m²) abs

Z = elevation above (+) or below (-) datum plane, ft (m)

γ = specific weight (force) of liquid (assumed the same between points), lb/ft³ (N/m³)

g = acceleration of gravity, 32.17 ft/s² (9.807 m/s²)

Σh_f = sum of piping losses between points, ft (m)

When the specific gravity of the liquid is known, the pressure head may be calculated from the following relationships:

$$\text{In feet} \quad \frac{p}{\gamma} = \frac{0.016 \text{ lb/ft}^2}{\text{sp. gr.}} \text{ or } \frac{2.3 \text{ lb/in}^2}{\text{sp. gr.}} \quad (8a)$$

$$\text{In meters} \quad \frac{p}{\gamma} = \frac{0.102 \text{ kPa}}{\text{sp. gr.}} \text{ or } \frac{1.02 \times 10^{-3} \text{ bar}}{\text{sp. gr.}} \quad (8b)$$

The velocity in a pipe may be calculated as follows:

$$\text{In feet per second} \quad V = \frac{(\text{gpm})(0.408)}{(\text{pipe ID in inches})^2} \quad (9a)$$

$$\text{In meters per second} \quad V = \frac{(\text{m}^3/\text{h})(3.54)}{(\text{pipe ID in cm})^2} \text{ or } \frac{(\text{liters/s})(12.7)}{(\text{pipe ID in cm})^2} \quad (9b)$$

The following example illustrates the use of Eqs. 4 and 7 for determining pump total head.

EXAMPLE 1 A centrifugal pump delivers 1000 gpm (227 m³/hr) of liquid of specific gravity 0.8 from the suction tank to the discharge tank through the piping shown in Figure 8. (a) Calculate pump total head using gages and the datum plane selected. (b) Calculate total head using the pressures at points 1 and 2 and the same datum plane as (a).

Given: Suction pipe ID = 8 in (203 mm), discharge pipe ID = 6 in (152 mm), h_f = pipe, valve, and fitting losses, h_{fs} = 3 ft (0.91 m), h_{fd} = 25 ft (7.62 m).

$$\text{In USCS units} \quad \text{Calculated pipe velocity} = \frac{(\text{gpm})(0.408)}{(\text{ID in inches})^2}$$

$$V_s = \frac{1000 \times 0.408}{8^2} = 6.38 \text{ ft/s}$$

$$V_d = \frac{1000 \times 0.408}{6^2} = 11.33 \text{ ft/s}$$

$$\text{In SI units} \quad \text{Calculated pipe velocity} = \frac{(\text{m}^3/\text{h})(3.54)}{(\text{ID in cm})^2}$$

$$V_s = \frac{227 \times 3.54}{20.3^2} = 1.95 \text{ m/s}$$

$$V_d = \frac{227 \times 3.54}{15.2^2} = 3.48 \text{ m/s}$$

(a) From Eq. 4,

$$TH = \left(\frac{V_d^2}{2g} + \frac{p_d}{\gamma_d} + Z_d \right) - \left(\frac{V_s^2}{2g} + \frac{p_s}{\gamma_s} + Z_s \right)$$

and Eq. 8,

$$\text{in USCS units} \quad \frac{p}{\gamma} = \frac{2.31 \text{ lb/in}^2}{\text{sp. gr.}}$$

$$\text{in SI units} \quad \frac{p}{\gamma} = \frac{0.102 \text{ kPa}}{\text{sp. gr.}}$$

Therefore,

in USCS units

$$\begin{aligned} TH &= \left(\frac{11.33^2}{2 \times 32.3} + \frac{2.31 \times 124}{0.8} + 4 \right) - \left(\frac{638^2}{2 \times 32.2} + \frac{2.31(-3.68)}{0.8} + 2 \right) \\ &= 364 - (-8) = 372 \text{ ft} \cdot \text{lb/lb, or ft} \end{aligned}$$

In SI units

$$\begin{aligned} TH &= \left(\frac{3.48^2}{2 \times 9.807} + \frac{0.102 \times 855}{0.8} + 1.22 \right) - \left(\frac{195^2}{2 \times 9.807} + \frac{0.102(-25.4)}{0.8} + 2 \right) \\ &= 110.9 - (-2.43) = 113.3 \text{ N} \cdot \text{m/N, or m} \end{aligned}$$

(b) from Eq. 7,

$$TH = \left(\frac{V_2^2}{2g} + \frac{p_2}{\gamma} + Z_2 \right) - \left(\frac{V_1^2}{2g} + \frac{p_1}{\gamma} + Z_1 \right) + \Sigma h_{f(1-2)}$$

and Eq. 8,

$$\text{in USCS units} \quad \frac{p}{\gamma} = \frac{2.31 \text{ lb/in}^2}{\text{sp. gr.}}$$

$$\text{in SI units} \quad \frac{p}{\gamma} = \frac{0.102 \text{ kPa}}{\text{sp. gr.}}$$

Therefore

$$\begin{aligned} \text{in USCS units} \quad TH &= \left(0 + \frac{2.31 \times 100}{0.8} + 50 \right) - (0 + 0 - 5) + (3 + 25) \\ &= 399 - (-5) + 28 = 372 \text{ ft} \cdot \text{lb/lb, or ft} \end{aligned}$$

in SI units

$$TH = \left(0 + \frac{0.102 \times 689.5}{0.8} + 15.24 \right) - (0 + 0 - 1.52) + (0.91 + 7.62)$$

$$= 103.2 - (-1.52) + 8.53 = 113.3 \text{ N} \cdot \text{m/N, or m}$$

ENERGY AND HYDRAULIC GRADIENT

The total energy at any point in a pumping system may be calculated for a particular rate of flow using Bernoulli's equation (Eq. 1). If some convenient datum plane is selected and the total energy, or head, at various locations along the system is plotted to scale, the line drawn through these points is called the *energy gradient*. Figure 9 shows the variation in total energy H measured in feet (meters) from the suction liquid surface point 3 to the discharge liquid surface point 4. A horizontal energy gradient indicates no loss of head.

The line drawn through the sum of the pressure and elevation heads at various points represents the pressure variation in flow measured above the datum plane. It also represents the height the liquid would rise in vertical columns relative to the datum plane when the columns are placed at various locations along pipes having positive pressure anywhere in the system. This line, shown dotted in Figure 9, is called the *hydraulic gradient*. The difference between the energy gradient line and the hydraulic gradient line is the velocity head in the pipe at that point.

The pump total head is the algebraic difference between the total energy at the pump discharge (point 2) and the total energy at the pump suction (point 1).

SYSTEM-HEAD CURVES

A pumping system may consist of piping, valves, fittings, open channels, vessels, nozzles, weirs, meters, process equipment, and other liquid-handling conduits through which flow is required for various reasons. When a particular system is being analyzed for the purpose

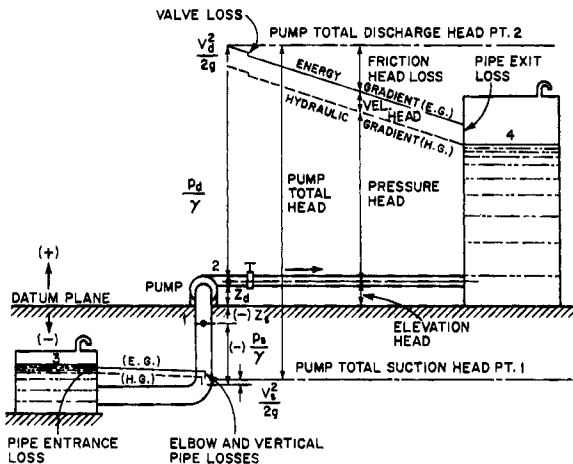


FIGURE 9 Energy and hydraulic gradients

of selecting a pump or pumps, the resistance to flow of the liquid through these various components must be calculated. It will be explained in more detail later in this section that the resistance increases with flow at a rate approximately equal to the square of the flow through the system. In addition to overcoming flow resistance, it may be necessary to add head to raise the liquid from suction level to a higher discharge level. In some systems the pressure at the discharge liquid surface may be higher than the pressure at the suction liquid surface, a condition that requires more pumping head. The latter two heads are *fixed system heads*, as they do not vary with rate of flow. Fixed system heads can also be negative, as would be the case if the discharge level elevation or the pressure above that level were lower than suction elevation or pressure. Fixed system heads are also called *static heads*.

A system-head curve is a plot of total system resistance, variable plus fixed, for various flow rates. It has many uses in centrifugal pump applications. It is preferable to express system head in feet (meters) rather than in pressure units because centrifugal pumps are rated in feet (meters), as previously explained. System-head curves usually show flow in gallons per minute, but when large quantities are involved, the units of cubic feet per second or million gallons per day are used. Although the standard SI units for volumetric flow are cubic meters per second, the units of cubic meters per hour are more common.

When the system head is required for several flows or when the pump flow is to be determined, a system-head curve is constructed using the following procedure. Define the pumping system and its length. Calculate (or measure) the fixed system head, which is the net change in total energy from the beginning to the end of the system due to elevation or pressure head differences. An increase in head in the direction of flow is a positive quantity. Next, calculate, for several flow rates, the variable system total head loss through all piping, valves fittings, and equipment in the system. As an example, see Figure 10, in which the pumping system is defined as starting at point 1 and ending at point 2. The fixed system head is the net change in total energy. The total head at point 1 is p_s/γ , and that at point 2 is $p_d/\gamma + Z$. The pressure and liquid levels do not vary with flow. The variable system head is pipe friction (including valves and fittings). The fixed head and variable heads for several flow rates are added together, resulting in a curve of total system head versus flow.

The flow produced by a centrifugal pump varies with the system head, whereas the flow of a positive displacement pump is independent of the system head. By superimposing the head-capacity characteristic curve of a centrifugal pump on a system-head curve, as shown in Figure 10, the flow of a pump can be determined. The curves will intersect at the flow rate of the pump, as this is the point at which the pump head is equal to the required system

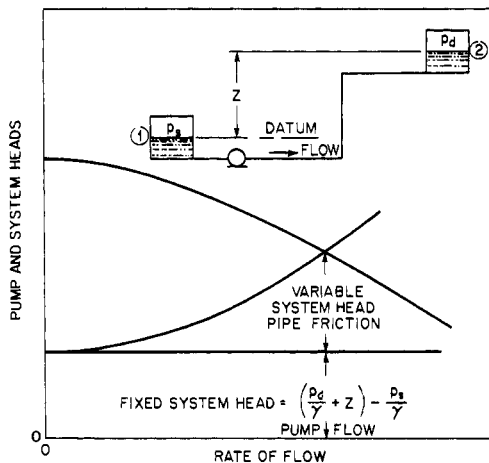


FIGURE 10 Construction of system total-head curve

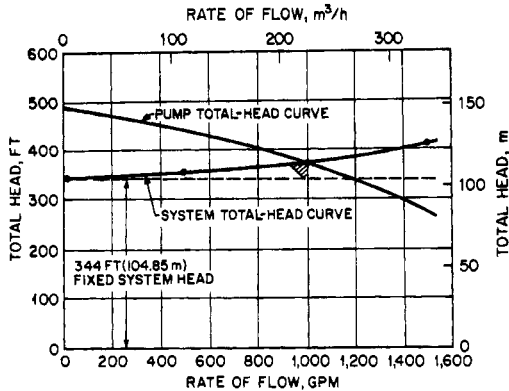


FIGURE 11 System total-head curve for Example 1

head for the same flow. When a pump is being purchased, it should be specified that the pump head-capacity curve intersect the system-head curve at the desired flow rate. This intersection should be at the pump's best efficiency capacity or very close to it.

The system-head curve for Example 1 is shown in Figure 11. This assumes that the suction and discharge liquid levels are 5 ft (1.5 m) below and 50 ft (15 m) above the datum plane, respectively, and do not vary with flow. The pressure in the discharge tank is also independent of flow and is 100 lb/in² (689.5 kPa) gage. These values are therefore fixed system heads. The pipe and fitting losses are assumed to vary with flow as a square function. The length of the pumping system is from point 1 to point 2. The difference in heads at these points plus the frictional losses at various flow rates are the total system head and the head required by a pump for the different flows. It is necessary to calculate the total system head for only one flow rate—say, design—which in this example is 1000 gpm (227 m³/h). The total head at other flow conditions is the fixed system head plus the variable system head multiplied by $(\text{gpm}/1000)^2$ ($\text{m}^3/\text{h} \div 227$)². If Example 1 is an existing system, the total head may be calculated by using gages at the pump suction and discharge connections. The total head measured will then be the head at the intersection of the pump and system curves, as shown in Figure 11. In this example, a correctly purchased pump would produce a total head of 372 ft (113 m) at the design flow of 1000 gpm (227 m³/h).

In systems that are open-ended and in which there is a decrease in elevation from inlet to outlet, a portion of the system-head curve will be negative (Figure 12). In this example, the pump is used to increase gravity flow. Without a pump in the system, the negative resistance, or static head, is the driving head that moves the liquid through the system. Steady-state gravity flow is sustained at the flow rate corresponding to zero total system head (negative static head plus system resistance equals zero). If a flow is required at any rate greater than that which gravity can produce, a pump is required to overcome the additional system resistance.

For additional information concerning the construction of system-head curves for flow in branch lines, refer to Section 11.2.

VARIANTS IN PUMPING SYSTEMS

For a fixed set of conditions in a pumping system, there is just one total head for each flow rate. Consequently, a centrifugal pump operating at a constant speed can deliver just one flow. In practice, however, conditions in a system vary as a result of either controllable or uncontrollable changes. Changes in the valve opening in the pump discharge or bypass line, changes in the suction or discharge liquid level, changes in the pressures at these levels, the aging of pipes, changes in the process, changes in the number of pumps pumping

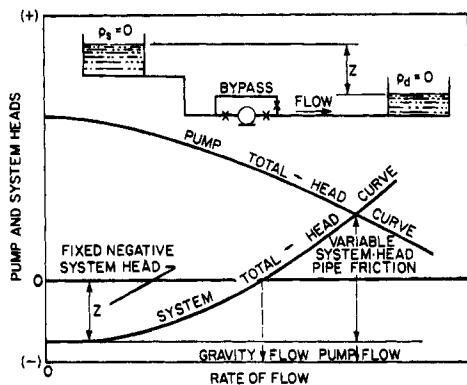


FIGURE 12 Construction of system total-head curve to determine gravity flow and centrifugal pump flow

into a common header, changes in the size, length, or number of pipes are all examples of either controllable or uncontrollable system changes. These changes in system conditions alter the shape of the system-head curve and, in turn, affect pump flow.

Methods of constructing system-head curves and determining the resultant pump flows for two of the more common of these variants are explained here.

Variable Static Head In a system where a pump is taking suction from one reservoir and filling another, the capacity of a centrifugal pump will decrease with an increase in static head. The system-head curve is constructed by plotting the variable system friction head versus flow for the piping. To this is added the anticipated minimum and maximum static heads (difference in discharge and suction levels). The resulting two curves are the total system heads for each condition. The flow rate of the pump is the point of intersection of the pump head-capacity curve with either one of the latter two system-head curves or with any intermediate system-head curve for other level conditions. A typical head versus flow curve for a varying static head system is shown in Figure 13.

If it is desired to maintain a constant pump flow for different static head conditions, the pump speed can be varied to adjust for an increase or decrease in the total system head. A typical variable-speed centrifugal pump operating in a varying static head system can have a constant flow, as shown in Figure 14.

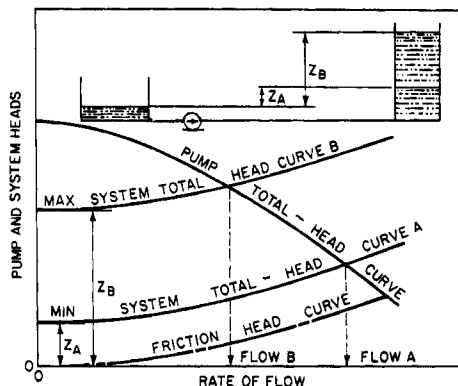


FIGURE 13 Construction of system total-head curves for a pumping system having variable static head

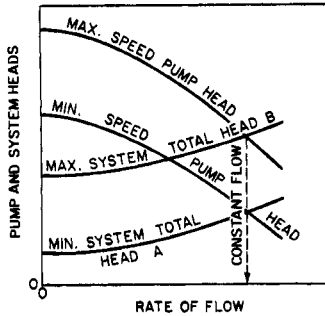


FIGURE 14 Varying centrifugal pump speed to maintain constant flow for the different reservoir levels shown in Figure 13

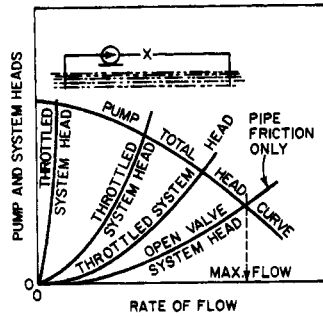


FIGURE 15 Construction of system total-head curves for various valve openings

It is important to select a pump that will have its best efficiency within the operating range of the system and preferably at the condition at which the pump will operate most often.

Variable System Resistance A valve or valves in the discharge line of a centrifugal pump alter the variable frictional head portion of the total system-head curve and consequently the pump flow. Figure 15, for example, illustrates the use of a discharge valve to change the system head for the purpose of varying pump flow during a shop performance test. The maximum flow is obtained with a completely open valve, and the only resistance to flow is the friction in the piping, fittings, and flowmeter. A closed valve results in the pump's operating at shutoff conditions and produces maximum head. Any flow between maximum and shutoff can be obtained by proper adjustment of the valve opening.

DIVIDING TOTAL HEAD AND SYSTEM-HEAD CURVES FOR CENTRIFUGAL PUMPS IN SERIES

Pump limitations or system component requirements may determine that two or more pumps must be used in series. There are practical limitations as to the maximum head that can be developed in a single pump, even if multistaged. When pumping through several system components, there may be pressure limitations that prevent using a single pump to develop all of the head required at the beginning of the system. If several pumps are to be used in series, how should the total head be divided among them?

The sum of the total heads of the pumps must be equal to the required total system head at the design flow. Although mathematically any division of the total head among the pumps to be used is possible (at long as the sum of the pump heads is equal to the total system head), the actual pressure required at various locations along the system flow path determines how the pumping heads are to be divided. An energy or pressure gradient should be drawn for the system. The number of pumps, their locations, and their total heads should be selected to produce the desired pressures (or range of pressures) at critical locations along the system. In addition to considering the pressure loss through components to overcome resistance to flow, consideration should also be given to the minimum pressures required to prevent flashing in piping, cavitation at pump inlets, and so on, as well as the maximum working pressures for different parts of the system.

If preferred, the total system can be divided into subsystems, one for each pump (or group of pumps). The end of one subsystem and the beginning of another can be selected anywhere between pumps in series because the pump total head will be unaffected by the division line. Consequently, several system-head curves can be drawn for specification and

purchasing purposes—for example, primary condensate pump system, secondary condensate pump system, feed pump system, in a total power plant system.

TRANSIENTS IN SYSTEM HEADS

During the starting of a centrifugal pump and prior to the time normal flow is reached, certain transient conditions can produce or require heads and consequently torques much higher than design. In some cases, the selection of the driver and the pump must be based on starting rather than normal flow conditions.

Low- and medium-specific-speed pumps of the radial- and mixed-flow types (less than approximately 5000 specific speed, rpm, gpm, ft units) have favorable starting characteristics. The pump head at shutoff is not significantly higher than that at normal flow, and the shutoff torque is less than that at normal flow. High-specific-speed pumps of the mixed- and axial-flow types (greater than approximately 5000 specific speed) develop relatively high shutoff heads, and their shutoff torque is greater than that at normal flow. These characteristics of high-specific-speed pumps require special attention during the starting period. Characteristics of pumps of different specific speeds are shown in Figures 16a and 16b.

Starting Against a Closed Valve When any centrifugal pump is started against a closed discharge valve, the pump head will be higher than normal. The shutoff head will vary with pump specific speed. As shown in Figure 16a, the higher the specific speed, the higher the shutoff head in percent of normal pump head. As a pump is accelerated from rest to full speed against a closed valve, the head on the pump at any speed is equal to the square of the ratio of the speed to the full speed times the shutoff head at full speed. Therefore, during starting, the head will vary from point *A* to point *E* in Figure 17. Points *B*, *C*, and *D* represent intermediate heads at intermediate speeds. The pump, the discharge valve, and any intermediate piping must be designed for maximum head at point *E*.

Pumps requiring less shutoff power and torque than at normal flow condition are usually started against a closed discharge valve. To prevent backflow from a static discharge head prior to starting, either a discharge shutoff valve, a check valve, or a broken siphon is required. When pumps are operated in parallel and are connected to a common discharge header that would permit flow from an operating pump to circulate back through an idle pump, a discharge valve or check valve must be used.

Figure 18 is a typical characteristic curve for a low-specific-speed pump. Figure 19 illustrates the variation of torque with pump speed when the pump is started against a closed discharge valve. The torque under shutoff conditions varies as the square of the ratio of speeds, similar to the variation in shutoff head, and is shown as curve *ABC*. At zero speed, the pump torque is not zero as a result of static friction in the pump hearings and stuffing box or boxes. This static friction is greater than the sum of running friction and power input to the impeller at very low speeds, which explains the dip in the pump torque curve between 0 and 10% speed. Also shown in Figure 19 is the speed-torque curve of a typical squirrel-cage induction motor. Note that the difference between motor and pump torque is the excess torque available to accelerate the pump from rest to full speed. During acceleration, the pump shaft must transmit not only the pump torque (curve *ABC*) but also the excess torque available in the motor. Therefore pump shaft torque follows the motor speed-torque curve less the torque required to accelerate the mass inertia (WK^2) of the motor's rotor.

High-specific-speed pumps, especially propeller pumps, requiring more than normal torque at shutoff are not normally started with a closed discharge valve because larger and more expensive drivers would be required. These pumps will also produce relatively high pressures in the pump and in the system between pump and discharge valve. Figure 20 is a typical characteristic curve for a high-specific-speed pump. Curve *ABC* of Figure 21 illustrates the variation of torque with speed when this pump is started against a closed discharge valve. A typical speed-torque curve of a squirrel-cage induction motor sized for normal pump torque is also shown. Note that the motor has insufficient torque to accelerate

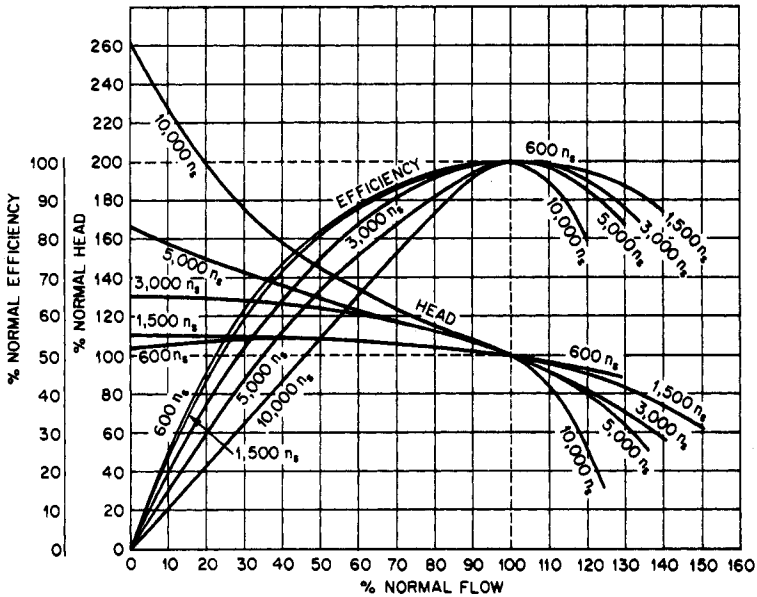


FIGURE 16A Approximate comparison of head and efficiency versus flow for impellers of different specific speeds in single-stage volute pumps. Specific speed $n_s = \text{rpm} \sqrt{\text{gpm}/\text{TH}}$ in $\text{ft}^{3/4}$
To convert to other units using

rpm, m^3/s , m: multiply by 0.01936; rpm, m^3/h , m: multiply by 1.163; rpm, L/s, m: multiply by 0.6123

To convert to the universal specific speed Ω , (defined in Section 2.1.1) divide n_s by 2733.

to full speed and would remain overloaded at point *C* until the discharge valve on the pump was opened. To avoid this situation, the discharge valve should be timed to open sufficiently to keep the motor from overloading when the pump reaches full speed. To accomplish this timing, it may be necessary to start opening the valve in advance of energizing the motor. Care should be taken not to start opening the discharge valve too soon because

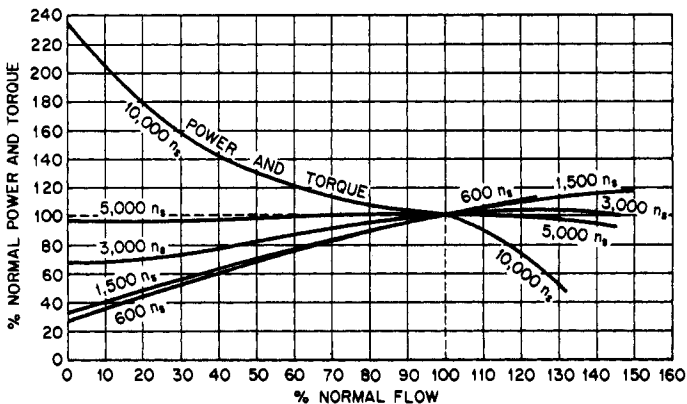


FIGURE 16B Approximate comparison of power and torque versus flow for impellers of different specific speeds in single-stage volute pumps

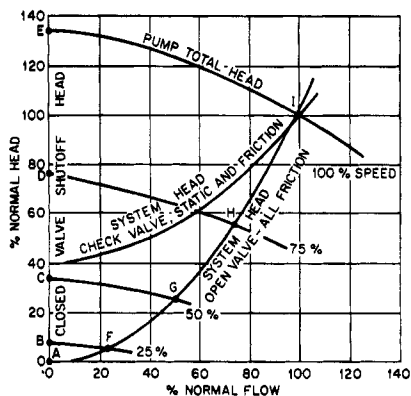


FIGURE 17 Variation in head when a centrifugal pump is started with a closed valve, an open valve, and a check valve

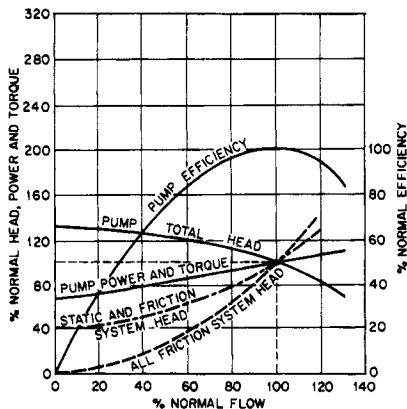


FIGURE 18 Typical constant-speed characteristic curves for a low-specific-speed pump

this would cause excessive reverse flow through the pump and require the motor to start under adverse reverse-speed conditions. If the driver is a synchronous motor, additional torque at pull-in speed is required, over and above that needed to overcome system head and accelerate the pump and driver rotors from rest. At the critical pull-in point, sufficient torque must be available to pull the load into synchronism in the prescribed time. A synchronous motor is started on low-torque, squirrel-cage windings prior to excitation of the field windings at the pull-in speed. The low starting torque, the torque required at pull-in, and possible voltage drop, which will lower the motor torque (varies as the square of the voltage), must all be taken into consideration when selecting a synchronous motor to start a high-specific-speed pump against a closed valve.

If a high-specific-speed pump is to be started against a closed discharge valve, high starting torques can also be avoided by the use of a bypass valve (see Section 11.2) or by an adjustable-blade pump.¹

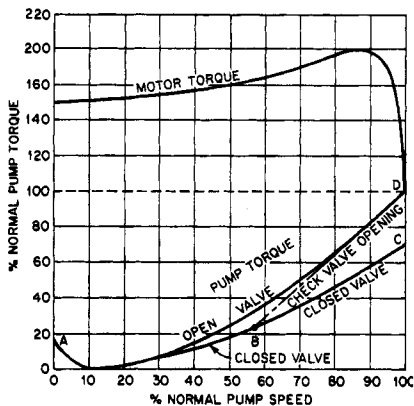


FIGURE 19 Variation of torque during start-up of a loss-specific-speed pump with a closed valve, an open valve, and a check valve. See Figure 18 for pump characteristics.

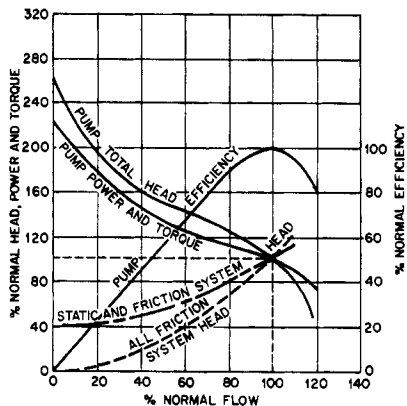


FIGURE 20 Typical constant-speed characteristic curves for a high-specific-speed pump

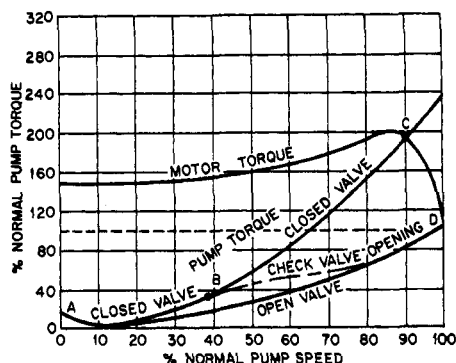


FIGURE 21 Variation of torque during start-up of a high-specific-speed pump with a closed valve, an open valve, and a check valve. See Figure 20 for pump characteristics.

Starting Against a Check Valve A check valve can be used to prevent reverse flow from static head or head from other pumps in the system. The check valve will open automatically when the head from the pump exceeds system head. When a centrifugal pump is started against a check valve, pump head and torque follow shutoff values until a speed is reached at which shutoff head exceeds system head. As the valve opens, the pump head continues to increase, and, at any flow, the head will be that necessary to overcome system static head or head from other pumps, frictional head, valve head loss, and the inertia of the liquid being pumped.

Figure 19, curve *ABD*, illustrates speed-torque variation when a low-specific-speed pump is started against a check valve with static head and system friction as shown in Figure 18. Figure 21, curve *ABD*, illustrates speed-torque variation for a high-specific-speed pump started against a check valve with static head and system friction as shown in Figure 20. The use of a quick-opening check valve with high-specific-speed pumps eliminates starting against higher than full-open valve shutoff heads and torques.

The speed-torque curves shown for the period during the acceleration of the liquid in the system have been drawn with the assumption that the head required to accelerate the liquid and overcome inertia is insignificant. Acceleration head is discussed in more detail later.

Starting Against an Open Valve If a centrifugal pump is to take suction from a reservoir and discharge to another reservoir having the same liquid elevation or the same equivalent total pressure, it can be started without a shutoff discharge valve or check valve. The system-head curve is essentially all frictional plus the head required to accelerate the liquid in the system during the starting period. Neglecting liquid inertia, the pump head would not be greater than normal at any speed during the starting period, as shown in curve *AFGHI* of Figure 17. Pump torque would not be greater than normal at any speed during the starting period, as shown in Figures 19 and 21, curves *AD*. Pump head and torque at any speed are equal to their values at normal condition times the square of the ratio of the speed to full speed, whereas the capacity varies directly with this ratio.

Starting a Pump Running in Reverse When a centrifugal pump discharges against a static head or into a common discharge header with other pumps and is then stopped, the flow will reverse through the pump unless the discharge valve is closed or unless there is a check valve in the system or a broken siphon in a siphon system. If the pump does not have a nonreversing device, it will turn in the reverse direction. A pump that discharges against a static head through a siphon system without a valve will have reverse flow and speed when the siphon is being primed prior to starting.

Figures 22 and 23 illustrate typical reverse-speed-torque characteristics for a low- and a high-specific-speed pump. When flow reverses through a pump and the driver offers very