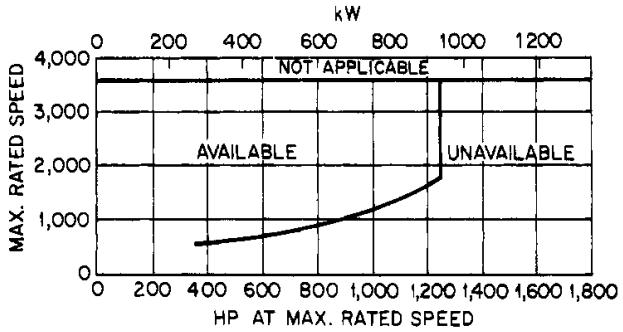


FIGURE 6C Approximate availability of explosion-proof vertical squirrel-cage induction motors (General Electric)

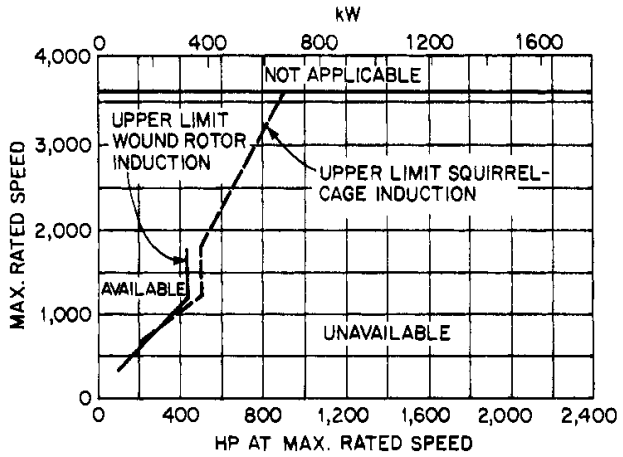


FIGURE 6D Approximate availability of explosion-proof horizontal squirrel-cage and wound-rotor induction motors (General Electric)

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

[a]Drive may be designed for constant torque or for torque varying at the square of the speed.

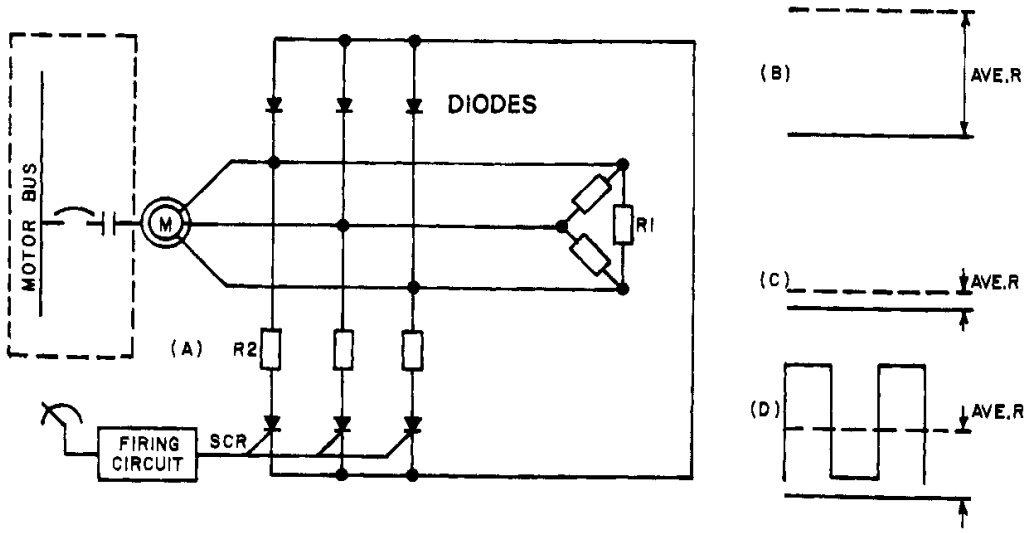


FIGURE 7 Block diagram for wound-rotor induction motor with Tirastat II secondary control power (General Electric)

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

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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

[a]Drive 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.

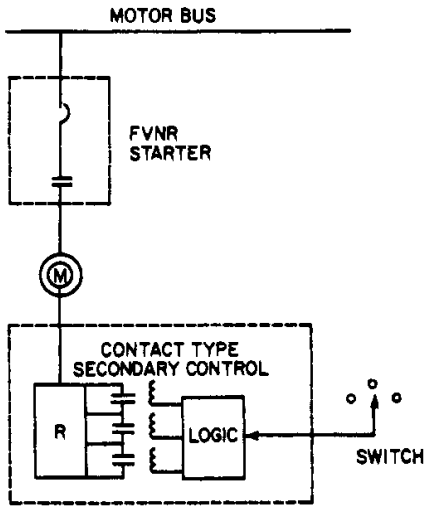


FIGURE 8 Block diagram for wound-rotor induction motor with magnetic secondary control power (General Electric)

TABLE 4 Wound-rotor inductions 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

[a]Drive 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.

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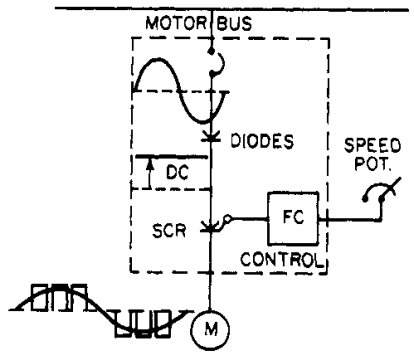


FIGURE 9 Block diagram for ac adjustable-frequency drive (General Electric)

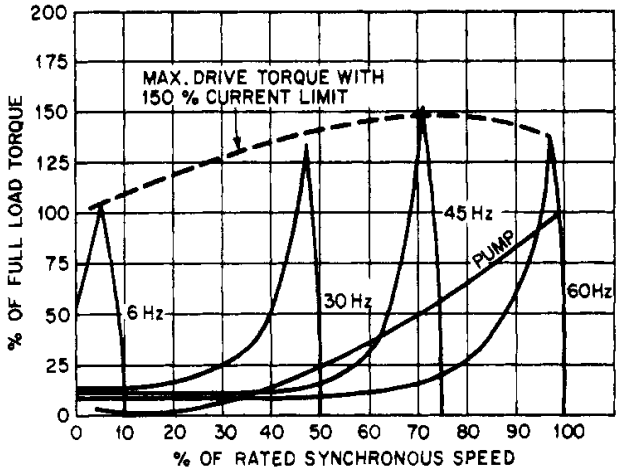


FIGURE 10 Torque versus synchronous speed for adjustable-frequency drive (General Electric)

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

[a]Drive may be designed for constant torque or for torque varying as the square of the speed.

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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.)

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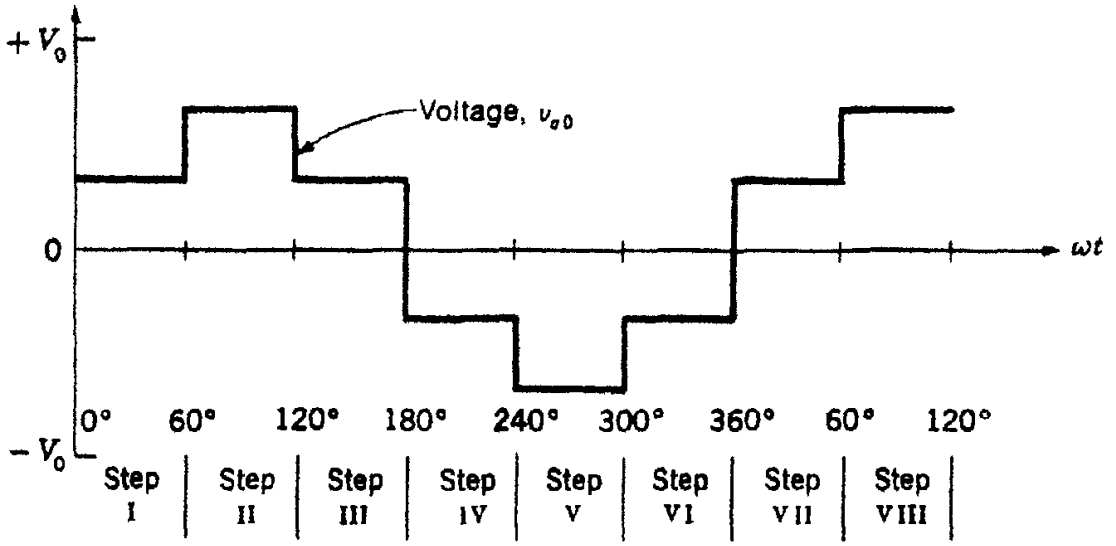


FIGURE 12 AC inverter (6) step simulated sine wave shown for one phase

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.

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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 aide of the converter unit.

Under running conditions, rotor slip energy of low voltage and frequency (Figure 13B) flows to the low-frequency aide 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 H 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.

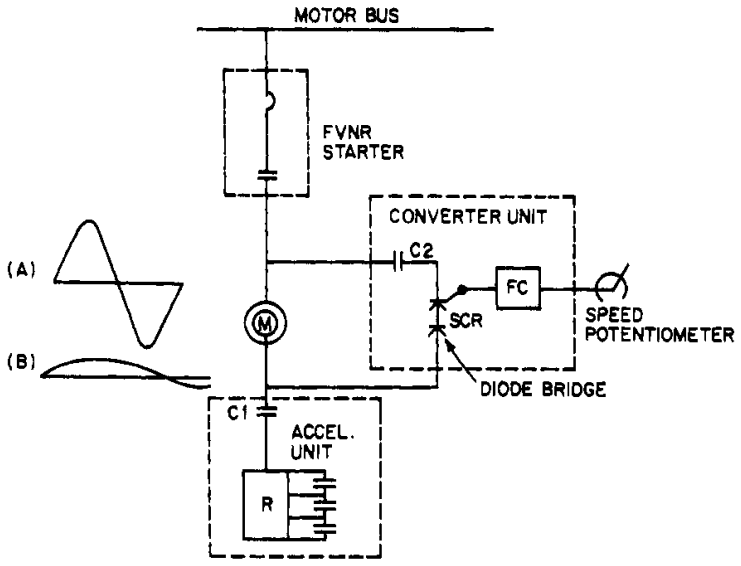


FIGURE 13 Block diagram for modified Kraemer drive

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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

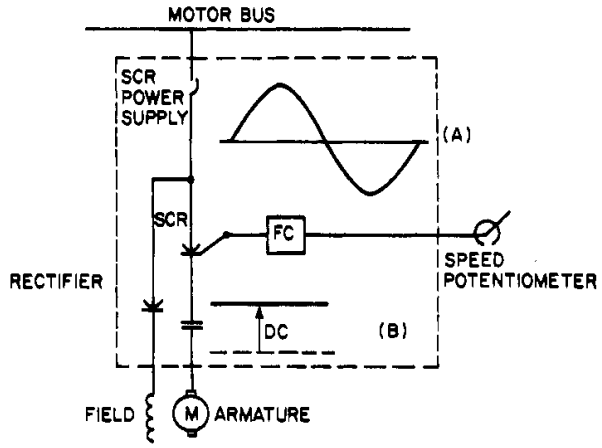


FIGURE 14 Block diagram for dc motor with SCR power supply

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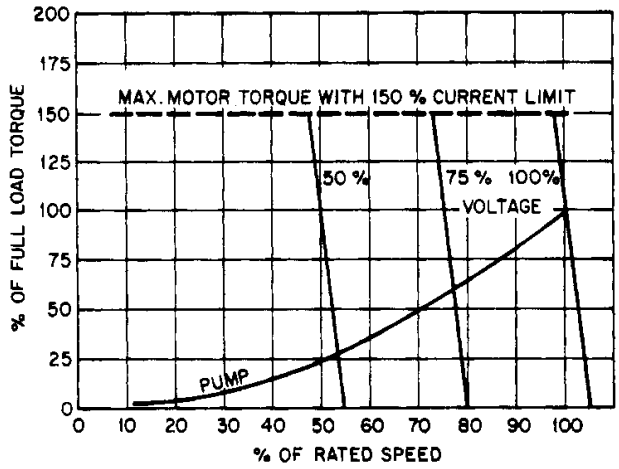


FIGURE 15 Torque versus speed for dc motor with SCR power supply

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[a]	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, upthrough 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

[a]Drive can power either constant-torque loads or those varying as the square of the speed.

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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

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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.

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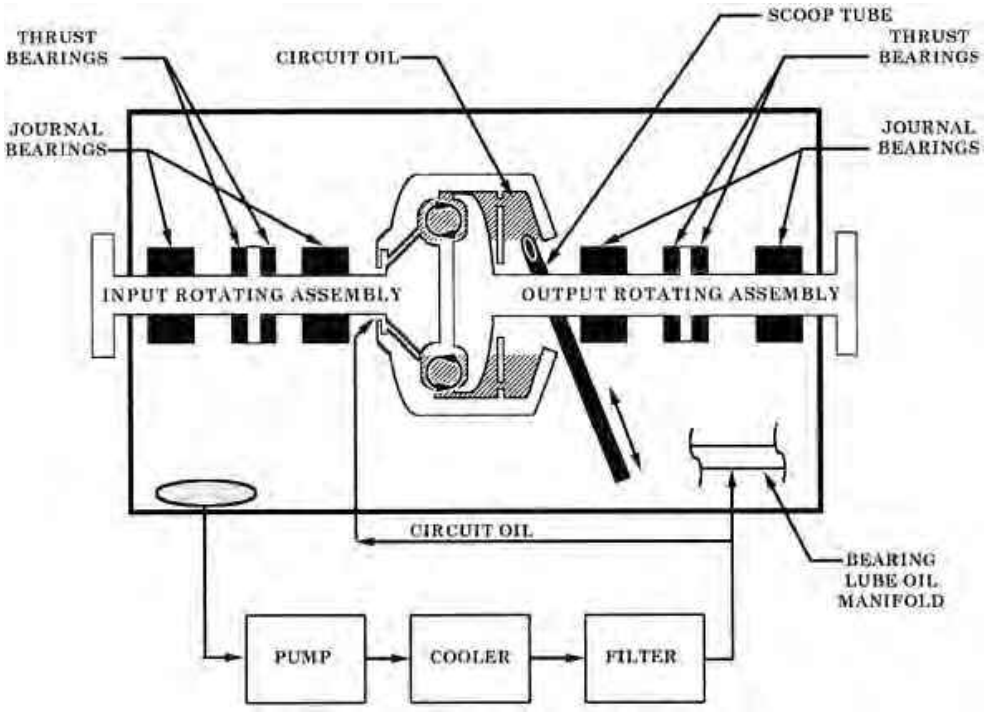


FIGURE 1 "Hydro-kinetic" variable speed fluid drive

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:

$$\text{Power (hp)} = \frac{K \times (N_{out})^3 \times D^5}{10^{14}}$$

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

$N_{out} = 0.975 \times N_{in}$ (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

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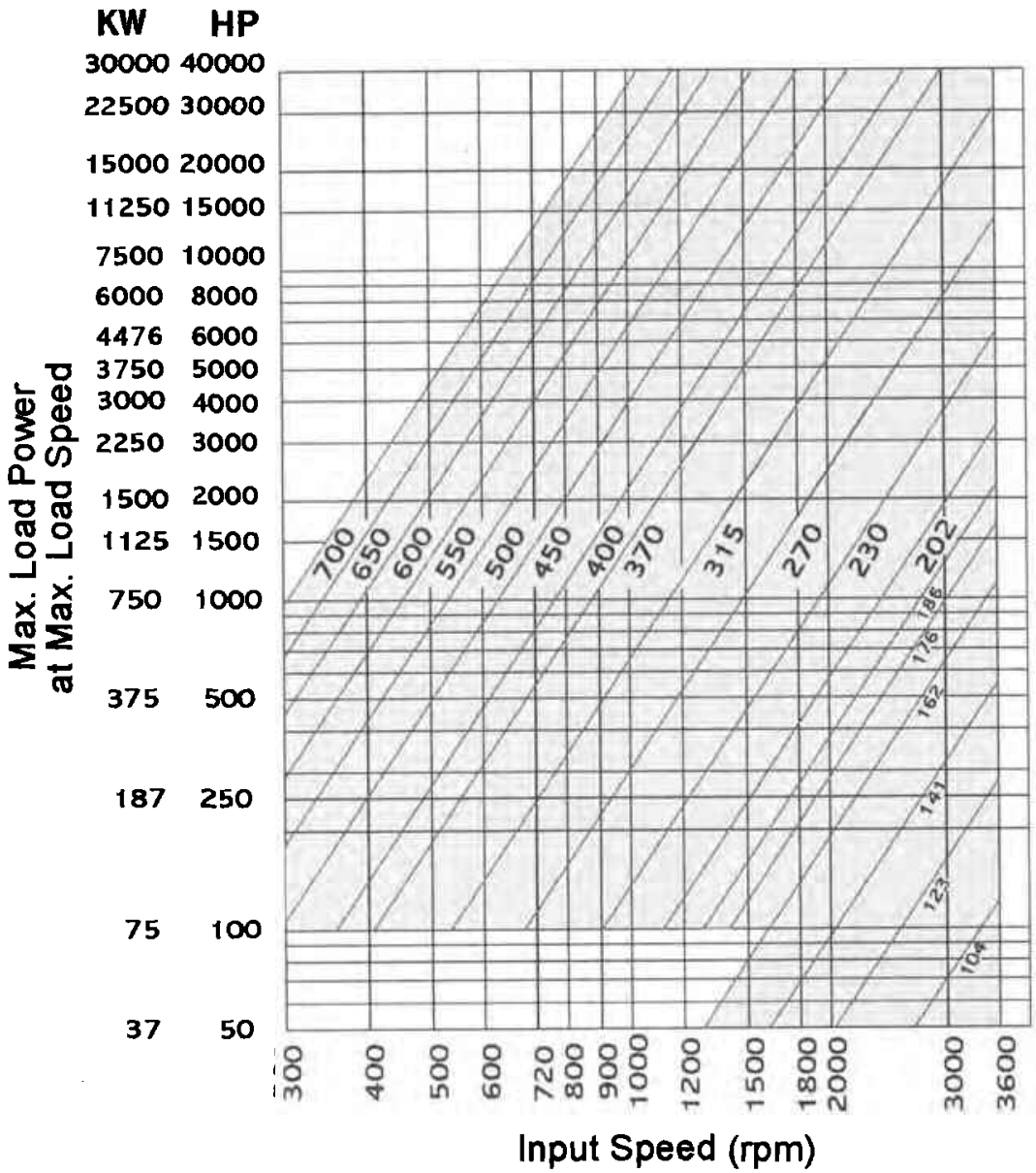


FIGURE 2 Sizing chart for fluid drives. Hydraulic diameters are shown on diagonal (For example, "370" signifies 37.0 in)

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.

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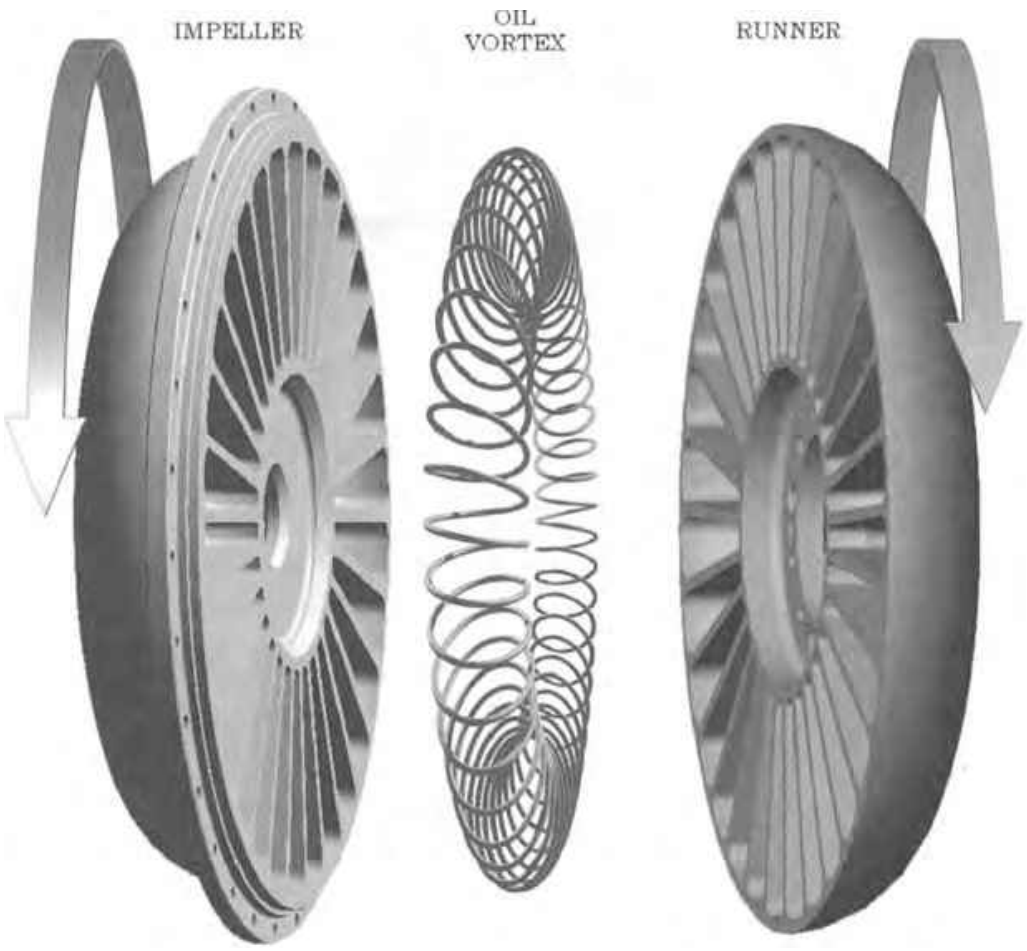


FIGURE 3 The trajectory of each packet of oil is a spiral through the impeller and runner as they rotate about the shaft centerline

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.

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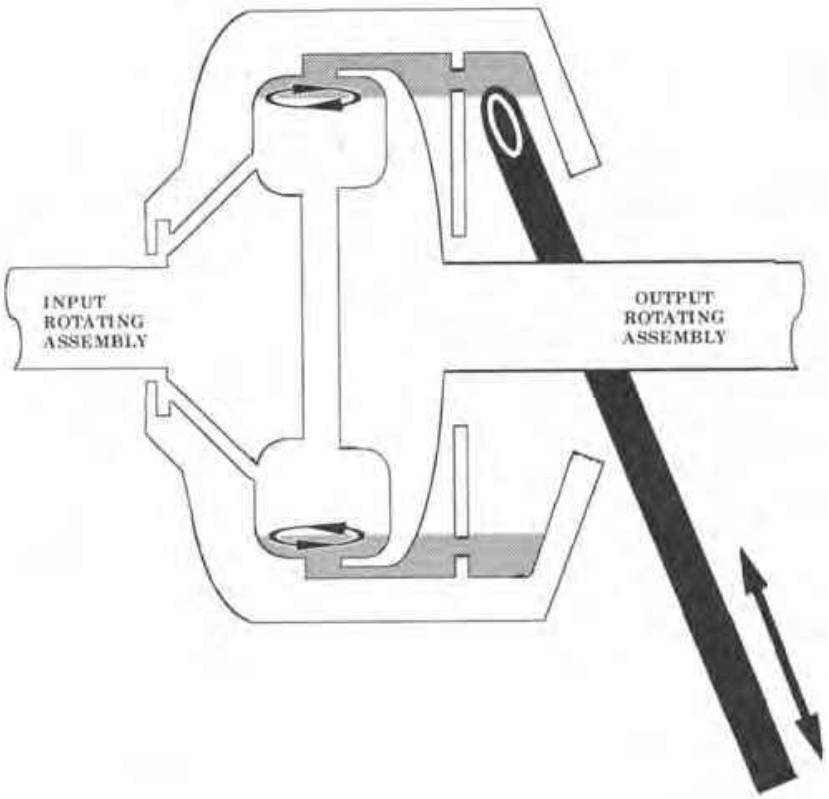


FIGURE 4A A small torus of circuit oil provides small coupling

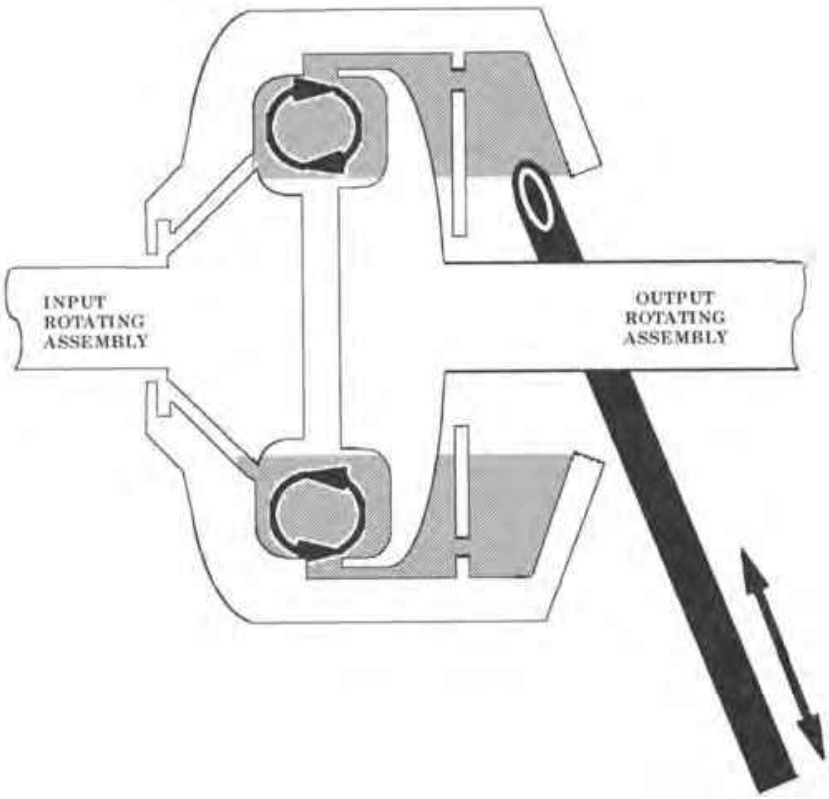


FIGURE 4B A large torus of circuit oil provides large coupling

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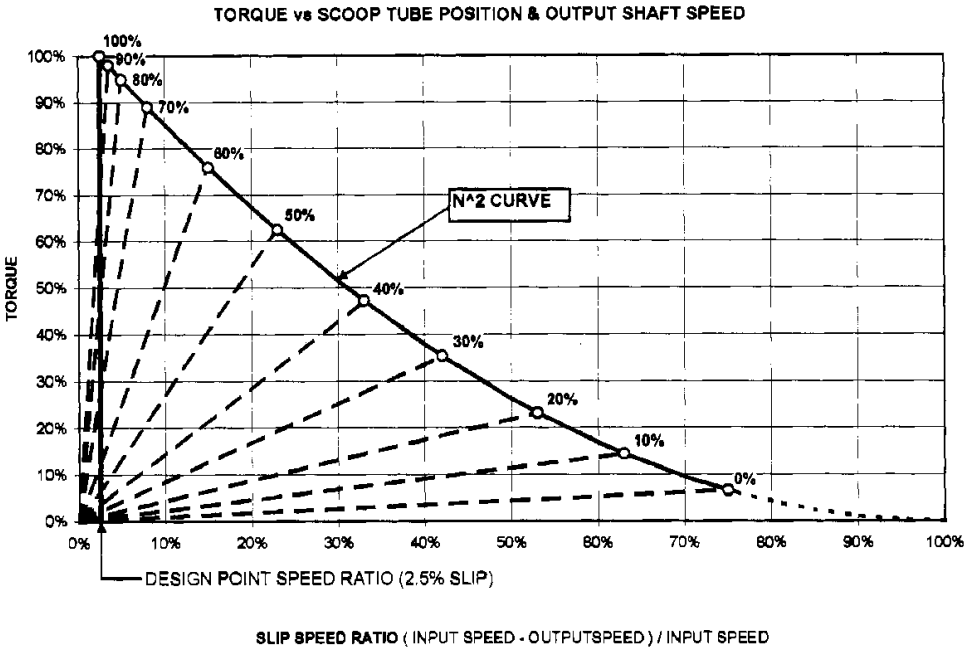


FIGURE 5 Chart of output torque ratio versus output speed ratio for different scoop tube positions

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:

Torque in = torque out = torque (lbf-in)

Power in = torque in (lbf-in) × input rotational frequency (rad/sec)

Power out = torque out (lbf-in) × output rotational frequency (rad/sec)

Power loss = power in – power out = torque (lbf-in) × (input rotational frequency – output rotational frequency) (rad/sec)

Heat power (btu/h) = power loss (hp × 2544 btu/hp · 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.

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The efficiency of a variable speed fluid drive of the hydro-kinetic type can typically be expressed as:

Efficiency = (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% (slip loss) − 3.5% (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,

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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

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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 Fottinger 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 Konstruction*, by Ing. Maurizio Wolf, Springer-Verlag, Berlin/Gottingen/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):

$$\text{(Power out)} = (N_{out})^3$$

And $\text{(Torque out)} = (N_{out})^2$

Now $\text{(Torque in)} = \text{(torque out)} = \text{torque}$

Hence $\text{(Power in)} = \text{(torque in)} \times (N_{in})$

$$\text{(Power in)} = (N_{out})^2 \times (N_{in})$$

$$\text{(Power loss)} = \text{(power in)} - \text{(power out)}$$

$$= (N_{out})^2 \times N_{in} \times (N_{out})^3$$

The maximum loss occurs at $(N_{out}) = 2/3\ (N_{in})$

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

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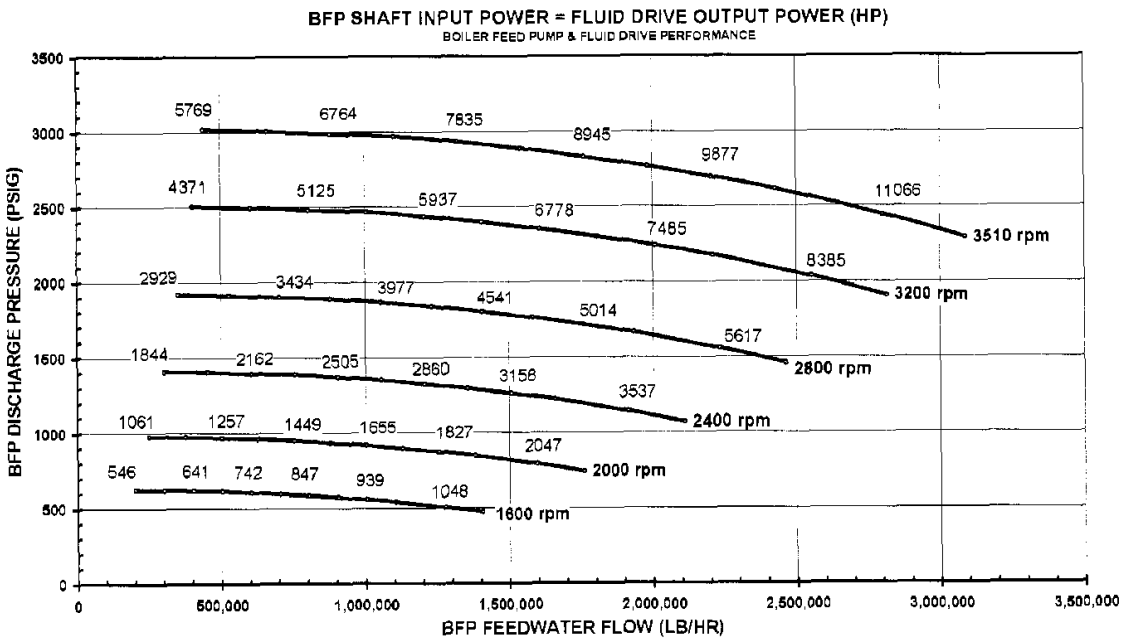


FIGURE 6 (Chart 1) Power absorbed by the pump shaft from the fluid drive output shaft

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.

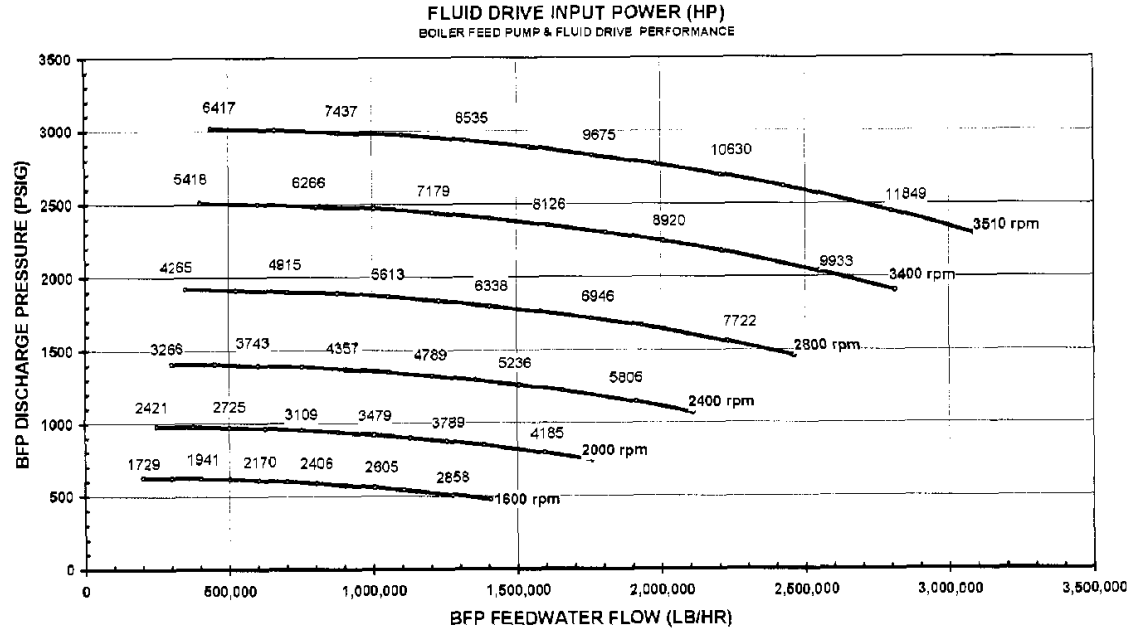


FIGURE 7 (Chart 2) Power absorbed by the fluid drive input shaft from the motor or turbine-generator output shaft at the same operating points of the grid

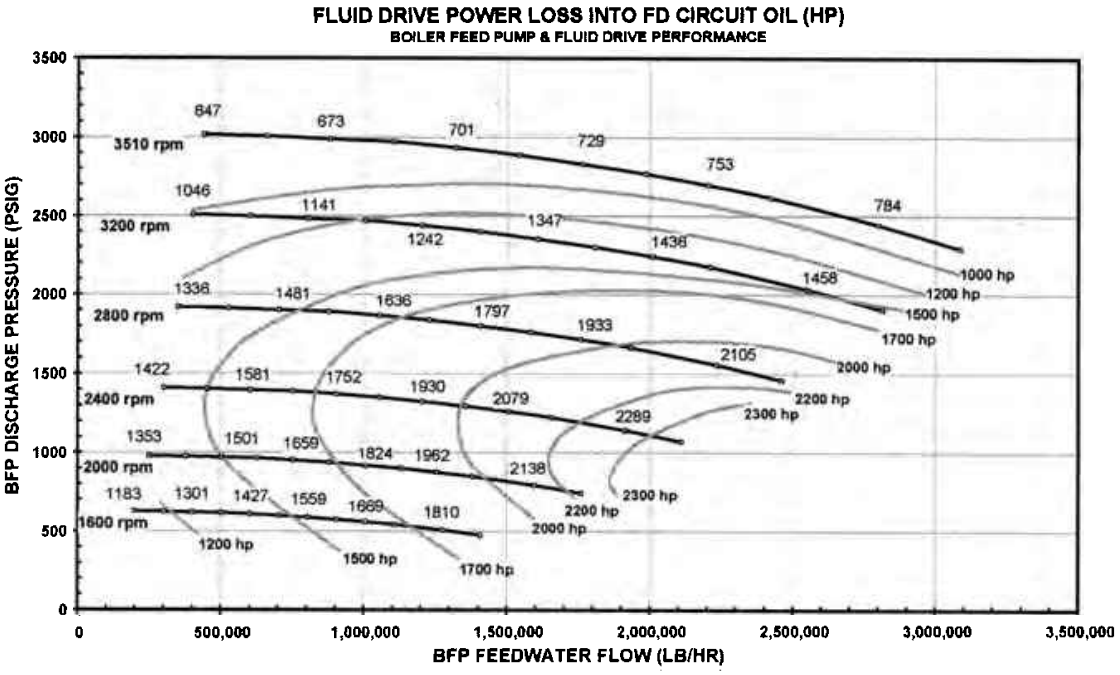


FIGURE 8 (Chart 3) Power lost to the circuit oil in the fluid drive element (equals power into the fluid drive minus power out of the fluid drive) at the same operating points of the grid

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).

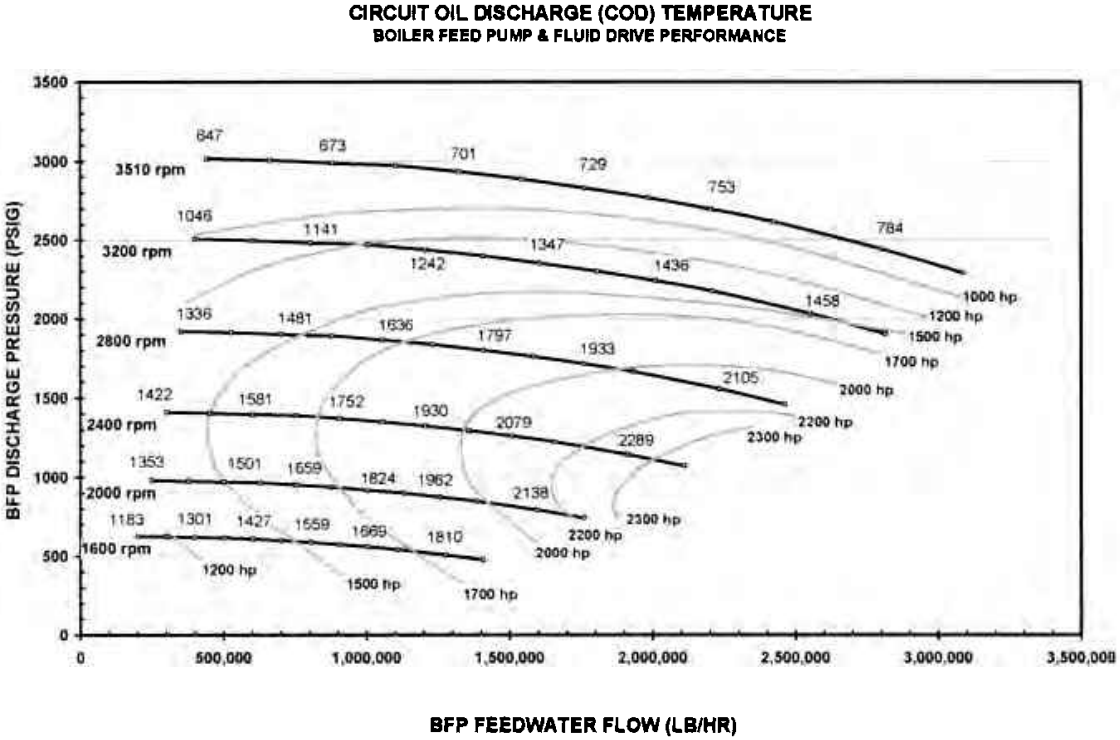


FIGURE 9 (Chart 4) The circuit oil discharge temperature that results from the power lost per Chart 3 with the circuit oil input temperature of 120°F (49°C) and a constant circuit oil flow rate of 400 gpm (91 m³/h)

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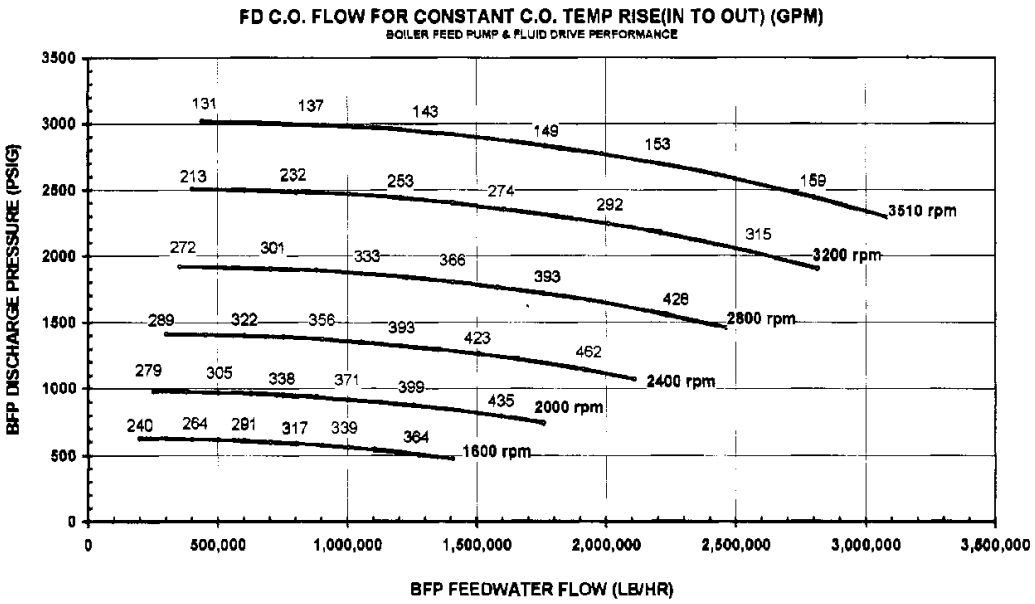


FIGURE 10 (Chart 5) Circuit oil flow rate (gpm) required to maintain a constant temperature rise of the circuit oil (the difference between the circuit oil temperature entering the fluid drive and the circuit oil discharge (COD) temperature from the fluid drive rotating element)

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

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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,

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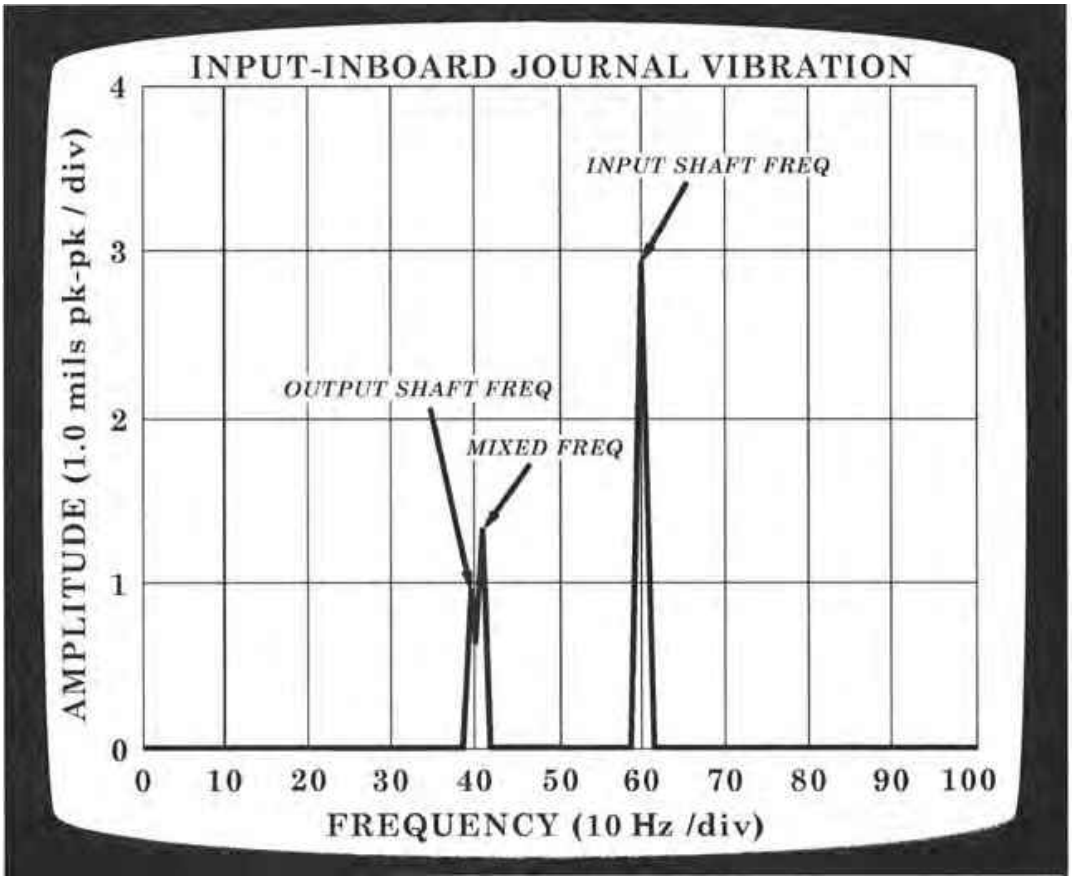


FIGURE 11 A typical vibration frequency analysis of the journal at the input-inboard bearing of a high power variable frequency fluid drive

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 worlds 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

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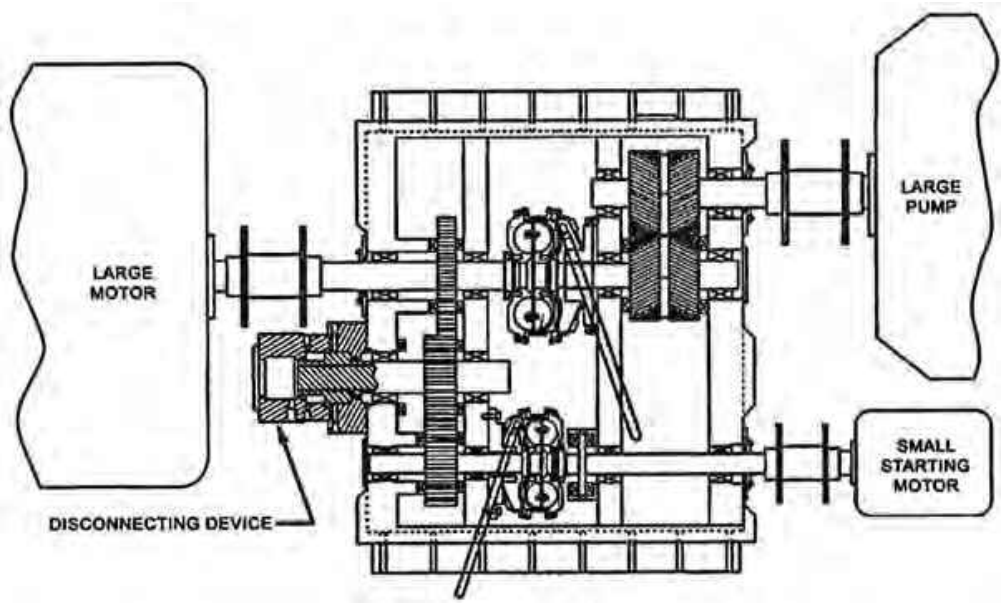


FIGURE 12 Variable speed geared fluid drive with parallel start-up capability that is used to start a very large motor which can then drive a very large pump (Turbo Research, Inc., U.S. Patent 7,171,870)

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.

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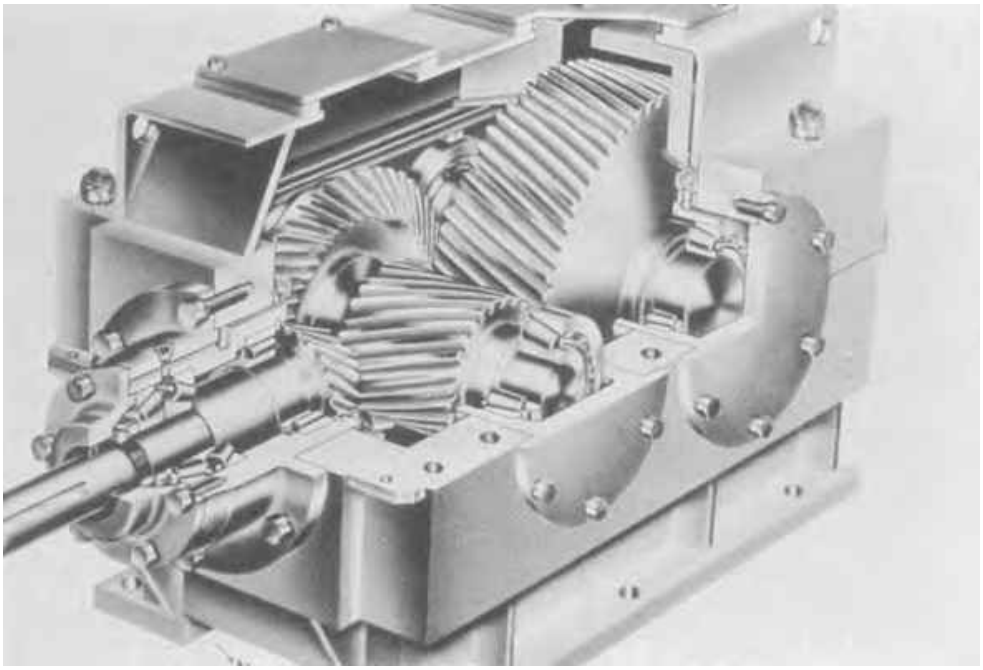


FIGURE 1 Typical pump drive gearbox (Lufkin Industries, Inc.)

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

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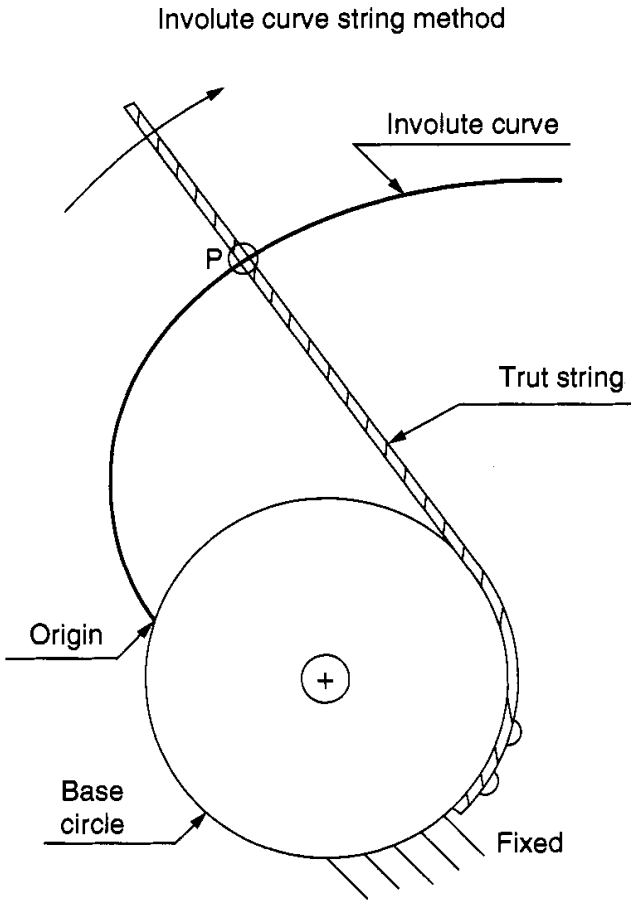


FIGURE 2 Generation of an involute (Lufkin Industries, Inc.)

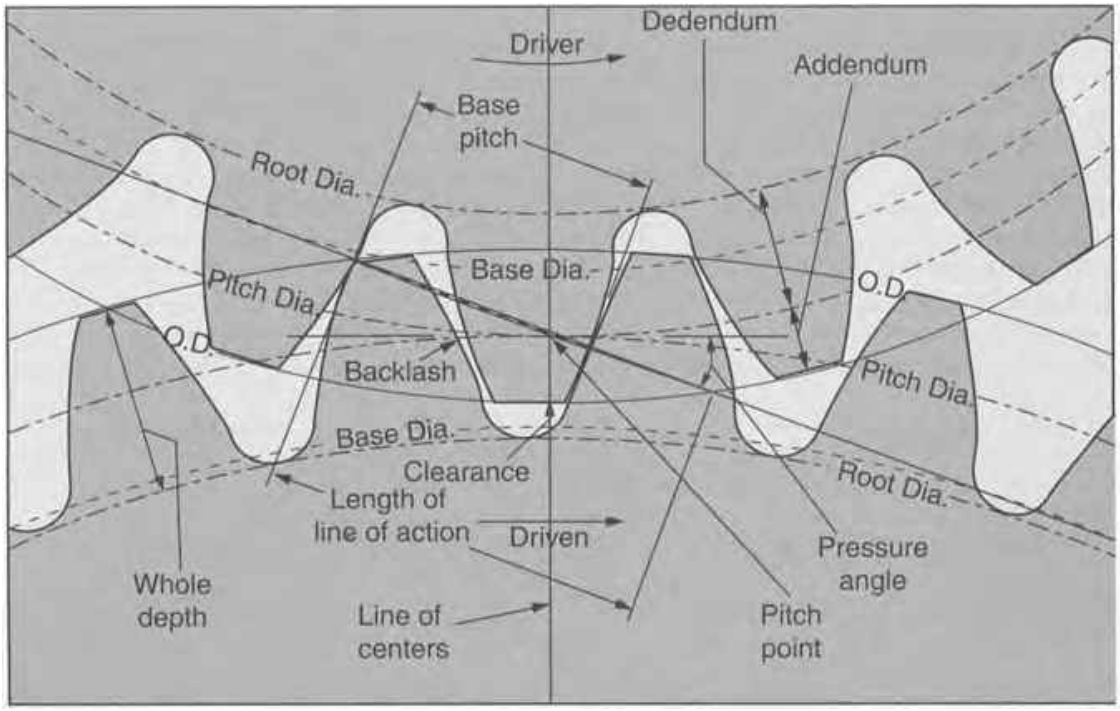


FIGURE 3 Gear tooth nomenclature (AGMA)

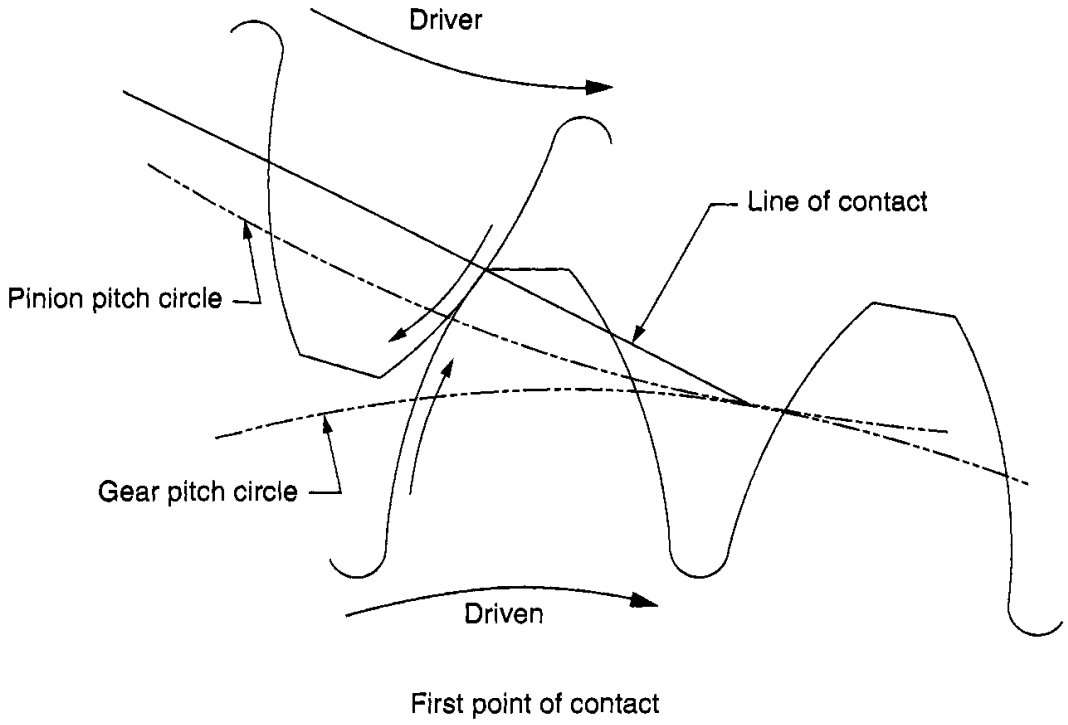


FIGURE 4 First point of contact (Lufkin Industries, Inc.)

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.

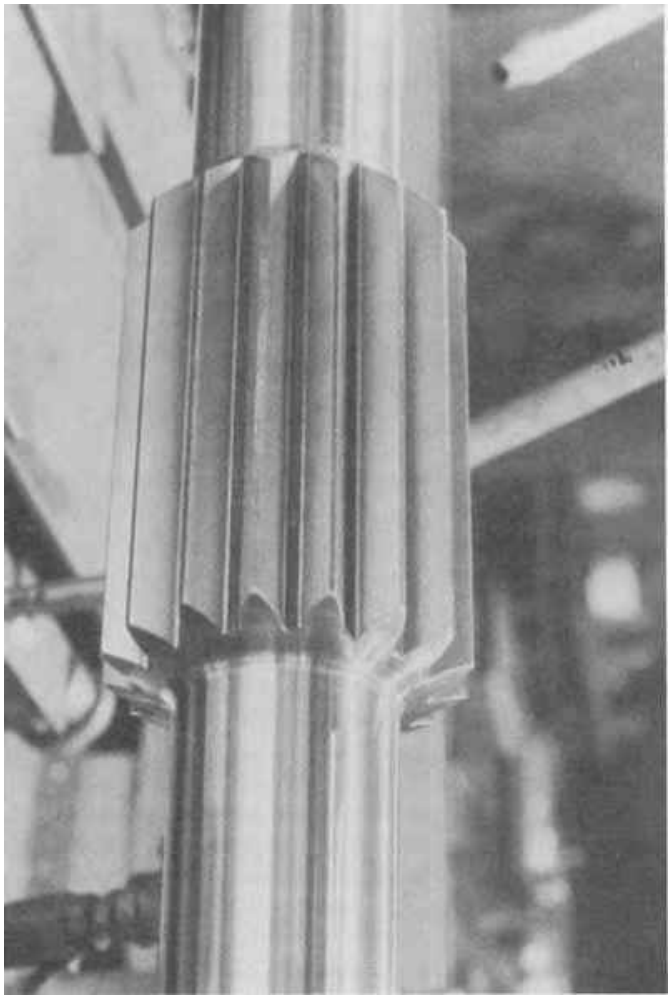


FIGURE 5 Spur gear (Lufkin Industries, Inc.)

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

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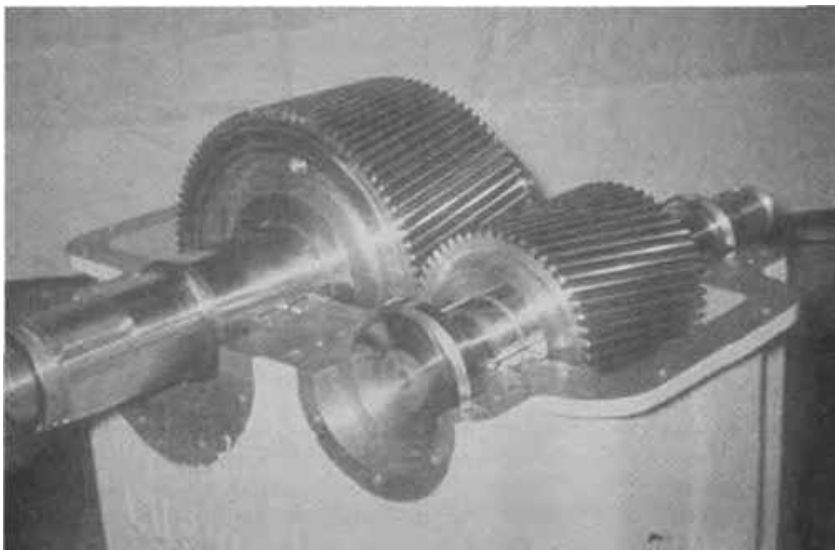


FIGURE 6 Single helical gearbox (Lufkin Industries, Inc.)

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

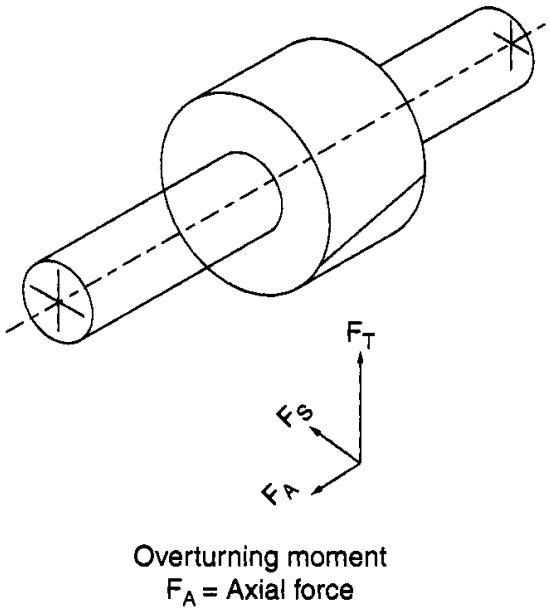


FIGURE 7 Single helical overturning moment (Lufkin Industries, Inc.)

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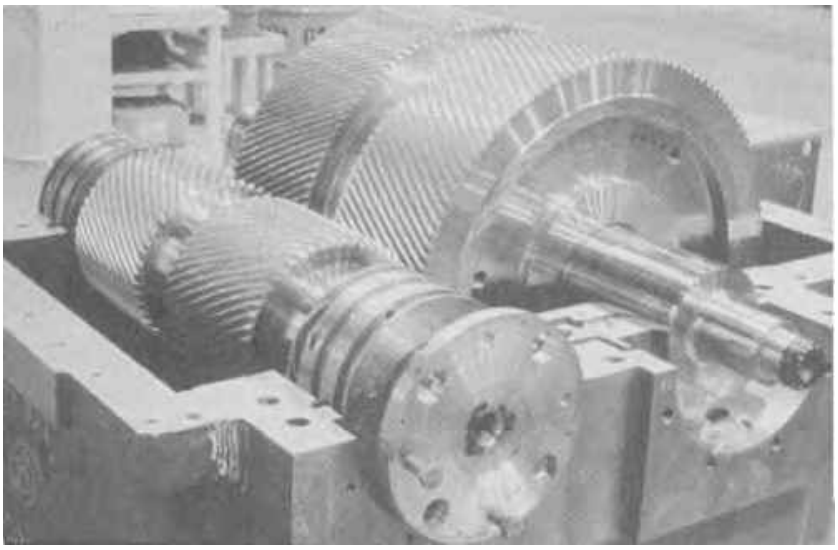


FIGURE 8 Double helical gearbox (Lufkin Industries, Inc.)

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.

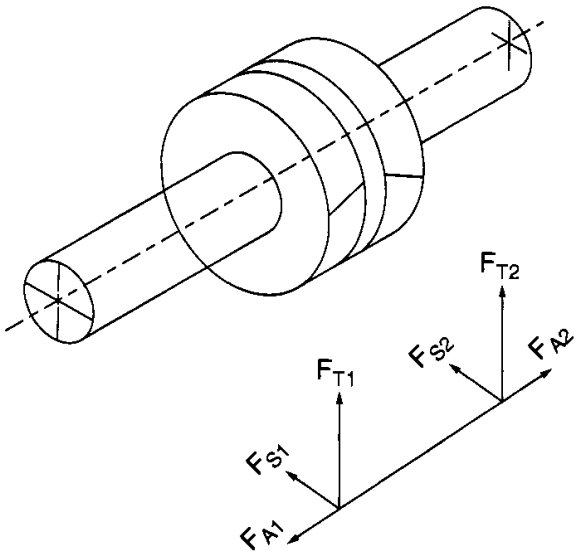


FIGURE 9 Double helical overturning moment (Lufkin Industries, Inc.)

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 gearbox). 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 helixes 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)

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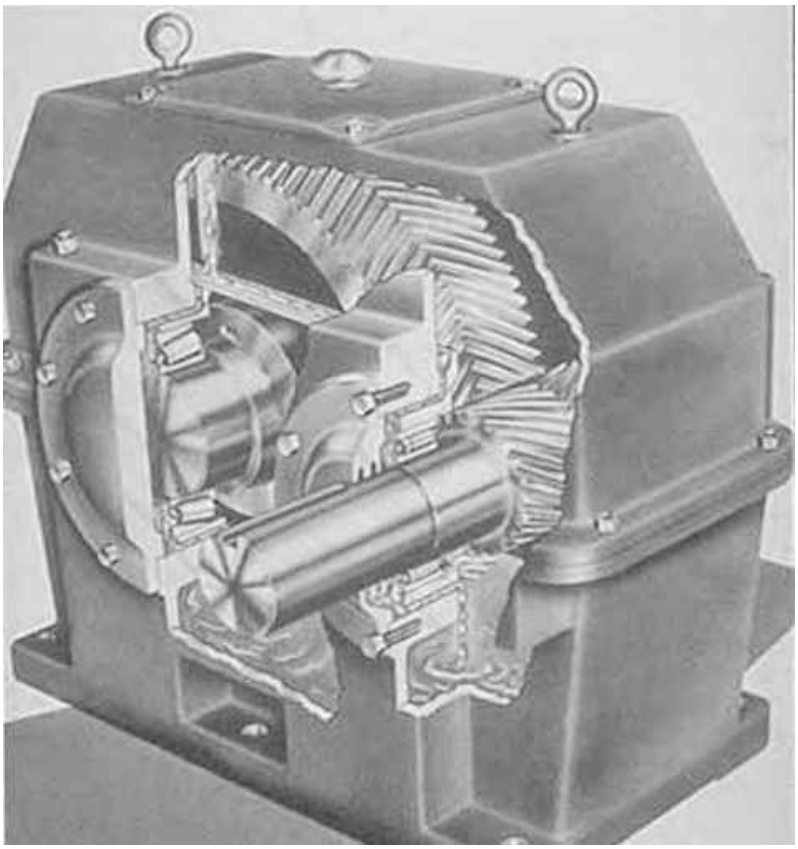


FIGURE 10 Herringbone gearbox (Lufkin Industries, Inc.)

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:

$$\begin{aligned} PD &= (2 \times 12)/[(4000/1200) + 1] \\ &= 5.538 \text{ inch} \end{aligned}$$

$$\begin{aligned} PLV &= 5.538 \times 3.14 \times 4000/12 \\ &= 5796 \text{ ft/min} \end{aligned}$$

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.

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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.

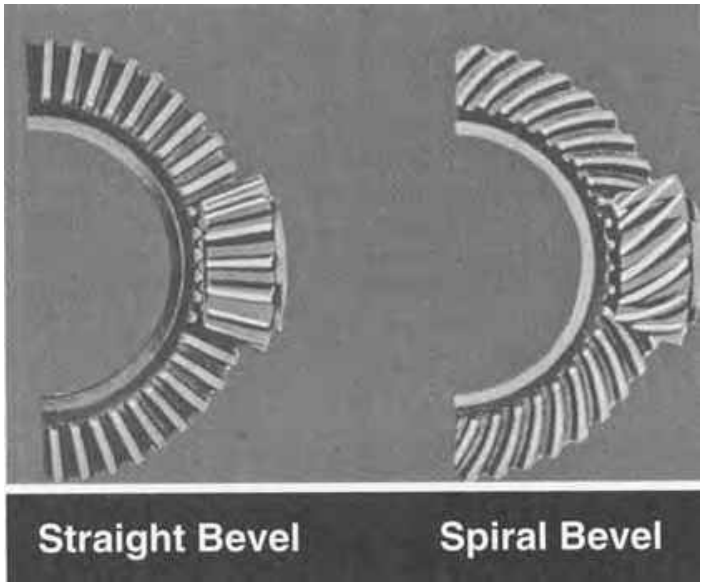


FIGURE 11 Straight and spiral bevel gear (AGMA)

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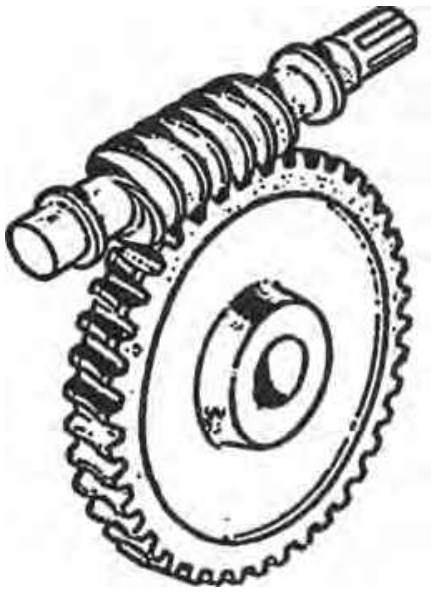


FIGURE 12 Worm gear and pinion (AGMA)

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

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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

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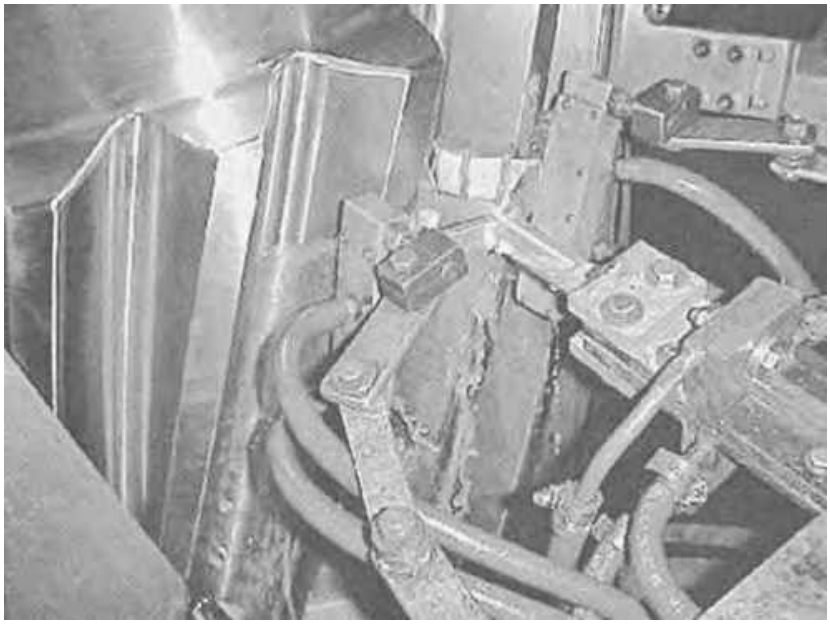


FIGURE 13 Induction hardening (Lufkin Industries, Inc.)

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.)

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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

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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.

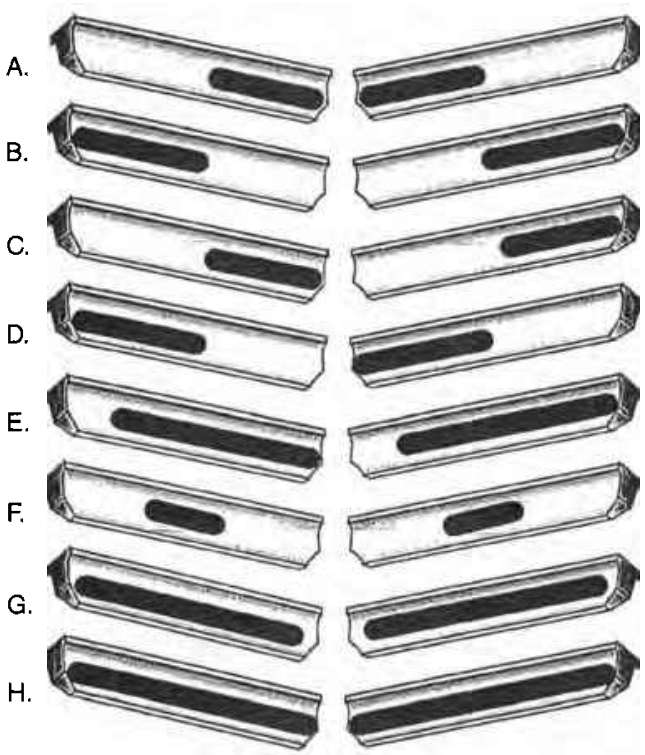


FIGURE 15 Various double helical gear set contacts (Lufkin Industries, Inc.)

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 Standard[1] 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:

$$\text{Viscosity} = 7000/(\sqrt{\text{PLV}}) \tag{4}$$

$$\text{Where viscosity} = \text{viscosity of oil in centistokes}$$

For the example used in the calculation of PLV, a 4000-rpm pump driven by a 1200 rpm motor:

$$\text{Viscosity} = 7000/(\sqrt{5796})$$
$$= 92 \text{ centistokes}$$

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.

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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 4406[3] 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

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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.

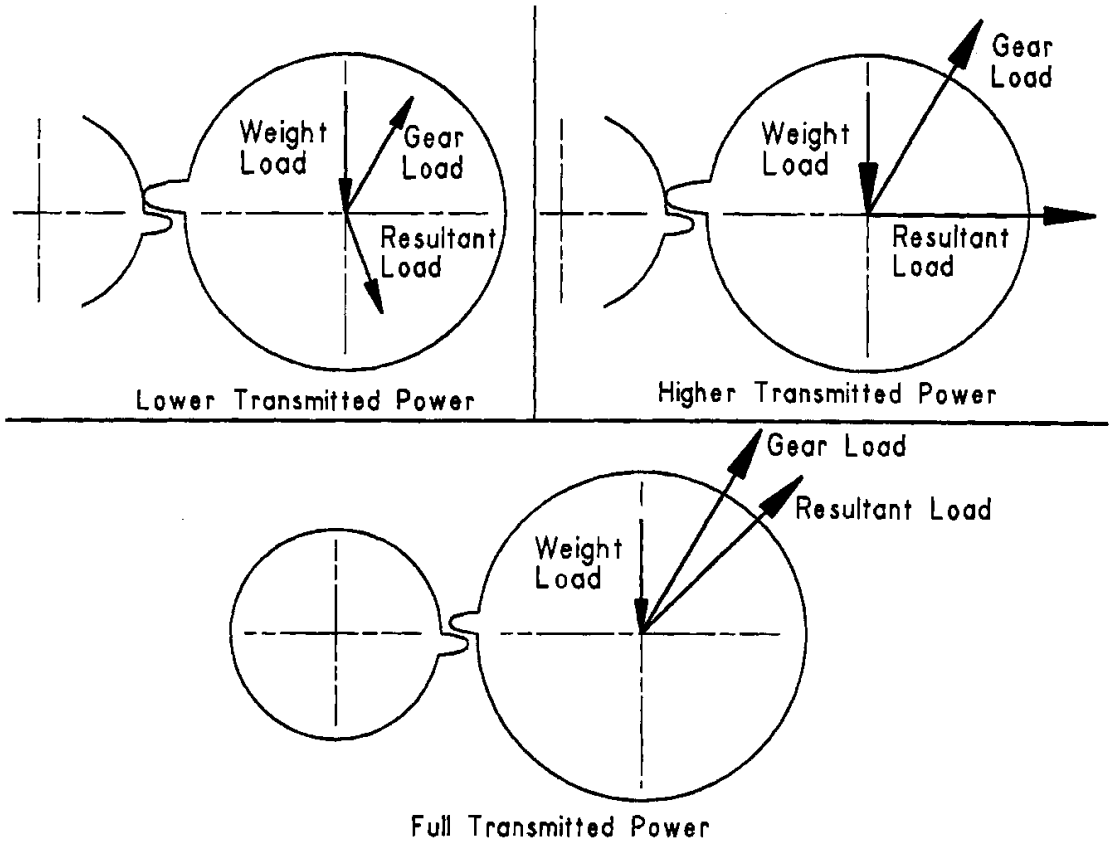


FIGURE 16 Uploaded versus downloaded gear (Lufkin Industries, Inc.)

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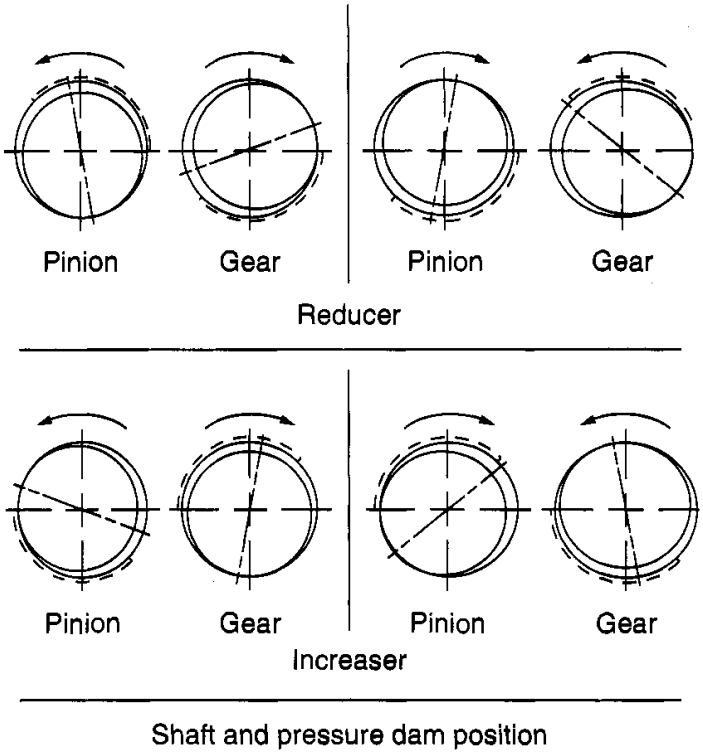


FIGURE 17 Pressure dam bearing orientation (Lufkin Industries, Inc.)

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.

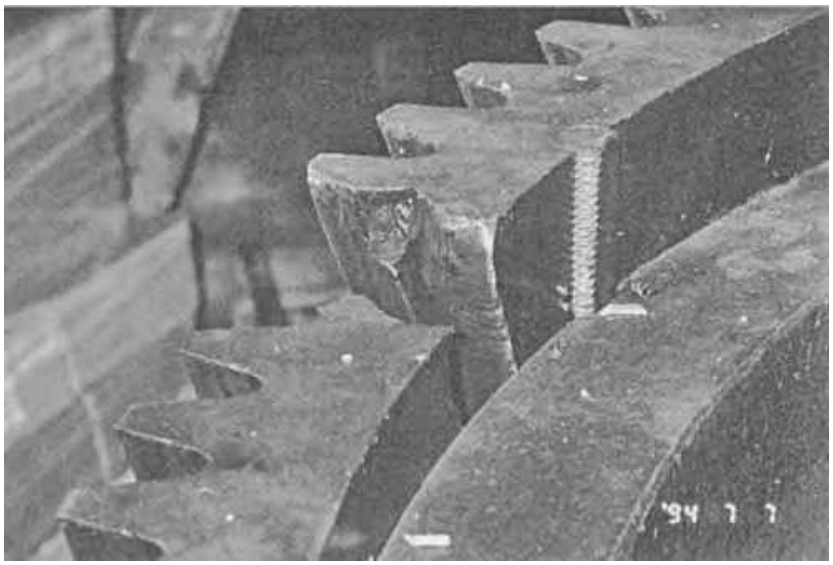


FIGURE 18 Removing a gear band (Lufkin Industries, Inc.)

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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.

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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.

REFERENCES

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.
6. International Organization for Standardization, ISO 19825, *Gears-Wear and Damage to Gear Teeth*.

Previous Page

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

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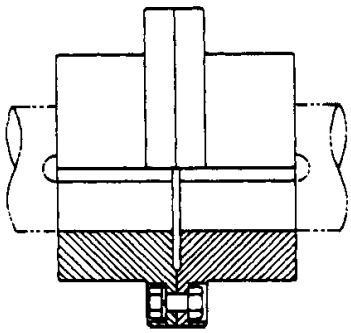


FIGURE 1 Flanged rigid coupling (Kop-Flex EPT)



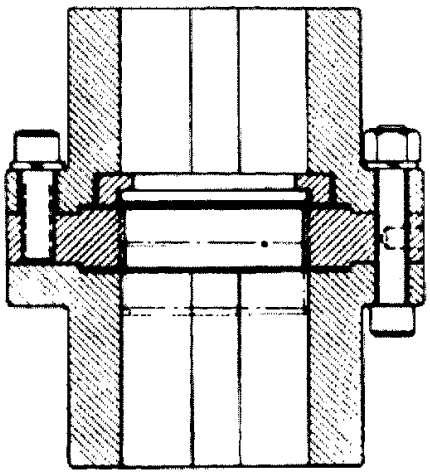
FIGURE 2 Split rigid coupling (Dodge Manufacturing Div. Reliance Electric)

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 lit 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.

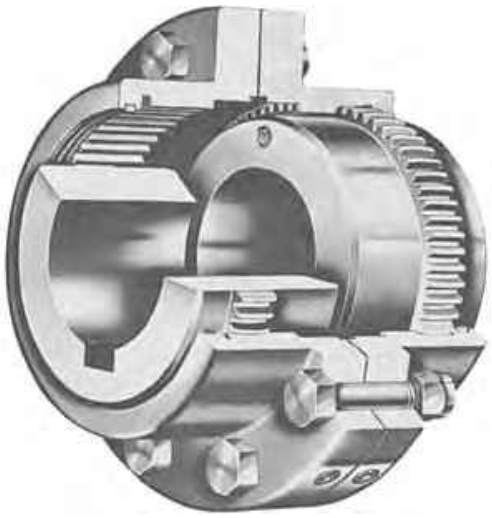


FIGURE 4 Gear-type mechanically flexible coupling (Lovejoy Inc)

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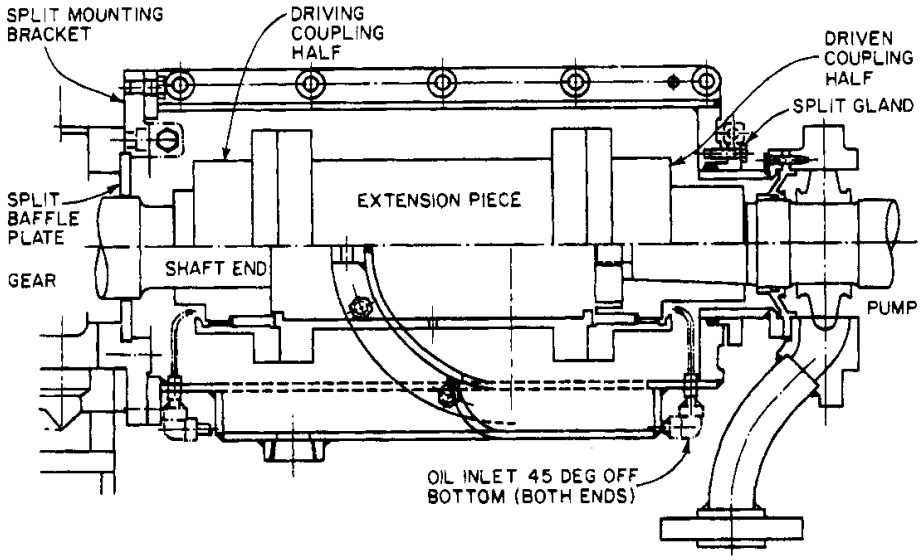


FIGURE 5 Continuously lubricated coupling (Kop-Flex EPT)

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)

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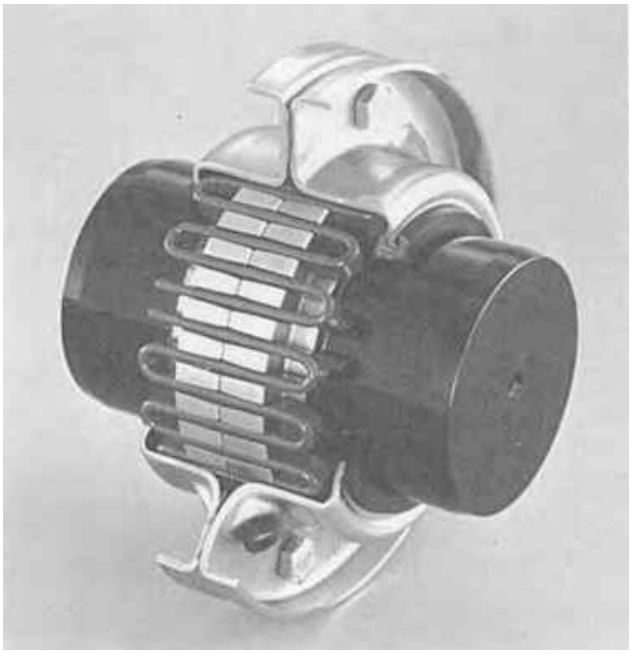


FIGURE 7 Spring-grid coupling (Lovejoy Inc)

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.

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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)

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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

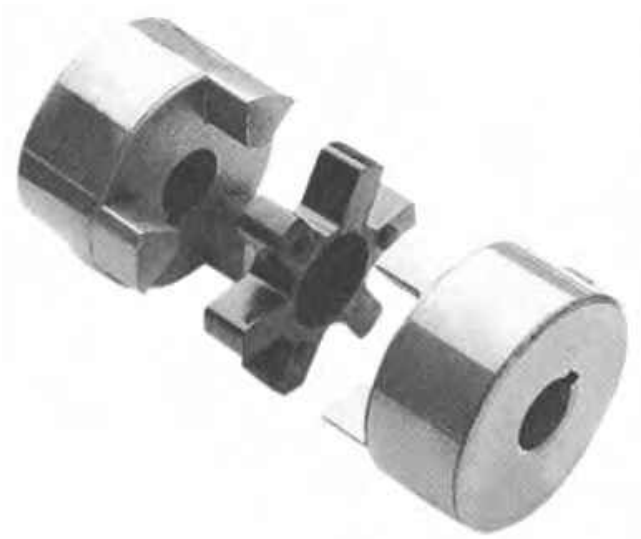


FIGURE 11A Jaw Coupling straight (Lovejoy Inc)

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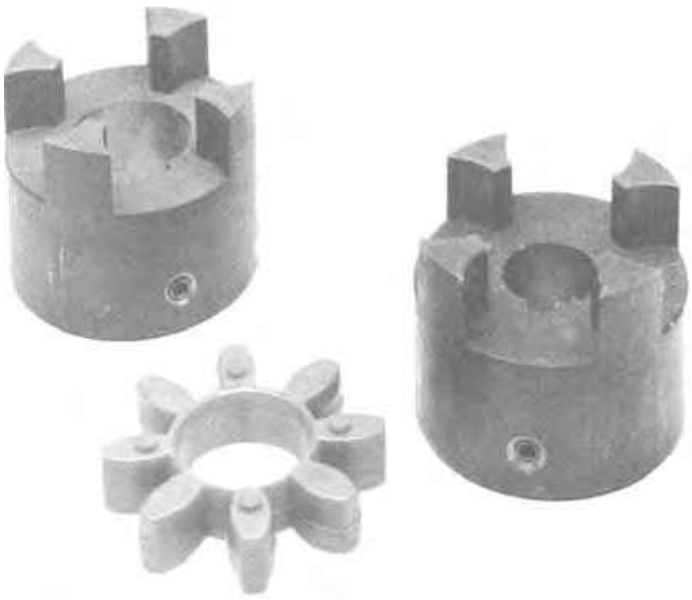


FIGURE 11B Jaw Coupling curved (Lovejoy Inc)

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

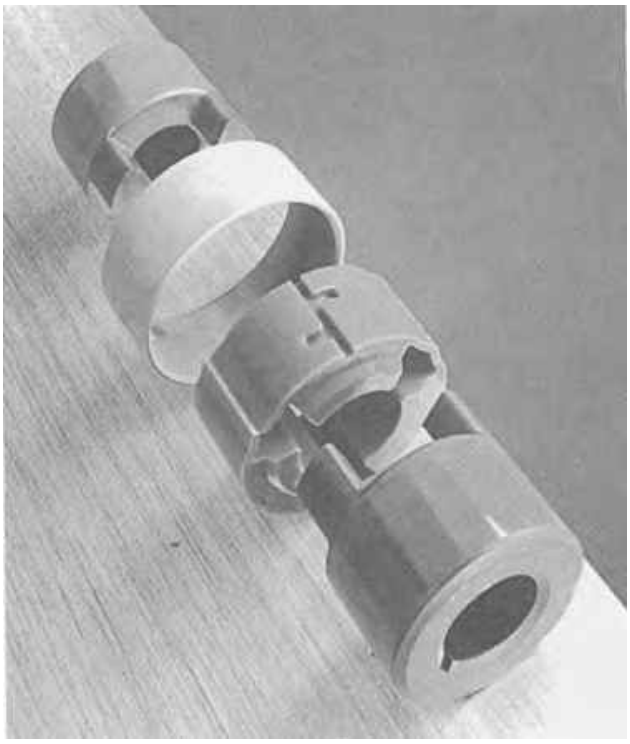


FIGURE 11C Jaw Coupling elastomer in shear (Lovejoy Inc)

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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

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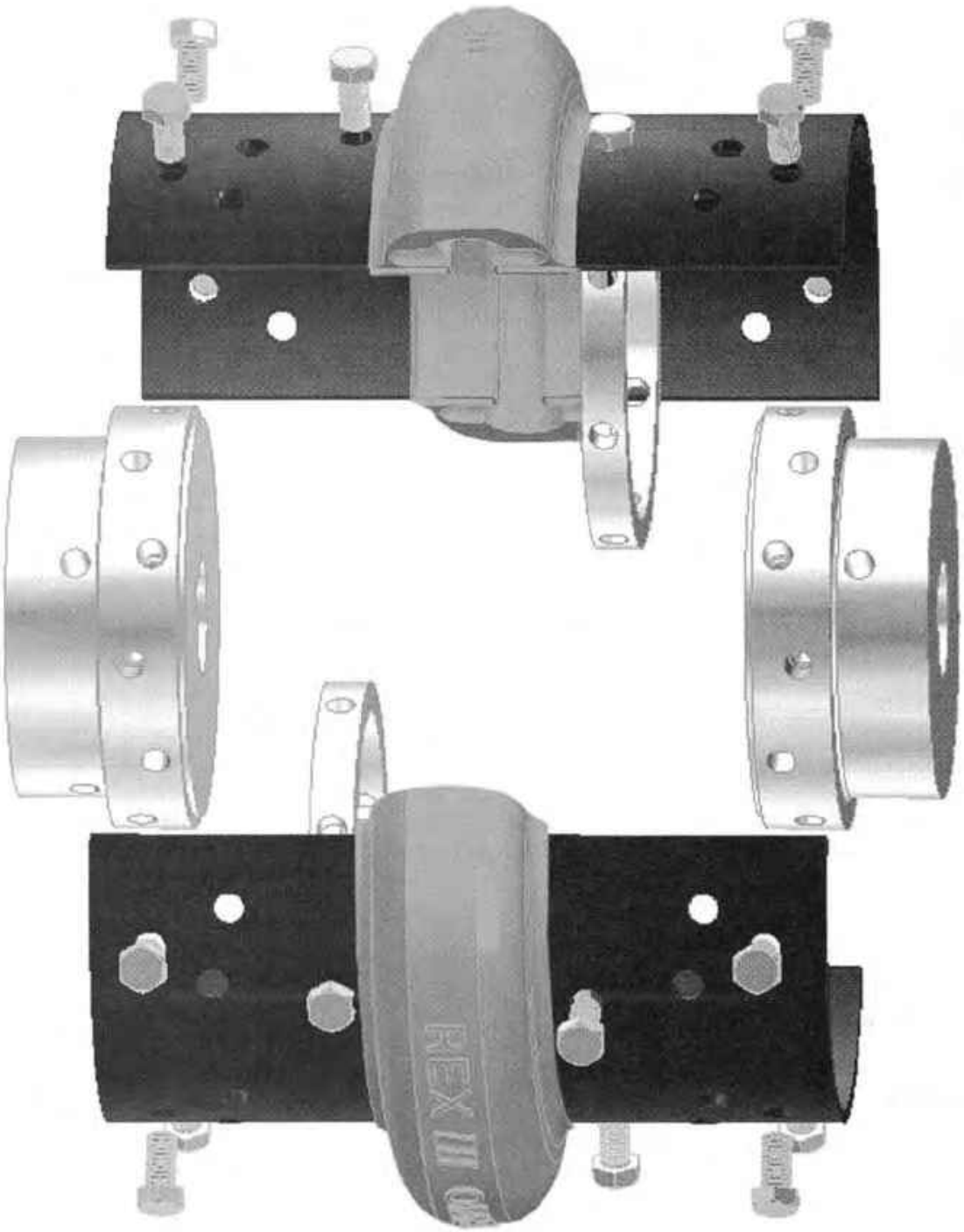


FIGURE 13 Spacer elastomer coupling (Falk Rexnord Couplings)

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

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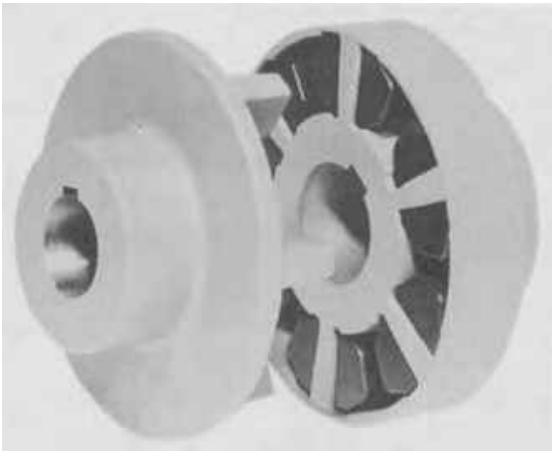


FIGURE 14 Compression loaded loose fitted elastomer coupling (Lovejoy Inc)

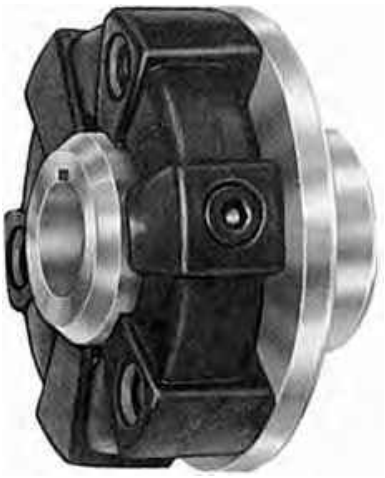


FIGURE 15 Compression loaded donut style elastomer coupling (Lovejoy Inc)



FIGURE 16 Pin-and-bushing elastomer coupling (Ajax Flexible Coupling)

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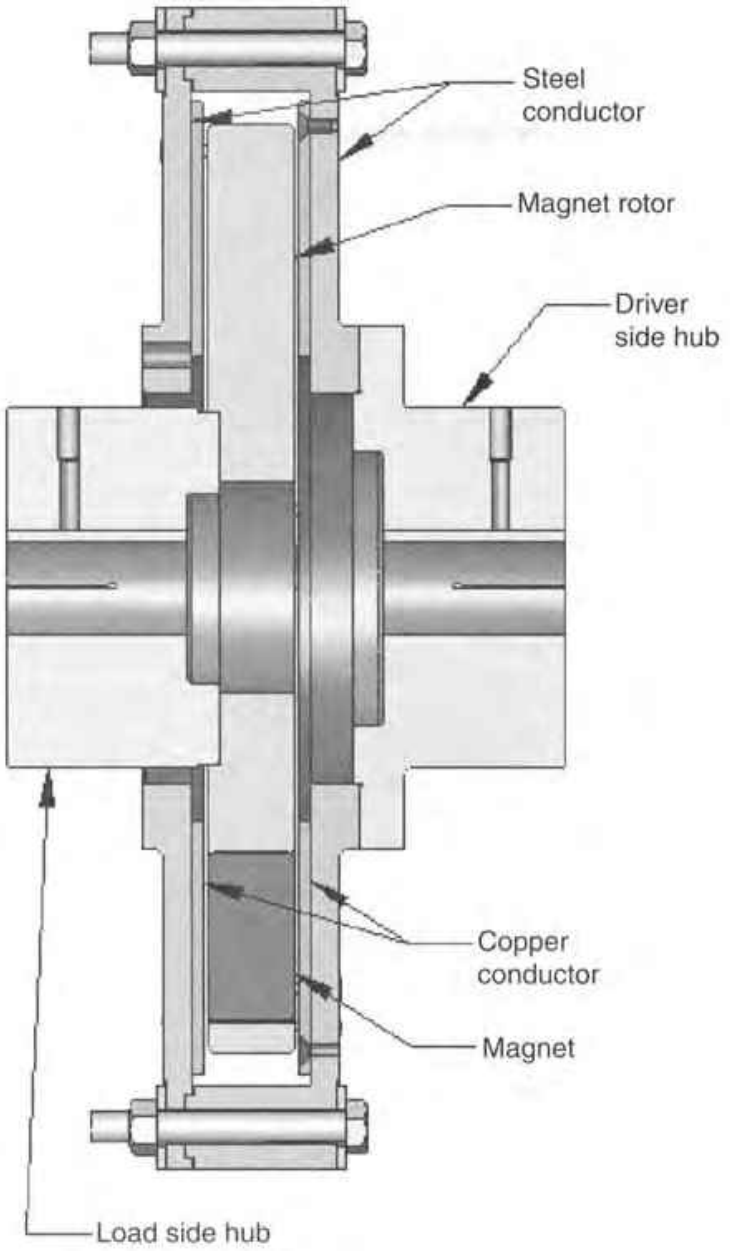


FIGURE 17 Cutaway of a shaft-mount, air-cooled permanent magnet coupling (MagnaDrive Corp)

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.

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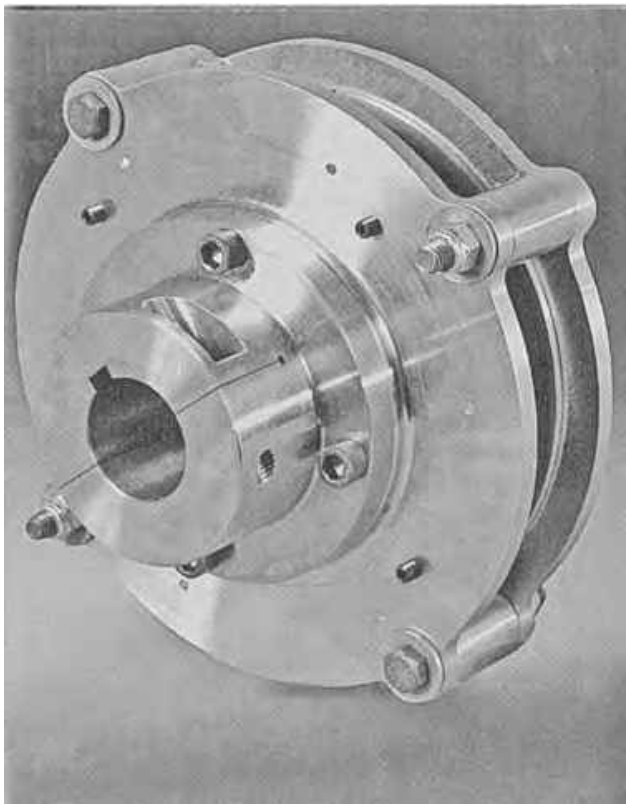


FIGURE 18 Key driven permanent magnet coupling (MagnaDrive Corp.)

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.

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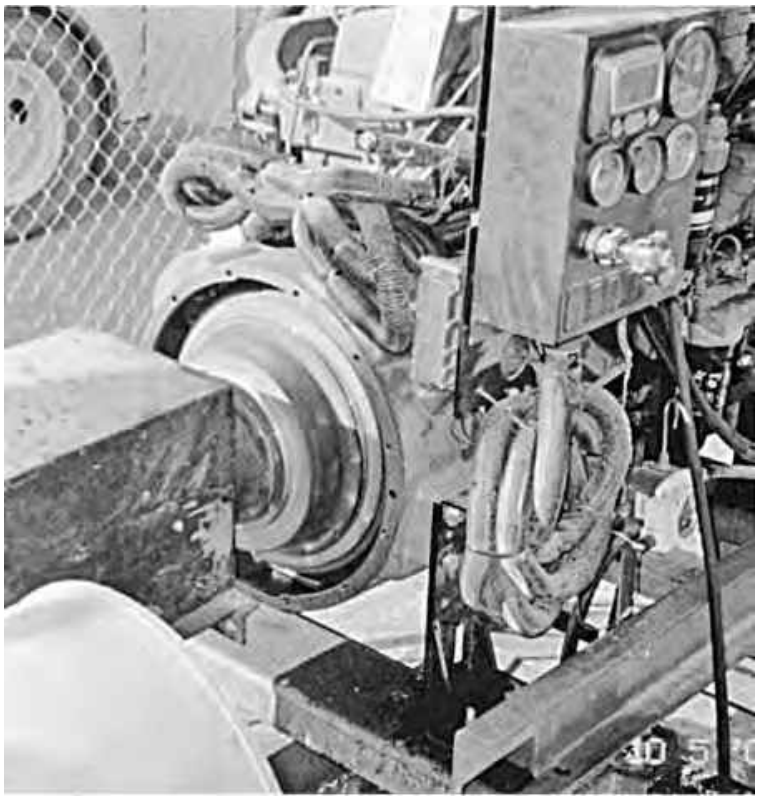


FIGURE 19 Flywheel coupling for vertical pump (Lovejoy Inc)

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

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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.

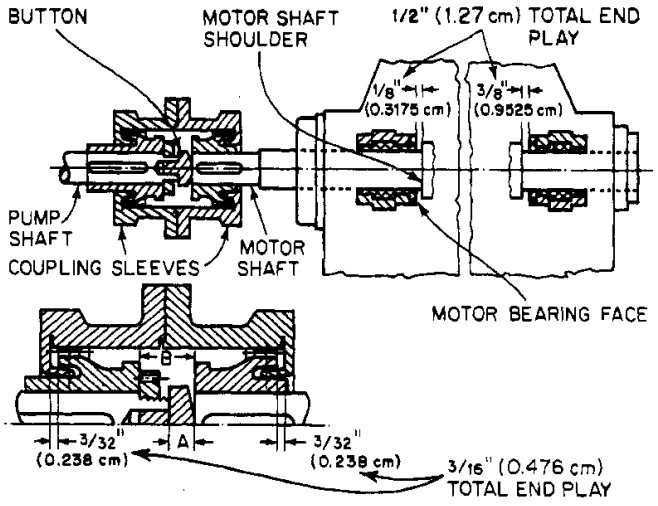


FIGURE 20 Motor-driven coupling end float limited by insertion of button between shaft ends (Flowserve Corp.)

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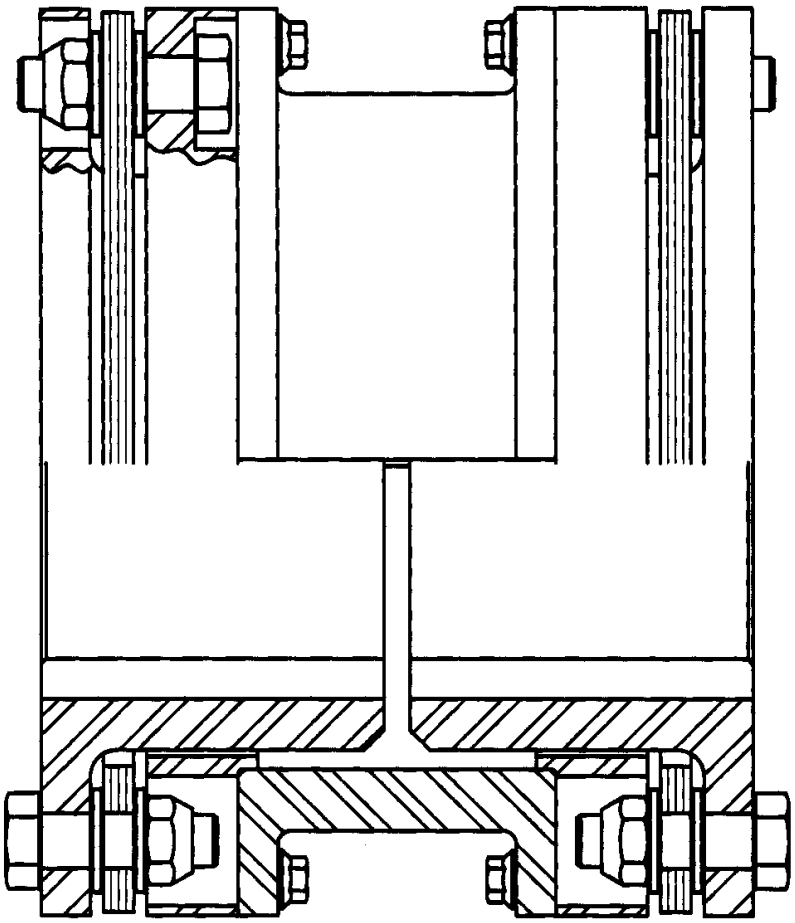


FIGURE 21 Close-coupled metal disk coupling having inherent float-restricting characteristics. (Falk Rexnord Couplings)

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

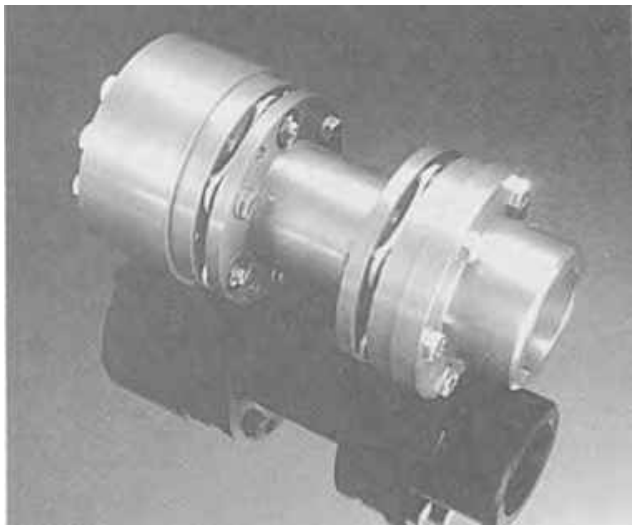


FIGURE 22 Spacer type metal disc coupling (Lovejoy Inc)

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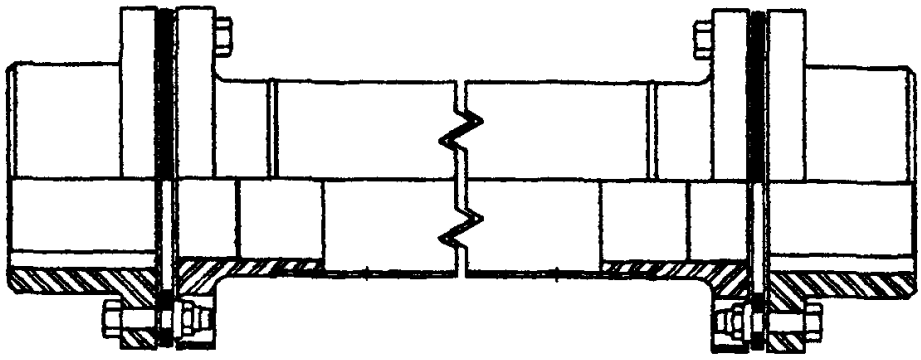


FIGURE 23 Gear coupling with tubular spacer (Lovejoy Inc)

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.

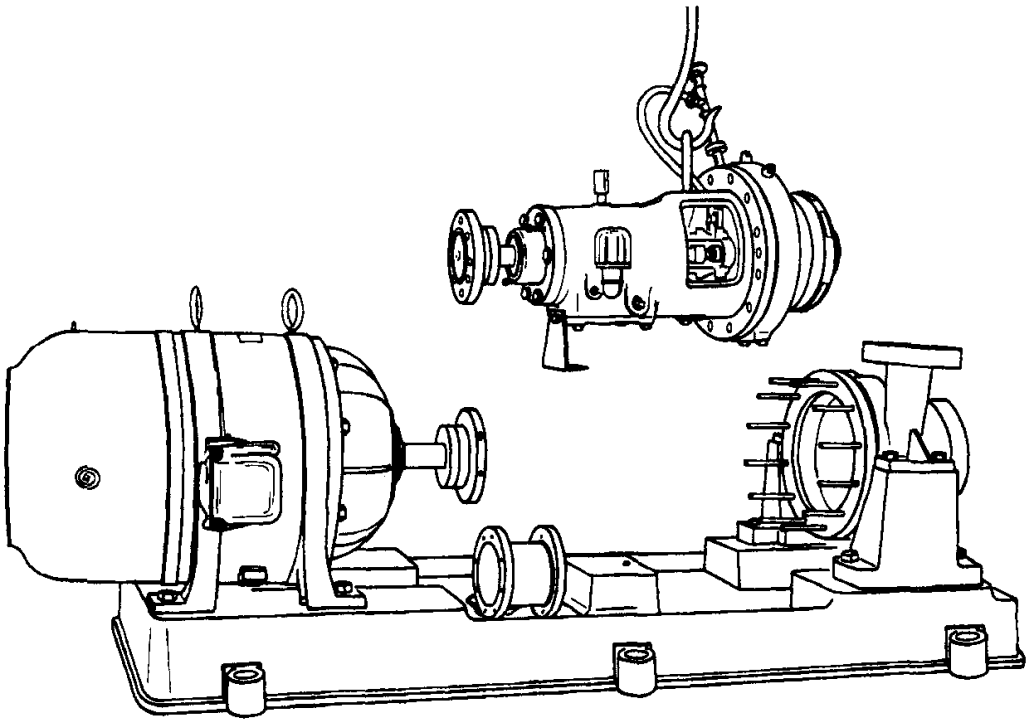


FIGURE 24 Spacer coupling enables end suction pump to be dismantled without disturbing piping, pump casing, or driver (Flowserve Corp)

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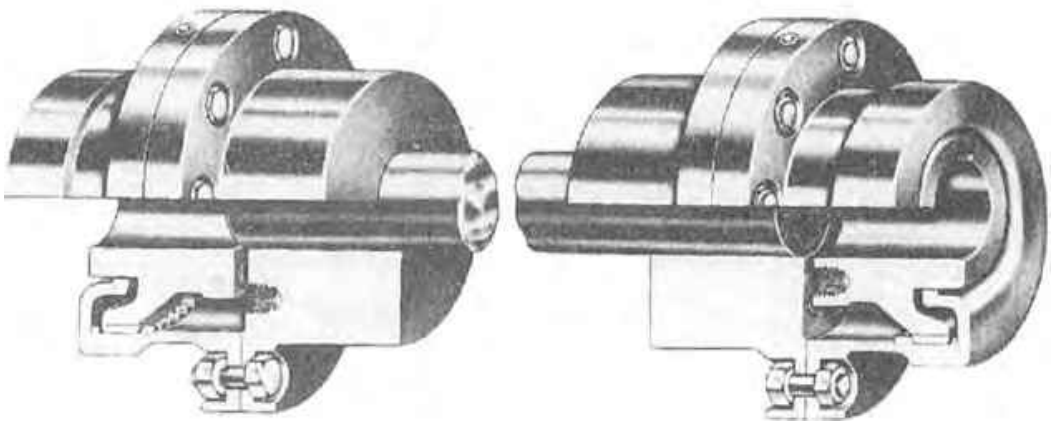


FIGURE 25 Section of floating shaft and coupling (Kop-Flex EPT)

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).

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FIGURE 26 Tubular intermediate shafting and universal joints (H. S. Watson)

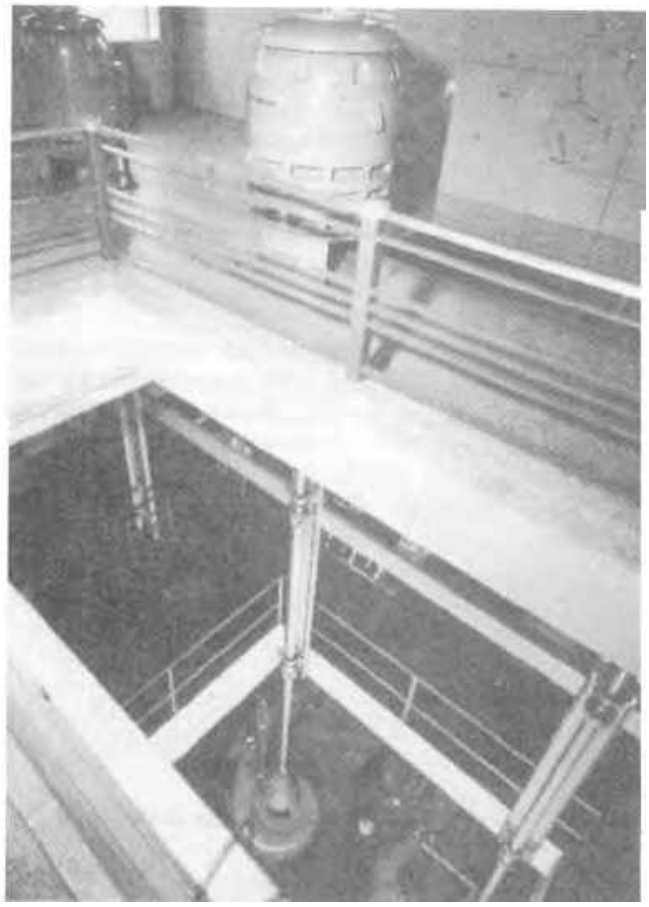


FIGURE 27 Sections of flexible shafts and intermediate guide bearings (Flowserve Corp)

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

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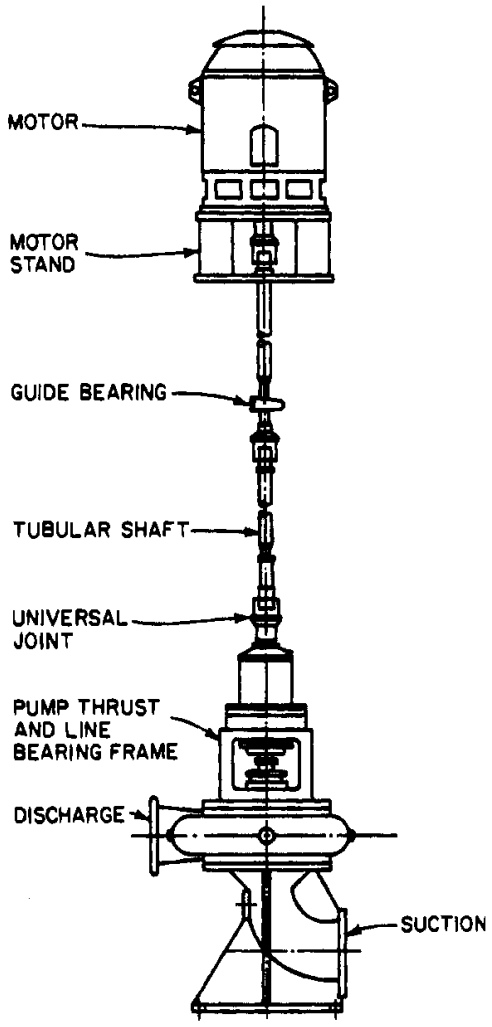


FIGURE 28 Vertical pump with flexible shafting (Flowserve Corp)

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:

$$S_s = \frac{16T}{\pi D_o^3} \qquad \text{for solid shafting}$$

$$S_s = \frac{16T}{\pi D_o^3 \left(1 - \frac{D^4}{D_o^4}\right)} \qquad \text{for tubular shafting}$$

where S_S = torsional shear stress, lb/in^2 (N/m^2)
T = transmitted torque, in · lb (N · m)
D_o = shaft outer diameter, in (m)
D = shaft inner diameter (tubular shaft only), in (m)

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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

$$I = \frac{\pi D_o^4}{64} \qquad \text{for solid shafting}$$

$$I = \frac{\pi(D_o^4 - D^4)}{64} \qquad \text{for tubular shafting}$$

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/in^2 (N/m^2)
I = moment of inertia, in^4 (m^4)

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 hearings. 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 hearing 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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1. American Petroleum Institute Standard 610. "Centrifugal Pumps for Petroleum, Petrochemical and Natural Gas Industries", 10th ed., 1220 L Street, Washington, DC 20005, 2004.

FURTHER READING

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ANSI/AGMA 9112 *Bores and Keyways for Flexible Couplings (Metric Series).*

ANSI/AGMA 9003 *Flexible Couplings—Keyless Fits.*

ANSI/AGMA 9004 *Flexible Couplings—Mass Elastic Properties and Other Characteristics.*

American Petroleum Institute Standard 671. "Special Purpose Couplings for Petroleum, Chemical, and Gas Industry Services." 3rd ed., 1220 L Street, Washington, DC 20005, October, 1998.

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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/in^2 to about 0.37 lb/in^2 (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 μbar). The sound pressure level in decibels is equal to 20 times the common logarithm of this ratio:

$$SPL = 20 \log\left(\frac{P}{0.0002}\right) \tag{1}$$

where P is the rms sound pressure in microbars.

*1 bar = 10^5 Pa.

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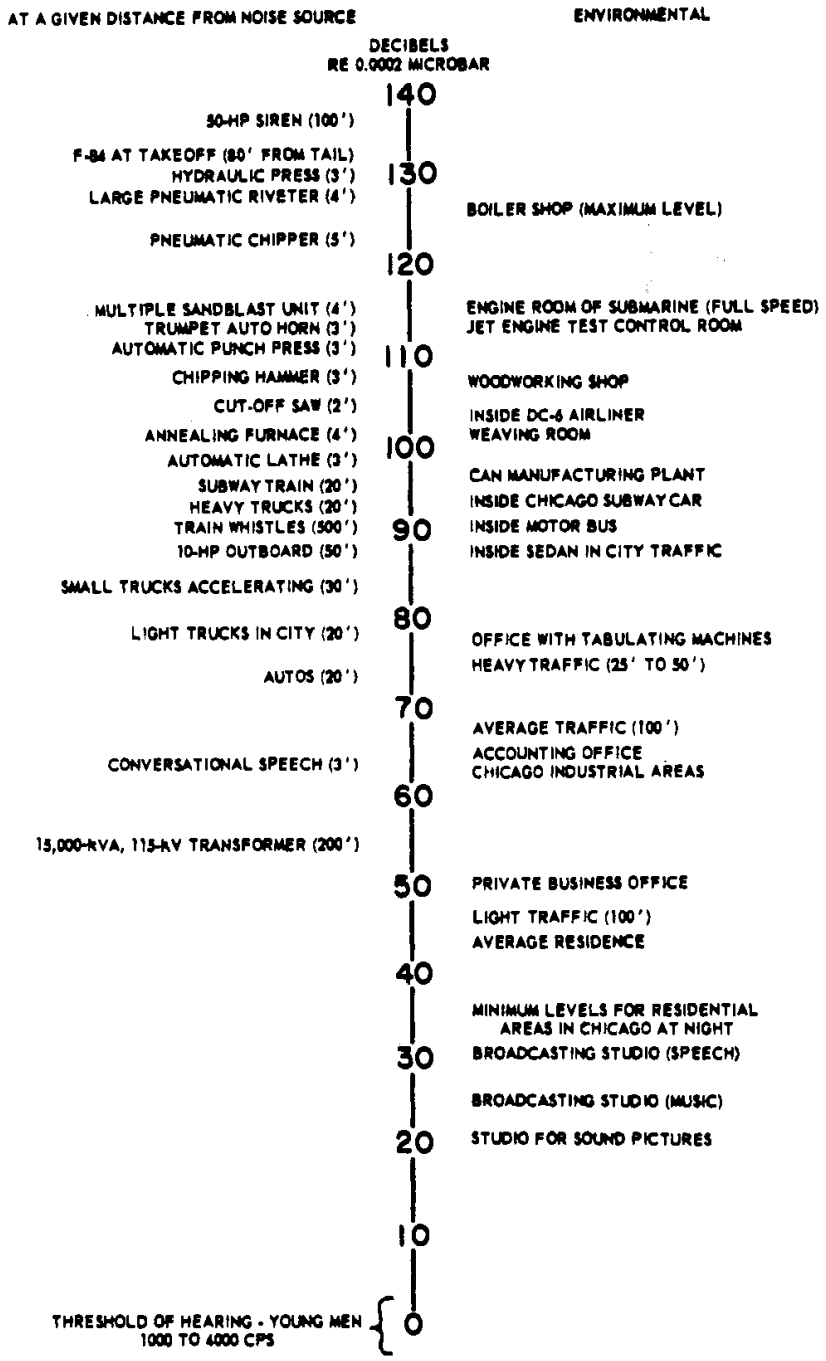


FIGURE 1 Typical overall sound levels (1 ft = 0.3048 m) (Reference 15)

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.

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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 broad-band 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

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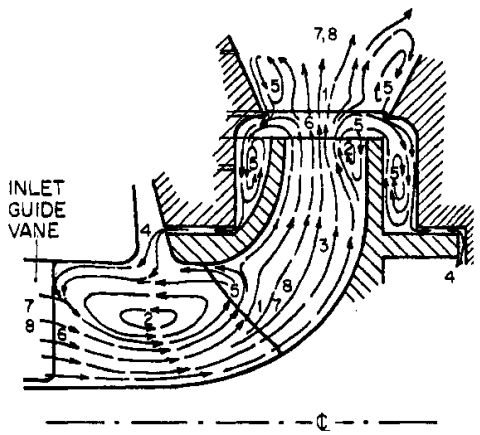


FIGURE 2 Secondary flows in and around a pump impeller stage (Reference 12)

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, . . .
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)

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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.

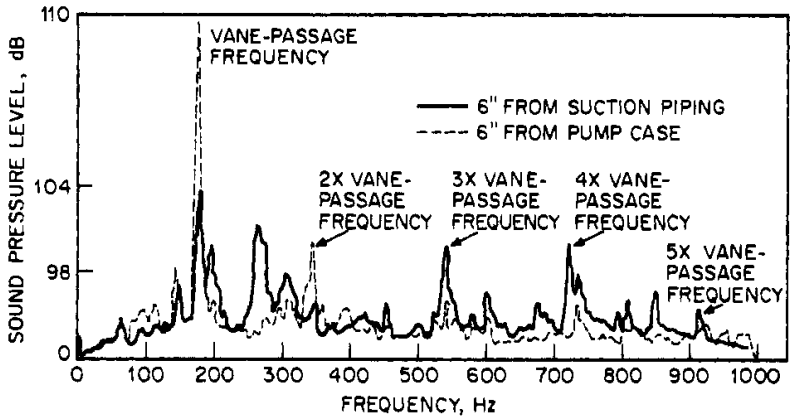


FIGURE 3 Noise spectra of cavitation and vane passage on a centrifugal pump

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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.

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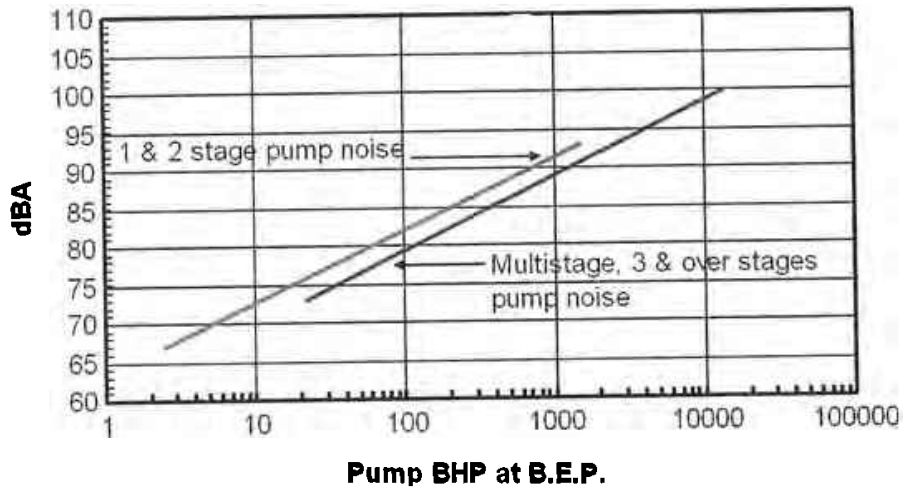


FIGURE 4 Estimate of dBA at BEP using pump BHP (BHP = 1.34 × kW)

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

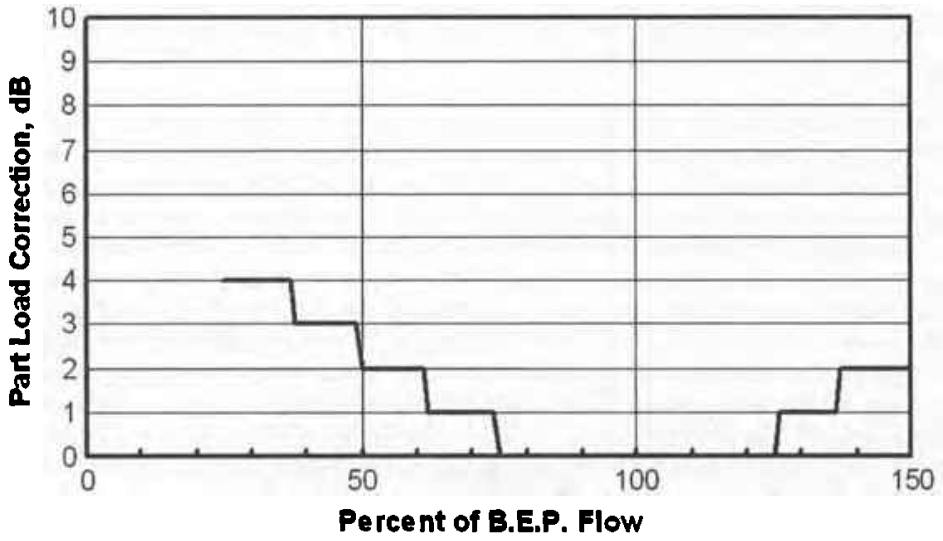


FIGURE 5 Part load correction to overall level at BEP

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TABLE 2 Octave, Mid-Band Corrections to Ocerall 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.

$$\text{dB Reduction} = 20\,Log_{10}\,(\text{Distance in meters}) \tag{3}$$

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.

$$\text{dB Total} = 10\,Log\,\Sigma\,10^{La/10} \tag{4}$$

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

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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 dB mid-band		–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

$$\text{dB Total} = 10Log(10^{82/10} + 10^{91/10} + 10^{88/10}) = 93.1 \text{ dbA}$$

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 overside) 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.

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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 Kondo[10] 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

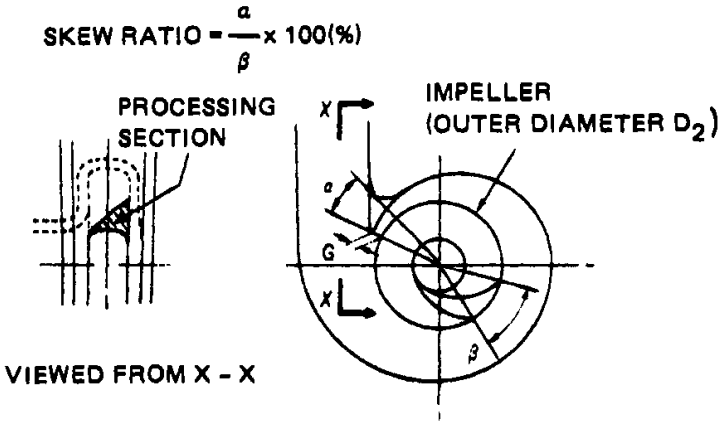


FIGURE 6 Cutwater of a spiral casing and impeller (Reference 10)

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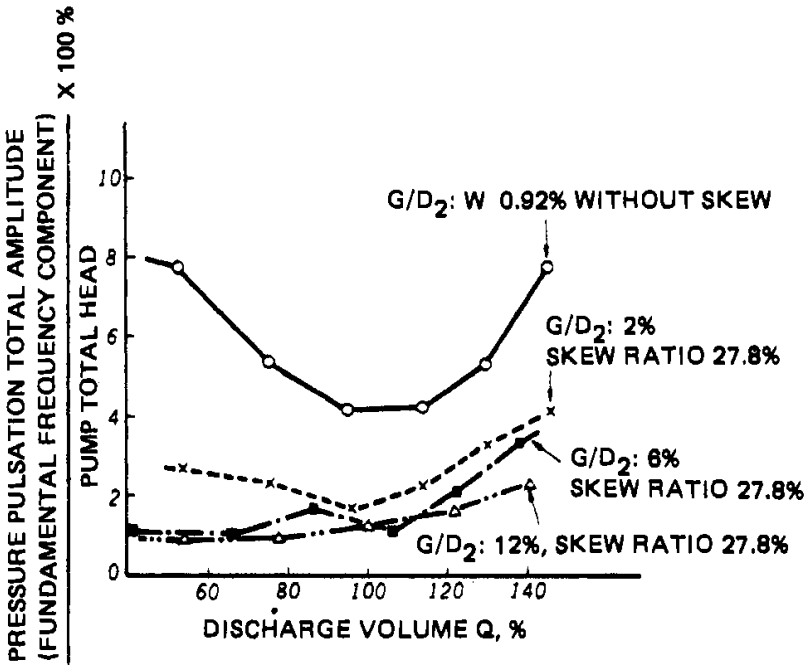


FIGURE 7 Effects of gap width between cutwater of a spiral casing and trailing edge of the impeller blade (Reference 10)

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 Zogg[11] 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

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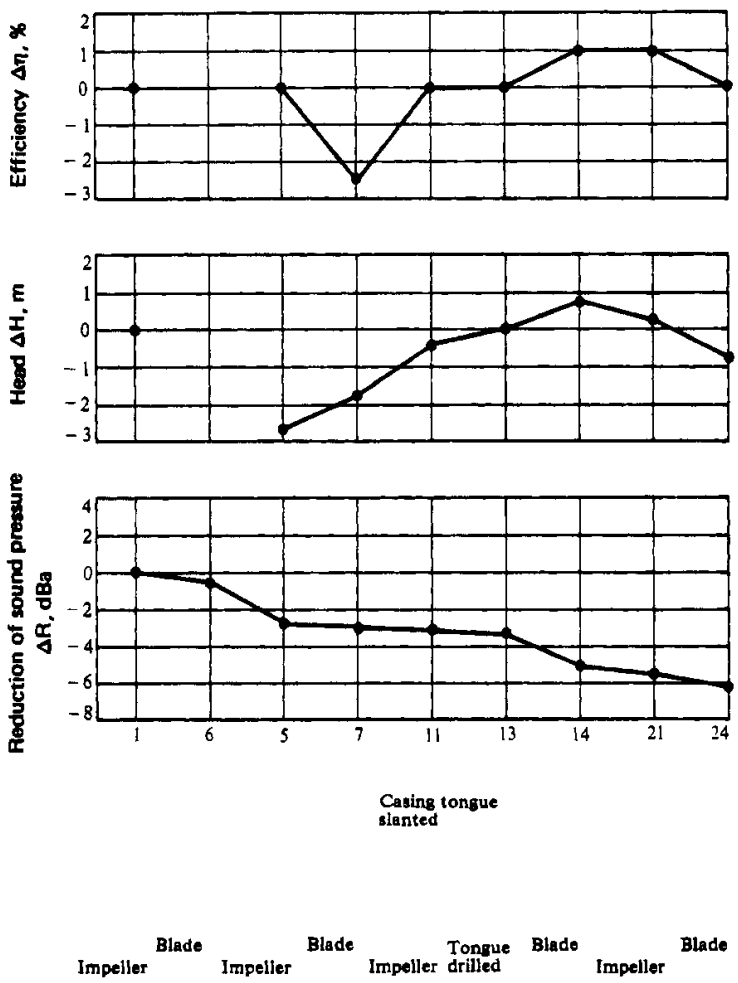


FIGURE 8 Reduction in sound pressure level with influences on head and efficiency at 100% flow for various impeller blade and casing tongue configurations (Reference 11)

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.

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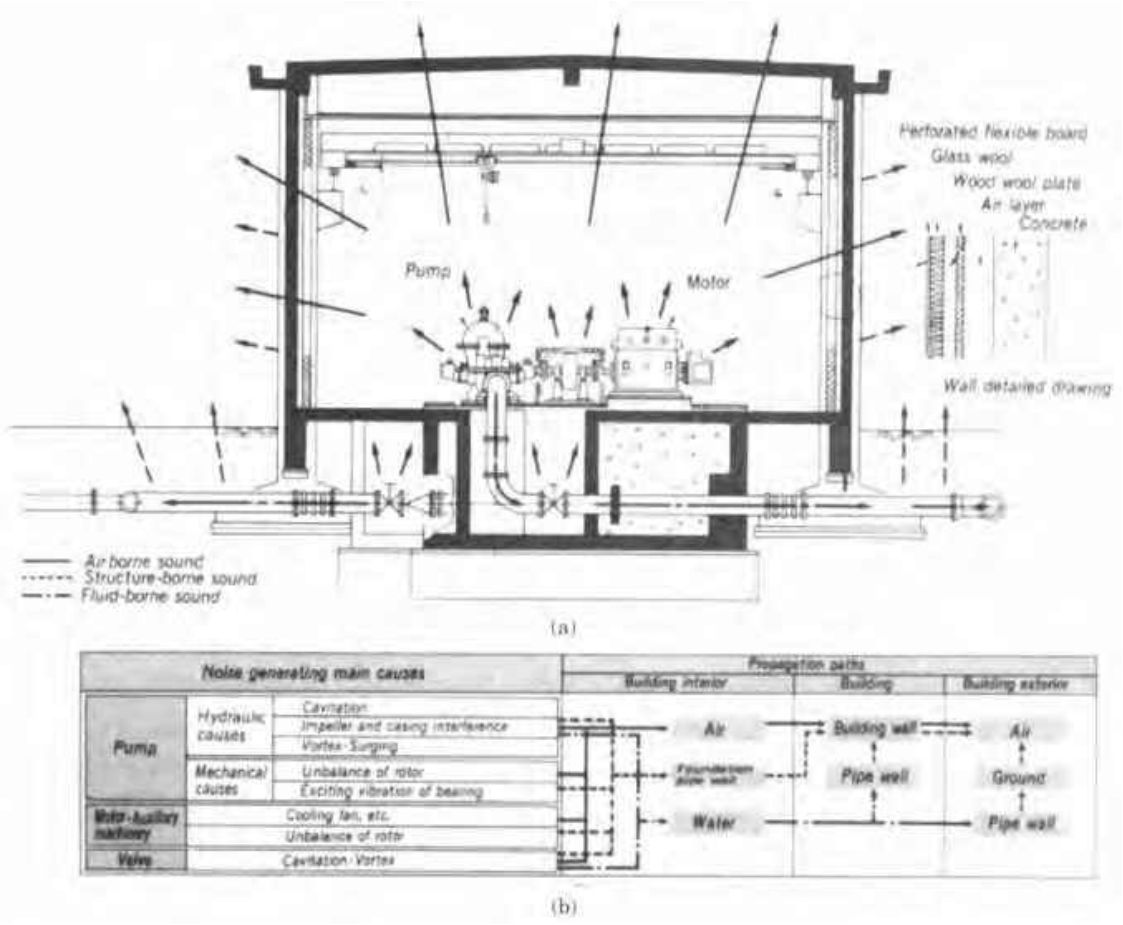


FIGURE 9 Noise propagation paths in a pumping plant (Reference 10)

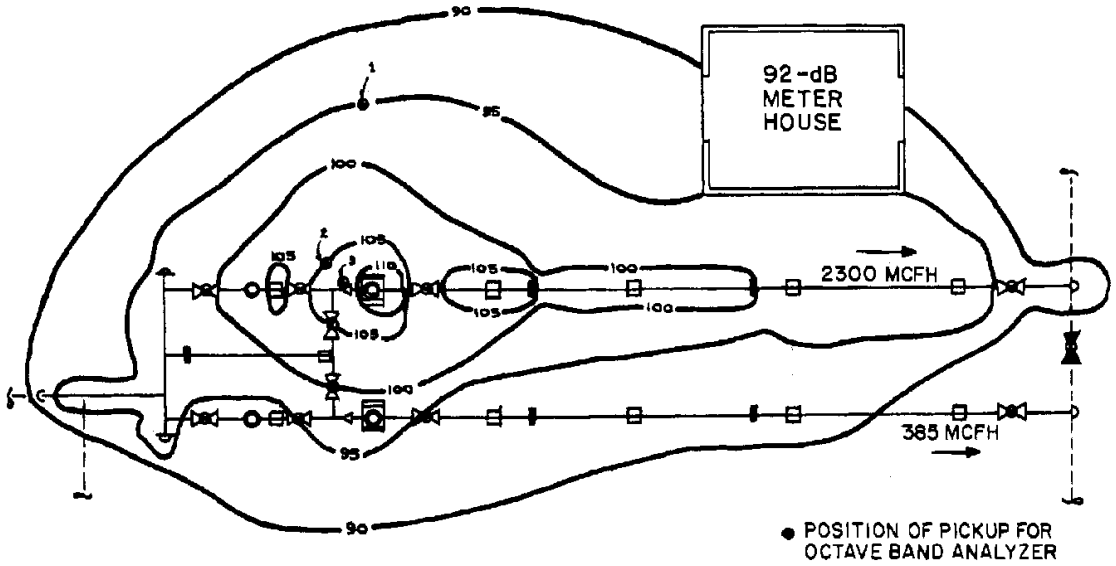


FIGURE 10 Contours of equal sound level (decibels) (Reference 14)

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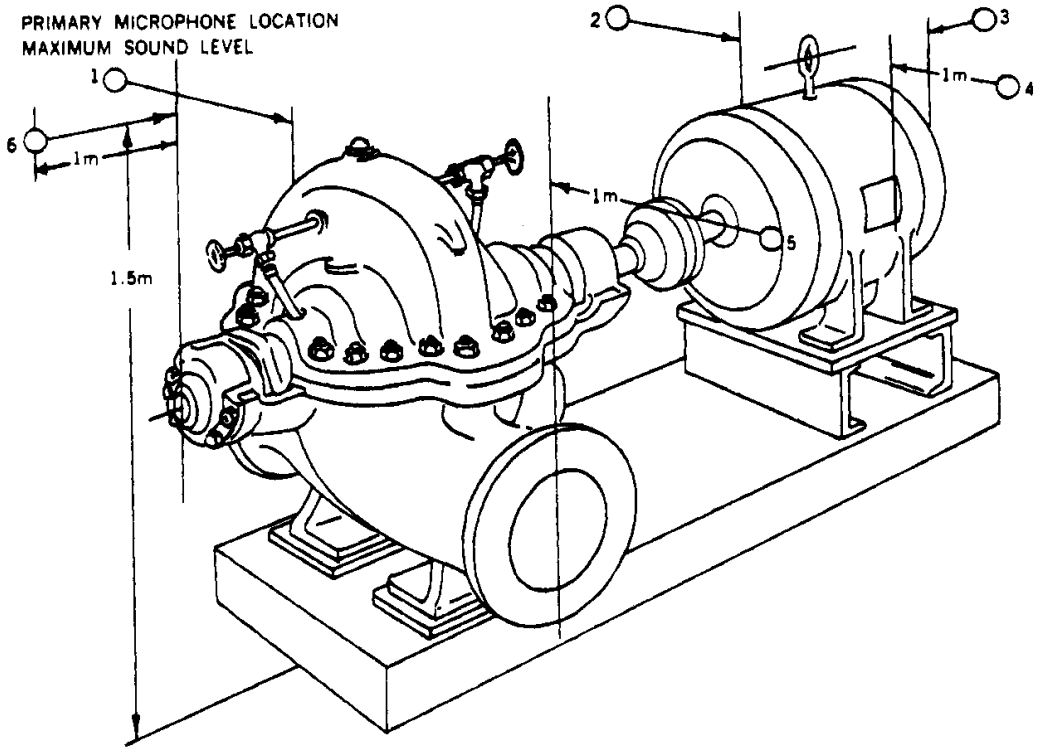


FIGURE 11 Placement of microphones on a horizontally split centrifugal pump (Hydraulic Institute ANSI/HI 2000 Edition Pump Standards, Reference 3)

The Hydraulic Institute standard[3] 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

$$NR = 10 \log \frac{\bar{\alpha}_a}{\bar{\alpha}_b} \tag{5}$$

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:

$$\bar{\alpha} = \frac{\alpha_1 A_1 + \alpha_2 A_2 + \alpha_3 A_3 + \cdots \alpha_n A_n}{A_1 + A_2 + A_3 + \cdots A_n} \tag{6}$$

where $\alpha_1, \alpha_2, \alpha_3, \ldots$ are absorption coefficients of various surface areas within the enclosure and $A_1, A_2, A_3, \ldots$ 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

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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 Microphone: 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. $2 \times 10^{-}$ N/m²	BACK-GROUND	LOCATION *										AV.
			1	2	3	4	5	6	7	8	9	10	
MIDBAND FREQ. Hz	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/ft^3 (80 to 96 kg/m^3), found in the rigid fiberglass board, are normally more effective than the 1 to 2 lb/ft^3 (16 to 32 kg/m^3) 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

[a]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

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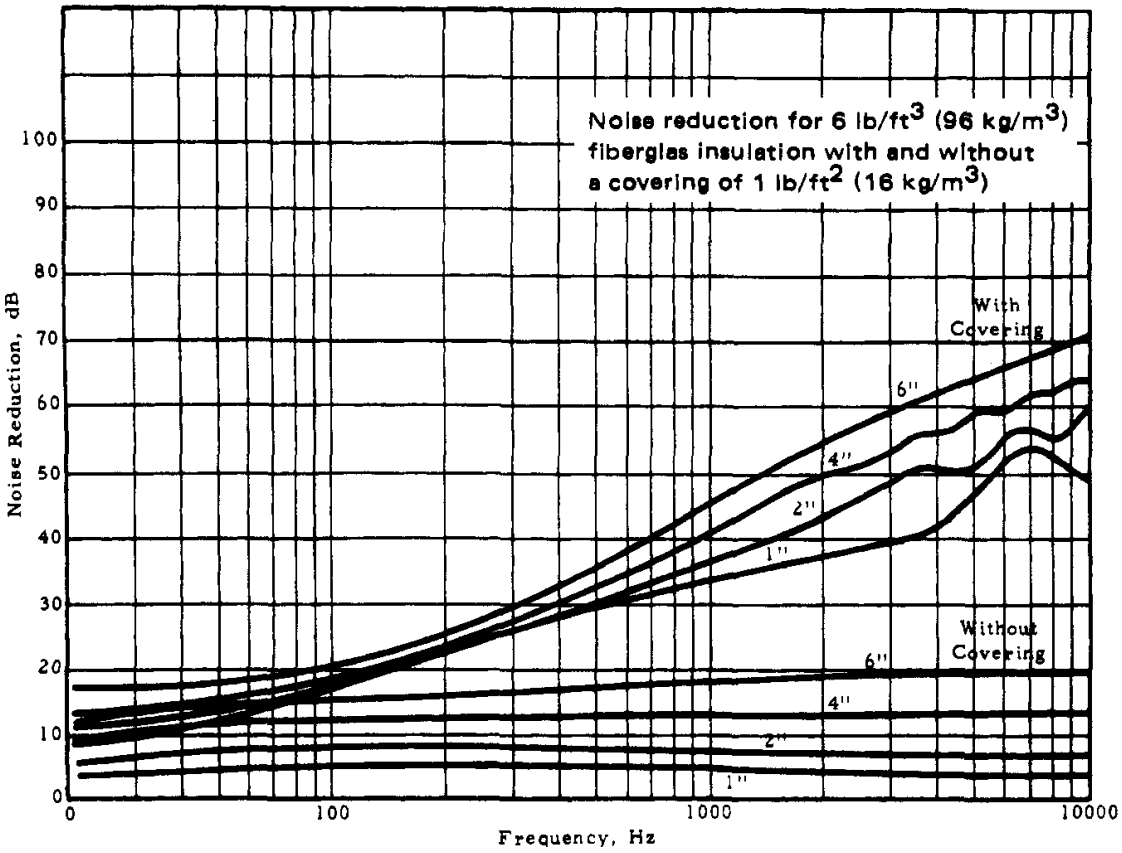


FIGURE 13 Noise reduction for piping with fiberglass insulation (1 in = 2.54 cm)

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

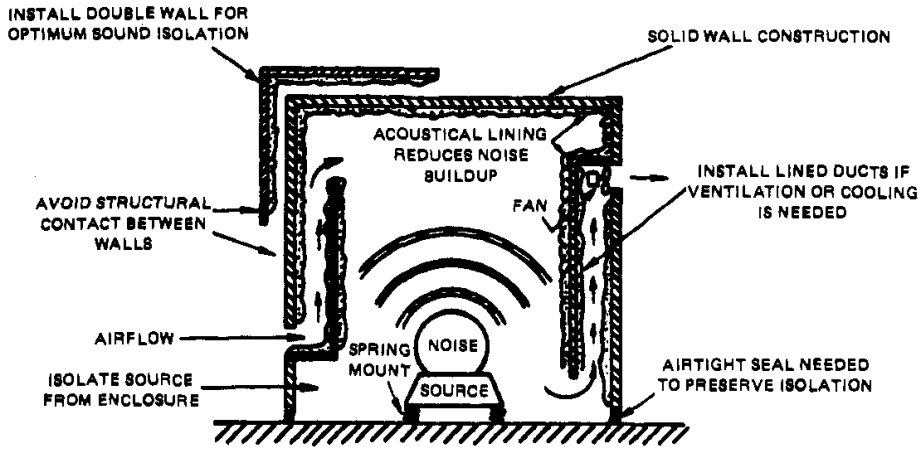


FIGURE 14 Design of an effective sound-isolating enclosure requiring air circulation for cooling (Reference 9)

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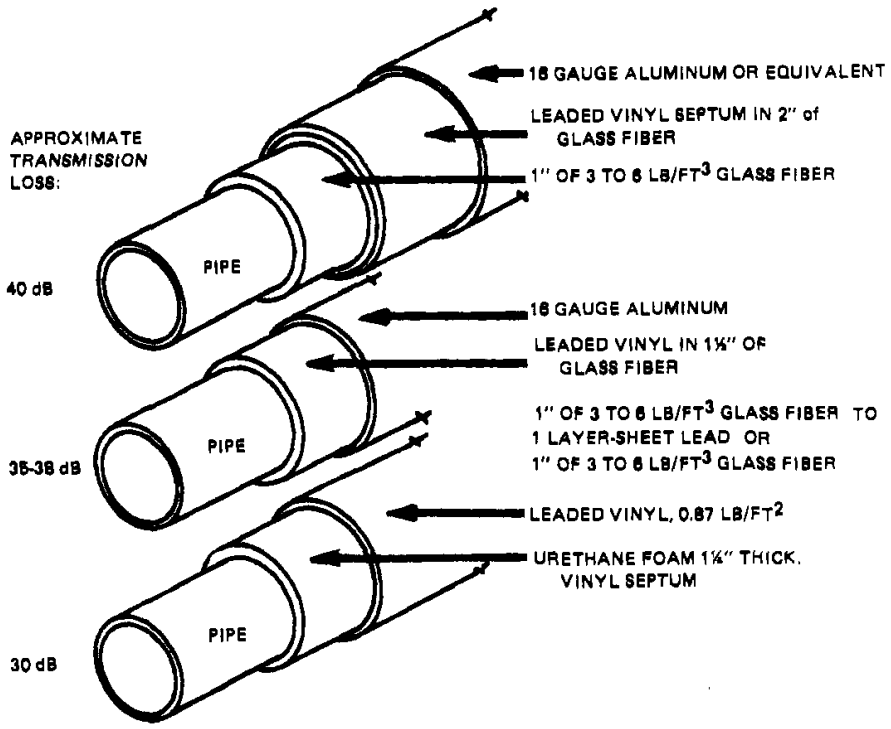


FIGURE 15 Three possible pipe wrap configurations (1 to = 2.54 cm; 1 lb/ft^3 = 16 kg/m^3) (Reference 14)

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

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TABLE 7 OSHA noise exposure limits

	Allowable exposure time	
Sound level, dBA	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 Kondo[10] 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

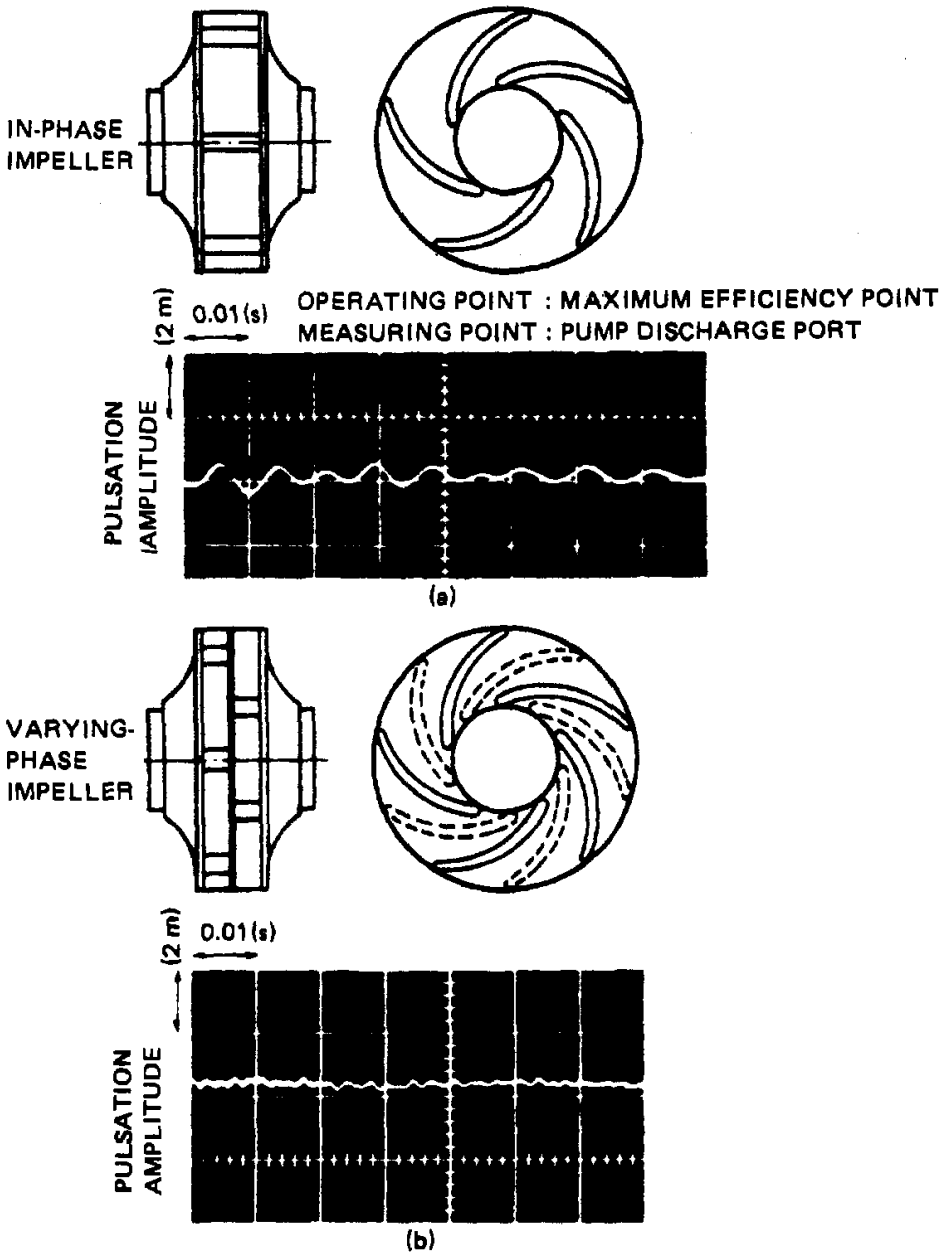


FIGURE 16 Comparison of pressure pulsation waveforms: (a) a double-suction in-phase centrifugal pump impeller and (b) a varying-phase (staggered) centrifugal pump impeller (Reference 10)

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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 bar[1]). 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

[1]1 bar = 10^5 Pa.

[2]Work 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.

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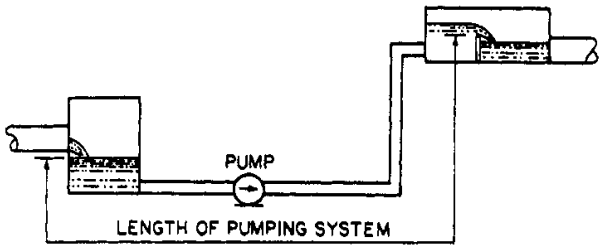


FIGURE 1 Length of system controlled by pump

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

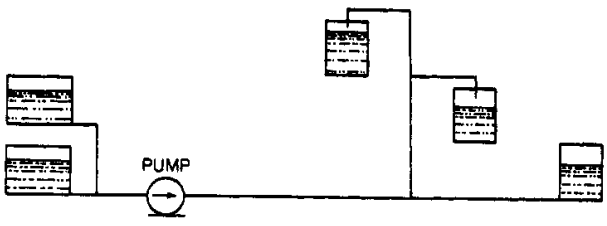


FIGURE 2 Branch-line pumping system

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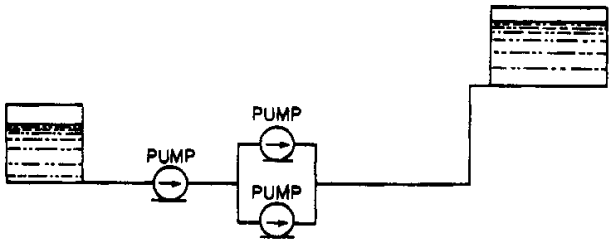


FIGURE 3 Pumps in series and in parallel

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

$$H = \frac{V^2}{2g} + \frac{p}{\gamma} + Z \tag{1}$$

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/s^2 (9.807 m/s^2)
p = pressure, lb/ft^2 (N/m^2)
γ = specific weight (force) of liquid, lb/ft^3 (N/m^3)
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

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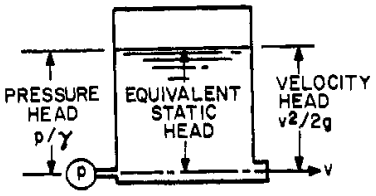


FIGURE 4 Equivalent static head

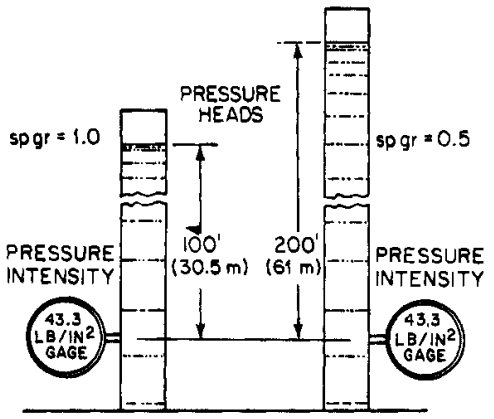


FIGURE 5 Liquids of different specific weights (also specific gravities) require different column heights or pressure heads to produce the same pressure intensity (43.3 lb/in² = 298.5 kPa)

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):

$$H = \frac{V^2}{2g} + \frac{p}{\gamma} + Z$$

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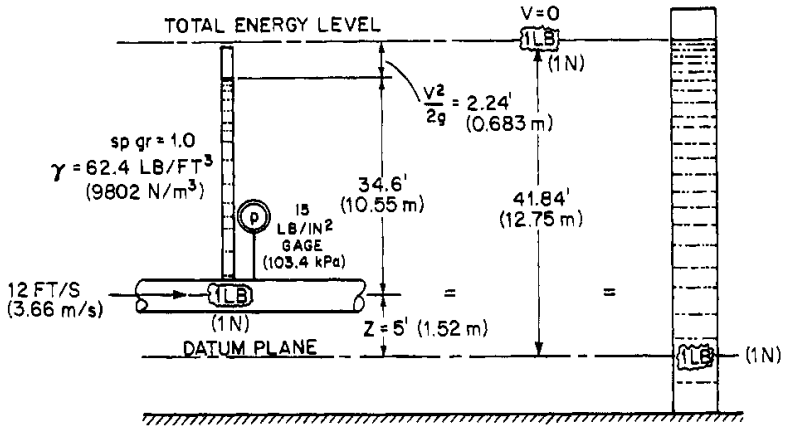


FIGURE 6 The total head, or energy, in foot-pounds per pound (newtonmeters per newton) is equal to the sum of the velocity, pressure, and elevation heads relative to a datum place. A unit weight (force) of the same liquid or any liquid raised to rest at the height shown, or under a column of liquid of this same height, has the same head as the unit weight (force) of liquid shown flowing in the pipe.

In USCS units

$$H = \frac{12^2}{2 \times 32.17} + \frac{144 \times 15}{62.4} + 5$$
$$= 2.24 + 34.6 + 5$$
$$= 41.84 \text{ ft·lb/lb, or ft}$$

In SI units

$$H = \frac{3.66^2}{2 \times 9.807} + \frac{103.4 \times 1000}{9802} + 1.52$$
$$= 0.683 + 10.55 + 1.52$$
$$= 12.75 \text{ N·m/N, or m}$$

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 41.48 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,

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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:

$$h_1 = \frac{p_1}{\gamma_1}$$

$$p_1 = h_1\gamma_1 = p_2$$

$$h_2 = \frac{p_2}{\gamma_2} = \frac{h_1\gamma_2}{\gamma_2}$$

$$= \frac{\gamma_1}{\gamma_1} h_1 \tag{2}$$

$$= \frac{\text{sp. gr.}_1}{\text{sp. gr.}_2} h_1 \tag{3}$$

Therefore

in USCS units $$h_2 = \frac{13.6}{1} \times \frac{24}{12} = 27.2 \text{ ft abs}$$

in SI units $$h_2 = \frac{13.6}{1} \times \frac{61}{100} = 8.3 \text{ m abs}$$

Also, $h_2 = (27.2 - 13.6)(\frac{30}{12}) = -6.8$ ft $[(8.3 - 13.6)(\frac{76}{100}) = -2.04$ 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

$$TH = H_d - H_s = \left(\frac{V_d^2}{2g} + \frac{p_d}{\gamma_d} + Z_d\right) - \left(\frac{V_s^2}{2g} + \frac{p_s}{\gamma_s} + Z_s\right) \tag{4}$$

The equation for pump differential pressure P_Δ in pounds per square foot (newtons per square meter) is

$$P_\Delta = P_d - P_s = \left[p_d + \gamma_d\left(Z_d + \frac{V_d^2}{2g}\right)\right] - \left[p_s + \gamma_s\left(Z_s + \frac{V_s^2}{2g}\right)\right] \tag{5}$$

$$P_\Delta \text{ in lb/in}^2 = P_\Delta \text{ in lb/ft}^2 \div 144$$

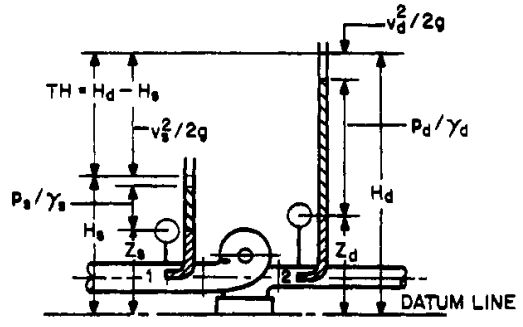


FIGURE 7 Centrifugal pump total head

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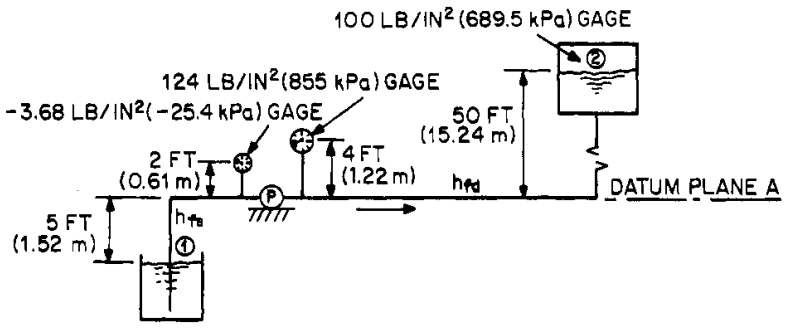


FIGURE 8 Example 1

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_Δ 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

$$TH = \frac{P_\Delta}{\gamma} \tag{6}$$

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:

$$\begin{aligned} TH &= (H_2 - H_1) + \Sigma h_{f(1-2)} \\ &= \left(\frac{V_2^2}{2g} + \frac{p_2}{\gamma} + Z_2\right) - \left(\frac{V_1^2}{2g} + \frac{p_1}{\gamma} + Z_1\right) + \Sigma h_{f(1-2)} \end{aligned} \tag{7}$$

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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/in^2 (N/m^2) gage or (+) lb/in^2 (N/m^2) abs
Z = elevation above (+) or below (–) datum plane, ft (m)
γ = specific weight (force) of liquid (assumed the same between points), lb/ft^3 (N/m^3)
g = acceleration of gravity, 32.17 ft/s^2 (9.807 m/s^2)
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:

In feet
$$\frac{p}{\gamma} = \frac{0.016 \text{ lb/ft}^2}{\text{sp. gr.}} \text{ or } \frac{2.3 \text{ lb/in}^2}{\text{sp. gr.}} \tag{8a}$$

In meters
$$\frac{p}{\gamma} = \frac{0.102 \text{ kPa}}{\text{sp. gr.}} \text{ or } \frac{1.02 \times 10^{-3} \text{ bar}}{\text{sp. gr.}} \tag{8b}$$

The velocity in a pipe may be calculated as follows:

In feet per second
$$V = \frac{(\text{gm})(0.408)}{(\text{pipe ID in inches})^2} \tag{9a}$$

In meters per second
$$V = \frac{(\text{m}^3/\text{h})(3.54)}{(\text{pipe ID in cm})^2} \text{ or } \frac{(\text{liters/s})(12.7)}{(\text{pipe ID in cm})^2} \tag{9b}$$

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 m^3/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 mm), h_{fd} = 25 ft (7.62 m).

In USCS units
$$\text{Calculated pipe velocity} = \frac{(\text{gpm})(0.408)}{(\text{ID in inches})^2}$$
$$V_s = \frac{1000 \times 0.408}{8^2} = 6.38 \text{ ft/s}$$
$$V_d = \frac{1000 \times 0.408}{6^2} = 11.33 \text{ ft/s}$$

In SI units
$$\text{Calculated pipe velocity} = \frac{(\text{m}^3/\text{h})(3.54)}{(\text{ID in cm})^2}$$

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$$V_s = \frac{227 \times 3.54}{20.3^2} = 1.95 \text{ m/s}$$

$$V_d = \frac{227 \times 3.54}{15.2^2} = 3.48 \text{ m/s}$$

(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,

in USCS units $$\frac{p}{\gamma} = \frac{2.31 \text{ lb/in}^2}{\text{sp. gr.}}$$

in SI units $$\frac{p}{\gamma} = \frac{0.102 \text{ kPa}}{\text{sp. gr.}}$$

Therefore,

in USCS units

$$TH = \left(\frac{11.33^2}{2 \times 32.3} + \frac{2.31 \times 124}{0.8} + 4\right) - \left(\frac{638^2}{2 \times 32.2} + \frac{2.31(-3.68)}{0.8} + 2\right)$$
$$= 364 - (-8) = 372 \text{ ft} \cdot \text{lb/lb, or ft}$$

In SI units

$$TH = \left(\frac{3.48^2}{2 \times 9.807} + \frac{0.102 \times 855}{0.8} + 1.22\right) - \left(\frac{195^2}{2 \times 9.807} + \frac{0.102(-25.4)}{0.8} + 2\right)$$
$$= 110.9 - (-2.43) = 113.3 \text{ N} \cdot \text{m/N, or m}$$

(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,

in USCS units $$\frac{p}{\gamma} = \frac{2.31 \text{ lb/in}^2}{\text{sp. gr.}}$$

in SI units $$\frac{p}{\gamma} = \frac{0.102 \text{ kPa}}{\text{sp. gr.}}$$

Therefore

in USCS units $$TH = \left(0 + \frac{2.31 \times 100}{0.8} + 50\right) - (0 + 0 - 5) + (3 + 25)$$
$$= 399 - (-5) + 28 = 372 \text{ ft} \cdot \text{lb/lb, or ft}$$

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

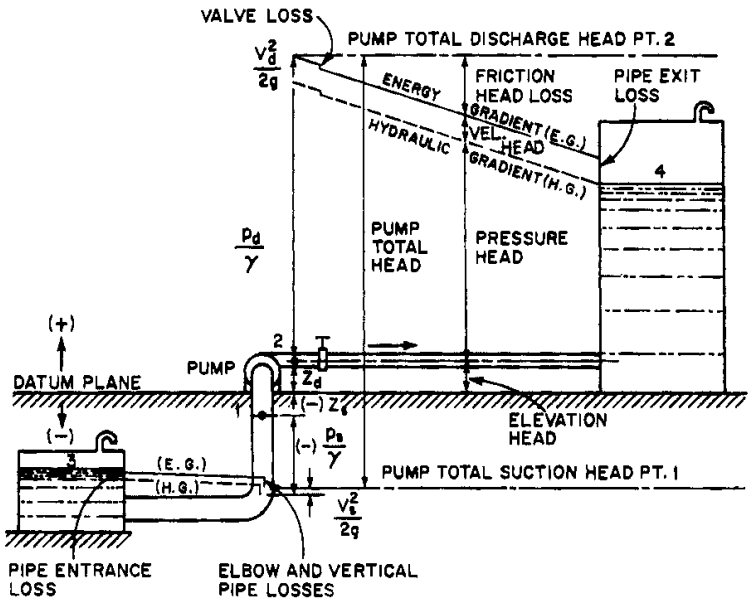


FIGURE 9 Energy and hydraulic gradients

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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

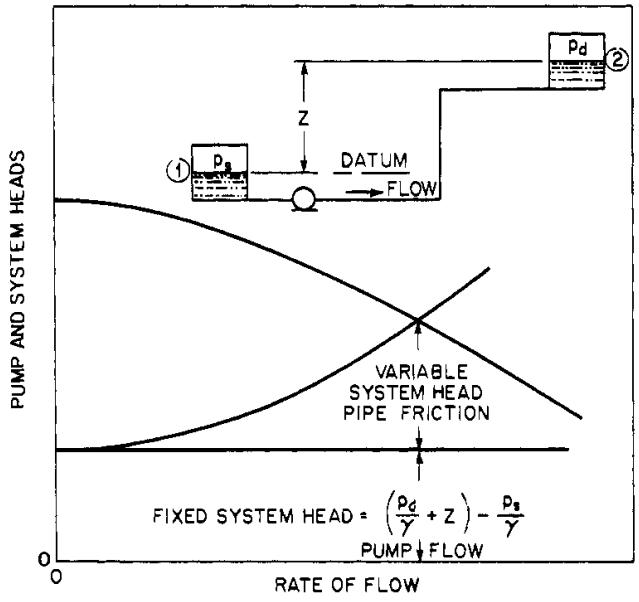


FIGURE 10 Construction of system total-head curve

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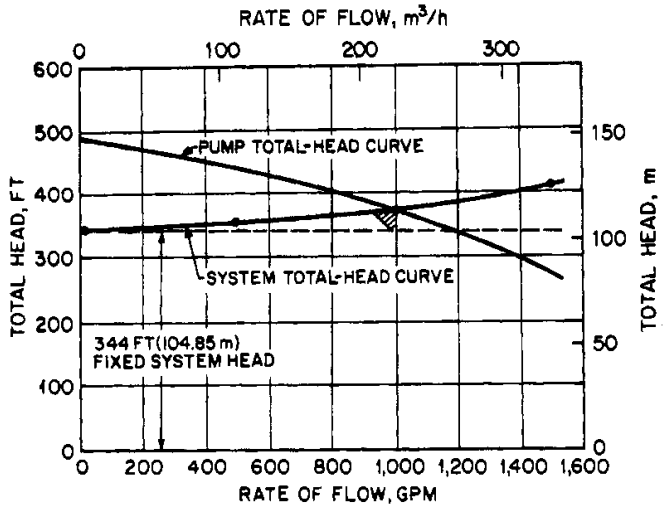


FIGURE 11 System total-head curve for Example 1

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/in^3 (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 m^3/h). The total head at other flow conditions is the fixed system head plus the variable system head multiplied by (gpm/1000)^2 (m^3/h ÷ 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 m^3/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

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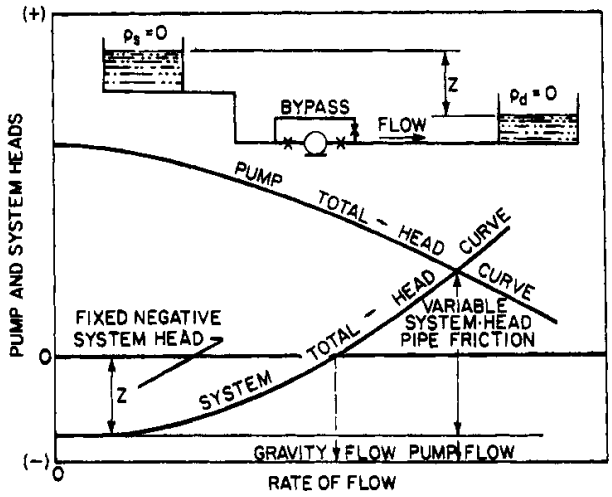


FIGURE 12 Construction of system total-head curve to determine gravity flow and centrifugal pump flow

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.

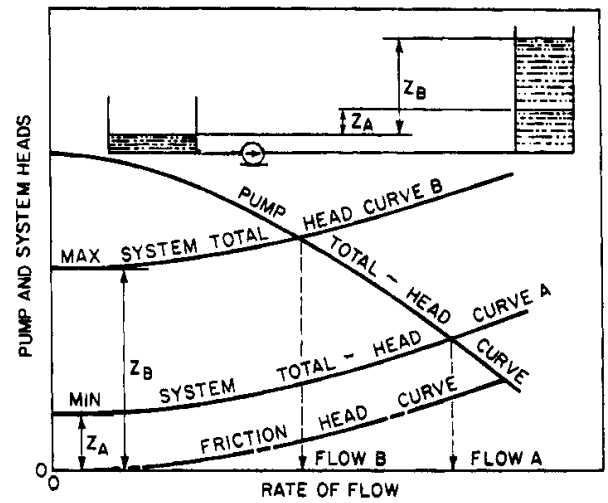


FIGURE 13 Construction of system total-head curves for a pumping system having variable static head

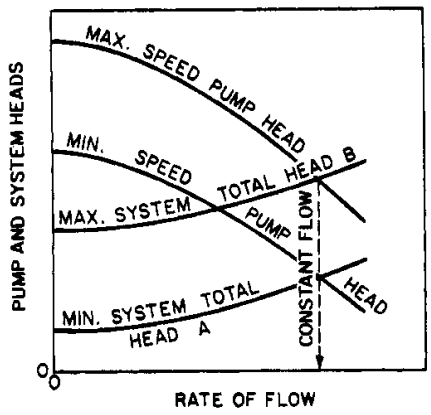


FIGURE 14 Varying centrifugal pump speed to maintain constant flow for the different reservoir levels shown in Figure 13

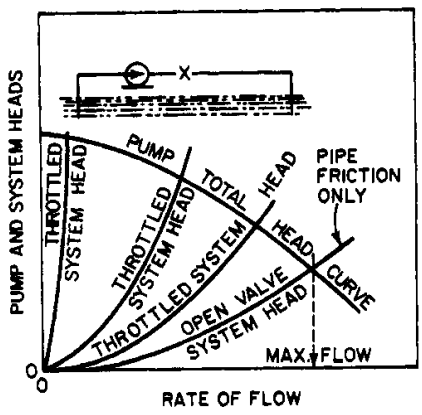


FIGURE 15 Construction of system total-head curves for various valve openings

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

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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

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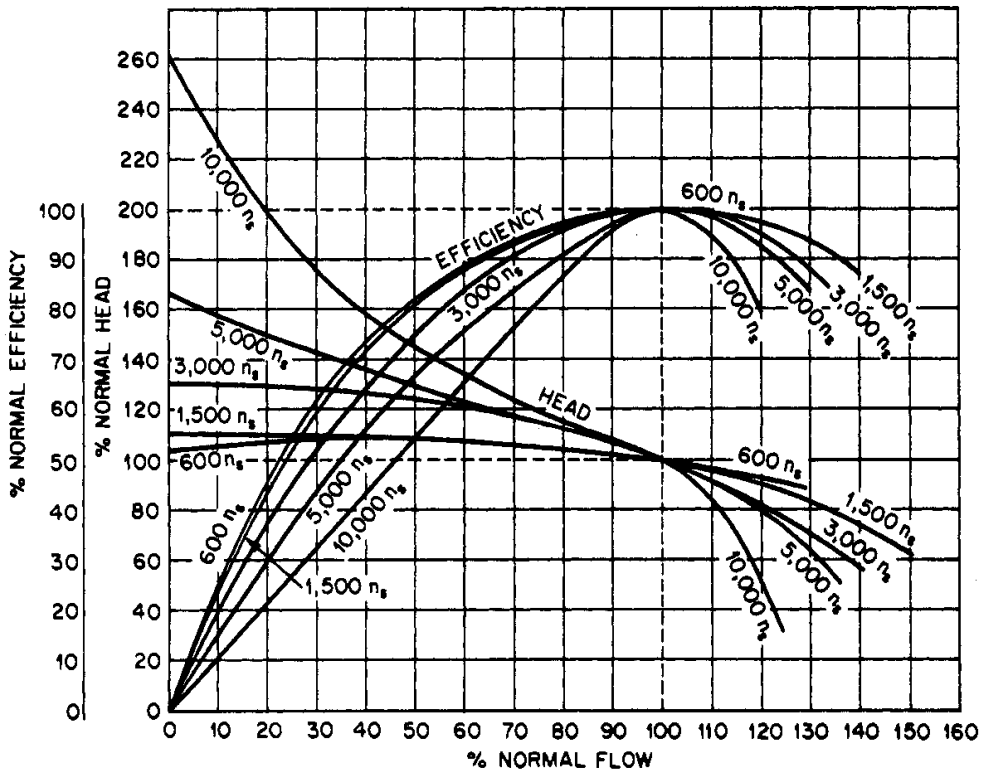


FIGURE 16A Approximate comparison of head and efficiency versus flow for impellers of different specific speeds in single-stage volute pumps. Specific speed n_s = rpm $\sqrt{\text{gpm}}/TH$ in $\text{ft}^{3/4}$
To convert to other units using
rpm, m^3/s, m: multiply by 0.01936; rpm, m^3/h, m: multiply by 1.163; rpm, L/s, m: multiply by 0.6123
To convert to the universal specific speed Ω_s (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

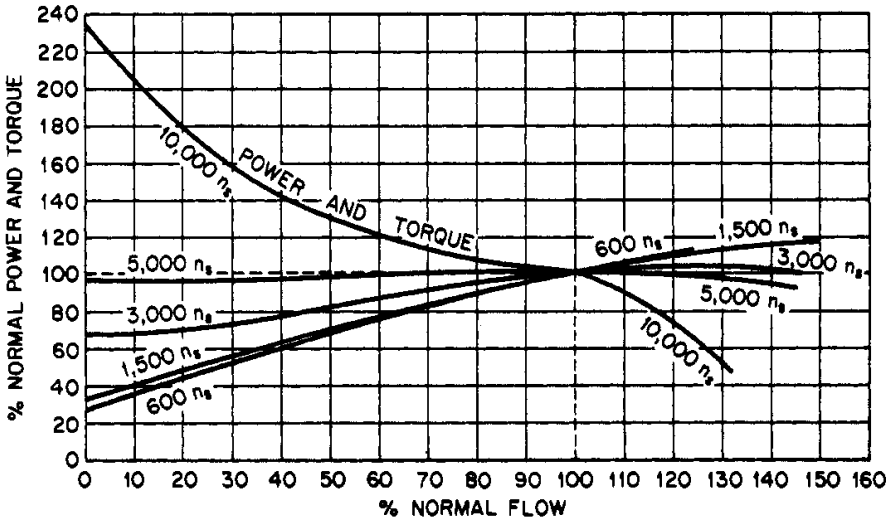


FIGURE 16B Approximate comparison of power and torque versus flow for impellers of different specific speeds in single-stage volute pumps

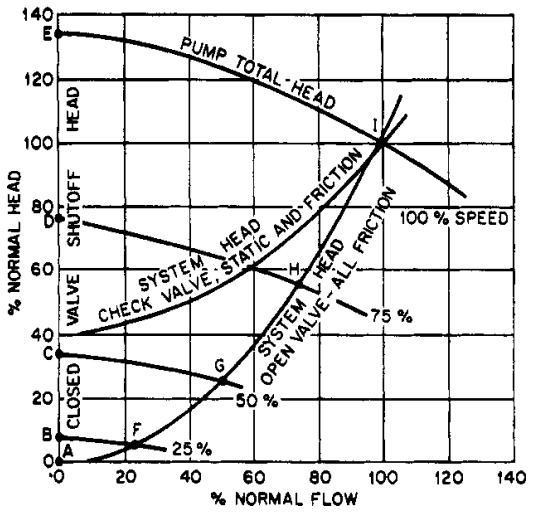


FIGURE 17 Variation in head when a centrifugal pump is started with a closed valve, an open valve, and a check valve

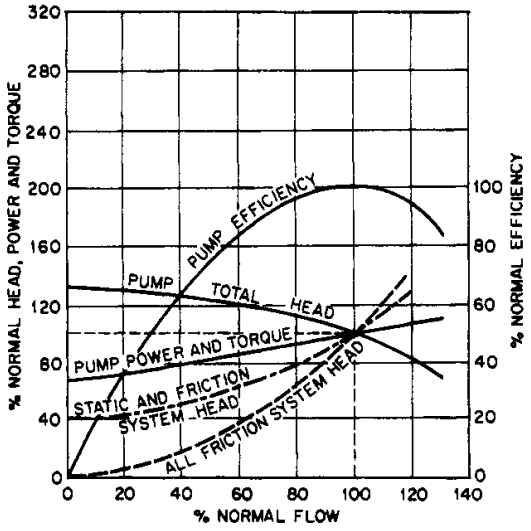


FIGURE 18 Typical constant-speed characteristic curves for a low-specific-speed pump

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]

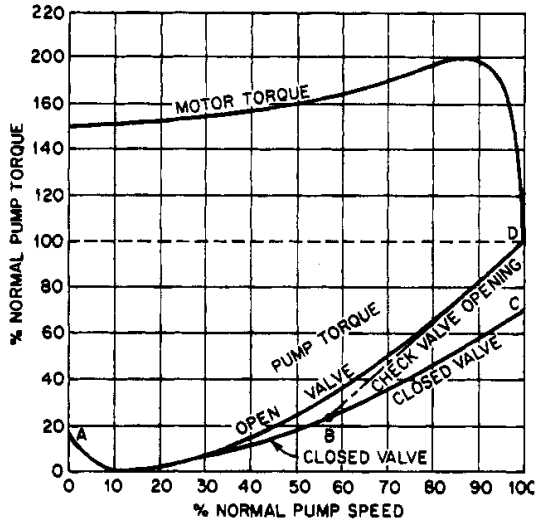


FIGURE 19 Variation of torque during start-up of a loss-specific-speed pump with a closed valve, an open valve, and a check valve. See Figure 18 for pump characteristics.

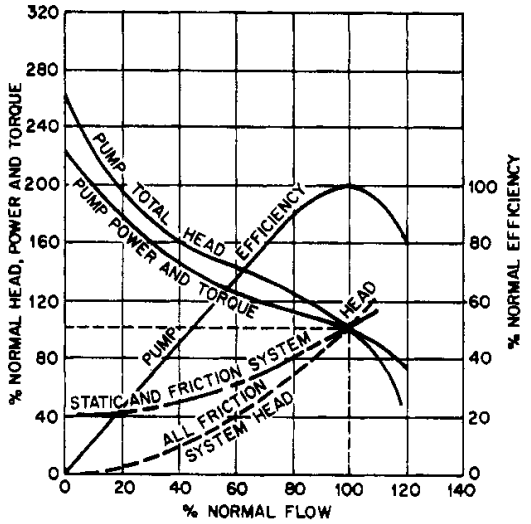


FIGURE 20 Typical constant-speed characteristic curves for a high-specific-speed pump

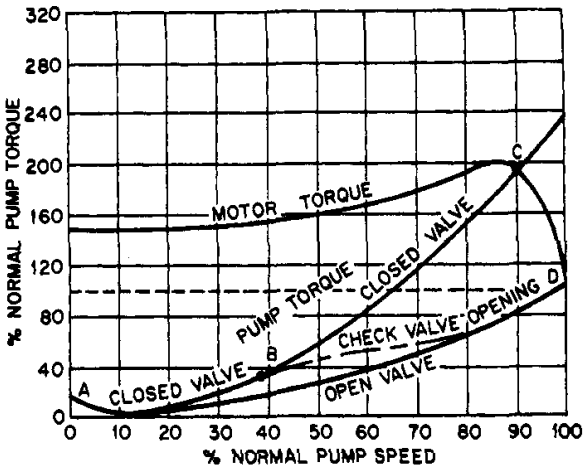


FIGURE 21 Variation of torque during start-up of a high-specific-speed pump with a closed valve, an open valve, and a check valve. See Figure 20 for pump characteristics.

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 cheek 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 and 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

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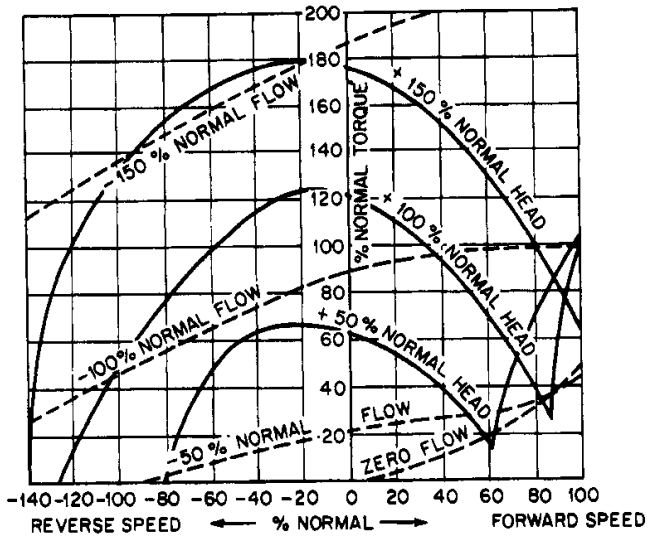


FIGURE 22 Typical reverse-speed-torque characteristics of a low-specific-speed, radial-flow, double-suction pump

little or no torque resistance, the pump will reach higher than normal forward speed in the reverse direction. This runaway speed will increase with specific speed and system head. Shown in Figures 22 and 23 are speed, torque, head, and flow, all expressed as a percentage of the pump design conditions for the normal forward speed. When a pump is running in reverse as a turbine under no load, the head on the pump will be static head (or head from other pumps) minus head loss as a result of friction due to the reverse flow.

If an attempt is made to start the pump while it is running in reverse, an electric motor must apply positive torque to the pump while the motor is initially running in a negative direction. Figures 22 and 23 show, for the two pumps, the torques required to decelerate, momentarily stop, and then accelerate the pump to normal speed. If either of these pumps were pumping into an all-static-head system, starting it in reverse would require overcoming 100% normal head, and it can be seen that a torque in excess of normal would be required by the driver while the driver is running in reverse. In addition to overcoming positive head, the driver must add additional torque to the pump to change the direction of the liquid. This could result in a prolonged starting time under higher than normal current demand. Characteristics of the motor, pump, and system must be analyzed together

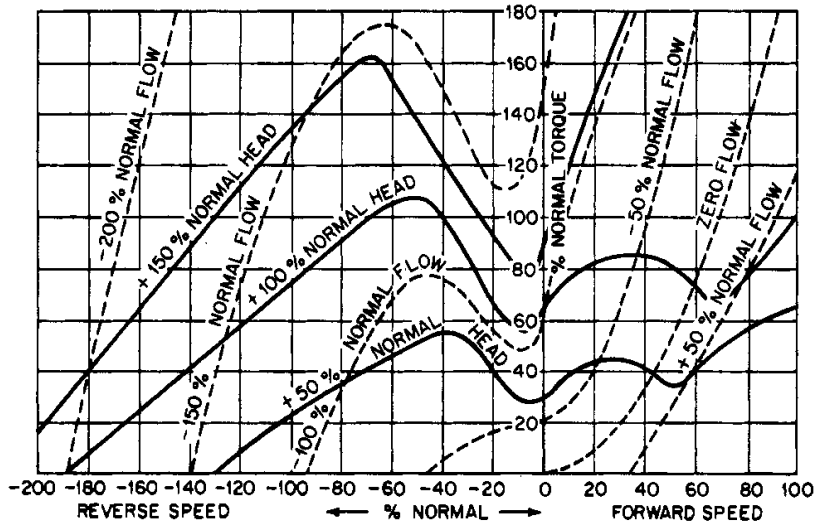


FIGURE 23 Typical reverse-speed-torque characteristics of a high-specific-speed, axial-flow, diffuser pump

to determine actual operating conditions during this transient period. Starting torques requiring running in reverse become less severe when the system head is partly or all friction head (See Abnormal operation and 4-quadrant characteristics in Sections 2.1.3, 11.3, and 12.14.)

Inertial Head If the system contains an appreciable amount of liquid, the inertia of the liquid mass could offer a significant resistance to any sudden change in velocity. Upon starting a primed pump and system without a valve, all the liquid in the system must accelerate from rest to a final condition of steady flow. Figure 24 illustrates a typical system head resistance that could be produced by a propeller pump pumping through a friction system when accelerated from rest to full speed. If the pump were accelerated very slowly, it would produce an all-friction resistance with zero liquid acceleration varying with flow approaching curve *OABCDEF*. Individual points on this curve represent system resistance at various constant pump speeds. If the pump were accelerated very rapidly, it would produce a system resistance approaching curve *OGHIJKL*. This is a shutoff condition that cannot be realized unless infinite driver torque is available. Individual points on this curve represent a system resistance at various constant speeds with no flow, which is the same as operating with a closed discharge valve. Curve *LF* represents the maximum total head the pump can produce as a result of both system friction and inertia at 100% speed. Head variation from *L* to *F* is a result of flow through the system, increasing at a decreasing rate of acceleration and increasing friction to the normal operating point *F*, where all the head is frictional. The actual total system-head resistance curve for any flow condition will be, therefore, the sum of the frictional resistance in feet (meters) for that steady-flow rate plus the inertial resistance, also expressed in feet (meters). The inertial resistance at any flow is dependent on the mass of liquid and the instantaneous rate of change of velocity at the flow condition. A typical total system resistance for a motor-driven propeller pump is shown as *OMNPQRF* in Figure 24. For a particular pump and system, different driver-speed-torque characteristics will result in a family of curves in the area *OLF*.

The added inertial system head produced momentarily when high-specific-speed pumps are started is important when considering the duration of high driver torques and currents, the pressure rise in the system, and the effect on the pump of operation at high heads and low flows. The approximate acceleration head h_a required to change the velocity of a mass of liquid at a uniform rate and cross section is

$$h_a = \frac{L\Delta V}{g\Delta t} \tag{10}$$

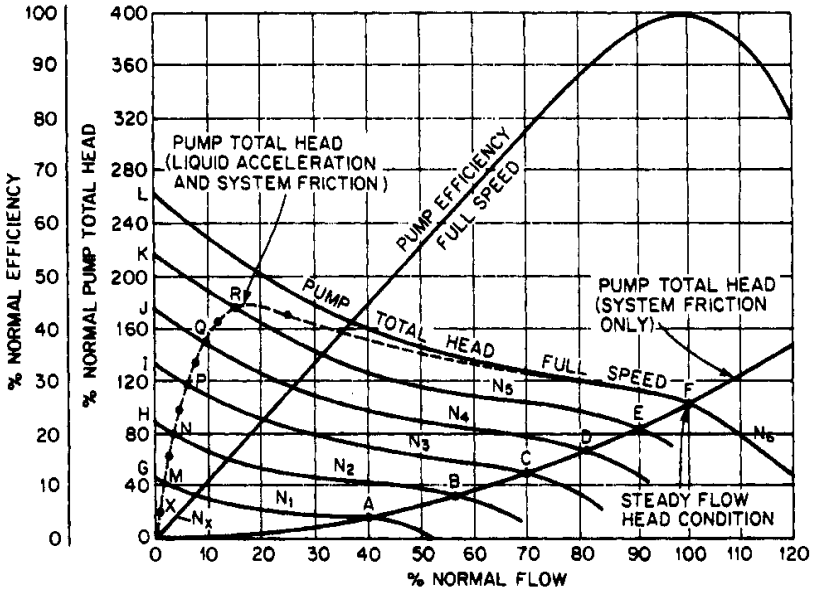


FIGURE 24 Transient system head as a result of liquid acceleration and system friction when a propeller pump is started with an open valve

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where L = length of constant-cross-section conduit, ft (m)
ΔV = velocity change, ft/s (m/s)
g = acceleration of gravity, 32.17 ft/s^2 (9.807 m/s^2)
Δt = time interval, s

To calculate the time to accelerate a centrifugal pump from rest or from some initial speed to a final speed, and to estimate the pump head variation during this interim, a trial-and-error solution may be used. Divide the speed change into several increments of equal no-flow heads, such as *OGHIKL* in Figure 24. For the first incremental speed change, point O to point N_1, estimate the total system head, point M, between G and A. Next estimate the total system head, point X, at the average speed for this first incremental speed change. These points are shown in Figure 24. Calculate the time in seconds for this incremental speed change to take place using the equation

in USCS units
$$\Delta t = \frac{\Sigma WK^2 \Delta N}{307(T_D - T_P)} \tag{11a}$$

where ΣWK^2 = total pump and motor rotor weight moment of inertia, W = weight (force), K = radius of gyration, lb · ft^2
ΔN = incremental speed change, rpm
T_D = motor torque at average speed, ft · lb
T_p = pump torque at average speed, ft · lb

in SI units
$$\Delta t = \frac{\Sigma MK^2 \Delta N}{9.55(T_D - T_P)} \tag{11b}$$

where ΣMK^2 = total pump and motor rotor mass moment of inertia, M = weight (mass), K = radius of gyration, kg · m^2
ΔN = incremental speed change, rpm
T_D = motor torque at average speed, N · m
T_P = pump torque at average speed, N · m

In SI units, when diameter of gyration D is used rather than radius of gyration K, and $MD^2 = 4MK^2$, then

$$\Delta t = \frac{\Sigma MD^2 \Delta N}{38.2(T_D - T_P)} \tag{11c}$$

Calculate the acceleration head required to change the flow in the system from point O to point M using Eq. 10 and time from Eq. 11. Add acceleration head to frictional head at the assumed average flow, and if this value is correct, it will fall on the average pump head-capacity curve, point X. Adjust points M and X until these assumed flows result in the total acceleration and frictional heads agreeing with flow X at the average speed. Repeat this procedure for other increments of speed change, adding incremental times to get total accelerating time to bring the pump up to its final speed. Plot system head versus average flow for each incremental speed change during this transient period, as shown in Figure 24.

Figures 25 and 26 illustrate how driver and pump torques can be determined from their respective speed-torque curves. Figure 25 is a family of curves that represents the torques required to produce flow against different heads without acceleration of the liquid or pump for the various speeds selected. Pump torque for any reduced speed can be calculated from the full-speed curve using the relation that torque varies as the second power and flow varies as the first power of the speed ratio. Point X is the torque at the average speed and the trial average flow during the first incremental speed change,

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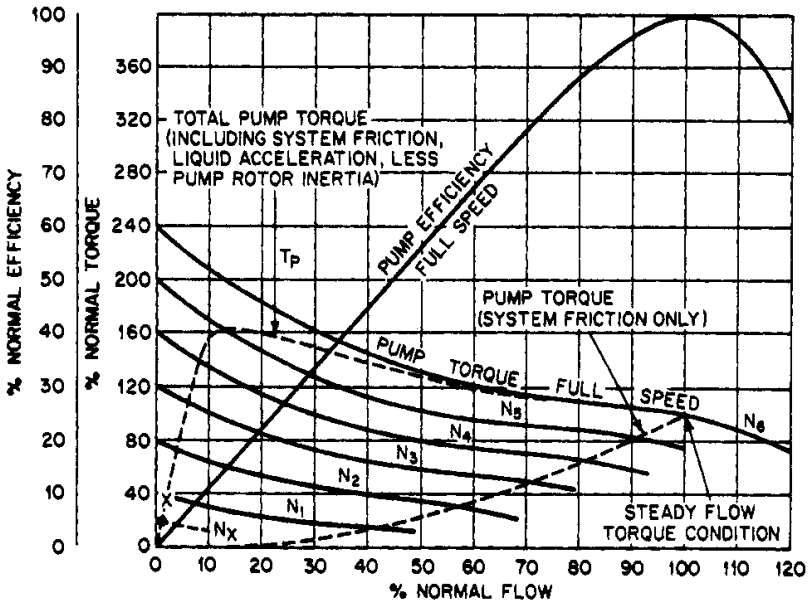


FIGURE 25 Transient total pump torque as a result of liquid acceleration and system friction when a propeller pump is started with an open valve. Pump head characteristics are shown in Figure 24.

which is adjusted for different assumed conditions. Figure 26 shows a typical squirrel-cage induction motor speed-torque curve, and, for the purpose of illustration, it has been selected to have the same torque rating as the pump requires at full speed (approximately 97% synchronous speed). Point X in this figure is the motor torque at the average speed during the first incremental speed change. In Figure 26, the developed torque curve for the different speeds shown in Figure 25 is redrawn as the total frictional and inertial pump torque T_p. For the conditions used in this example, and as shown in Figure 26, after approximately 88% synchronous speed, very little excess torque ($T_D - T_p$) is available for

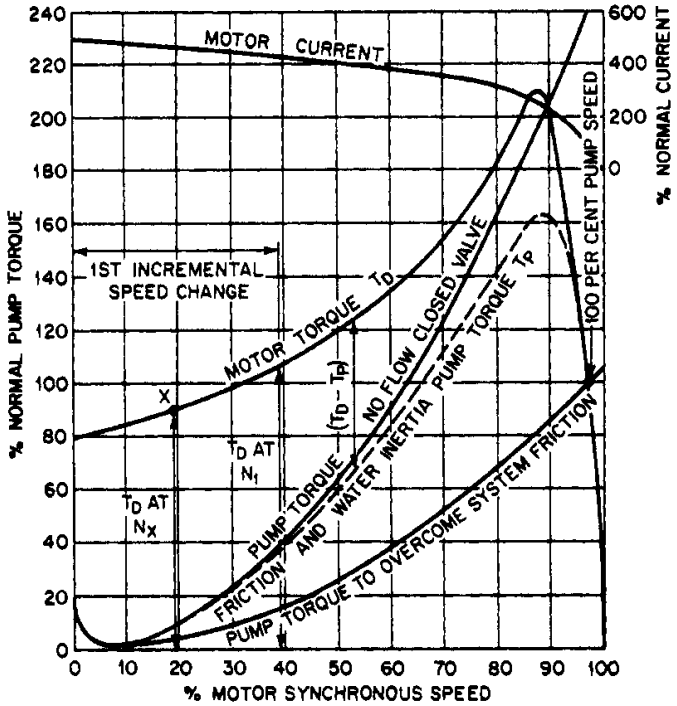


FIGURE 26 Propeller pump and motor speed-torque curves, showing effect of accelerating the liquid in the system during the starting period

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accelerating the pump and motor rotor inertia. However, as long as an induction motor has adequate torque to drive the pump at the normal condition, full speed will be reached providing the time-current demand can be tolerated. A synchronous motor, on the other hand, may not have sufficient torque to pull into step. Liquid acceleration decreases as motor torque decreases, adjusting to the available excess motor torque. The pump speed changes very slowly during the final period. Figure 26 also shows the motor current for the different speeds and torques.

A motor having a higher than normal pump torque rating or a motor having high starting torque characteristics will reduce the starting time, but higher heads will be produced during the acceleration period. Note, however, that heads produced cannot exceed shutoff at any speed. The total torque input to the pump shaft is equal to the sum of the torques required to overcome system friction, liquid inertia, and pump rotor inertia during acceleration. Torque at the pump shaft will therefore follow the motor speed-torque curve less the torque required to overcome the motor's rotor inertia. To reduce the starting inertial pump head to an acceptable amount, if desired, other alternative starting schemes can be used. A short bypass line from the pump discharge back to the suction can be provided to divert flow from the main system. The bypass valve is closed slowly after the motor reaches full speed. A variable-speed or a two-speed motor will reduce the inertial head by controlling motor torque and speed, thereby increasing the accelerating time.

This procedure for developing the actual system head and pump torque, including liquid inertia, becomes more complex if the pump must employ a discharge valve. To avoid high pump starting heads and torques, the discharge valve must be partially open on starting and then opened a sufficient amount before full speed is reached. The valve resistance must be added to the system friction and inertia curves if an exact solution is required.

Siphon Head Between any two points having the same elevation in a pumping system, no head is lost because of piping elevation changes because the net change in elevation is zero. If the net change in elevation between two points is not zero, additional pump head is required if there is an increase in elevation and less head is required if there is a decrease in elevation.

When piping is laid over and under obstacles with no net change in elevation, no pumping head is required to sustain flow other than that needed to overcome frictional and minor losses. As the piping rises, the liquid pressure head is transformed to elevation head, and the reverse takes place as the piping falls. A pipe or other closed conduit that rises and falls is called a *siphon,* and one that falls and rises is called an *inverted siphon.* The siphon principle is valid provided the conduit flows full and free of liquid vapor and air so the densities of the liquid columns are alike. It is this requirement that determines the limiting height of a siphon for complete recovery because the liquid can vaporize under certain conditions.

Pressure in a siphon is minimum at the summit, or just downstream from it, and Bernoulli's equation can be used to determine if the liquid pressure is above or below vapor pressure. Referring to Figure 27, observe the following. The absolute pressure head H_s in feet (meters) at the top of the siphon is

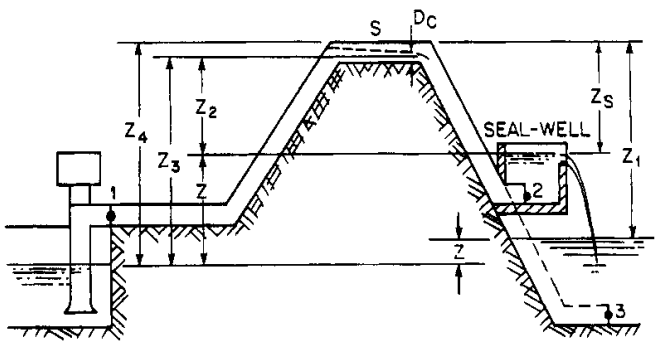


FIGURE 27 Pumping system using a siphon for head recovery

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$$H_S = H_B - Z_S + h_{f(S-2)} - \frac{V_S^2}{2g} \tag{12}$$

where H_B = barometric pressure head of liquid pumped, ft (m)

Z_S = siphon height to top of conduit (= Z_1 if no seal well is used), ft (m)

$h_{f(s-2)}$ = frictional and minor losses from S to 2 (or 3 if no seal well is used), including exit velocity head loss at 2 (or 3), ft (m)

$V^2_S/2g$ = velocity head at summit, ft (m)

The absolute pressure head at the summit can also be calculated using conditions in the up leg by adding the barometric pressure head to the pump head (TH) and deducting the distance from suction level to the top of the conduit (Z_4), the frictional loss in the up leg ($h_{f(1-S)}$), and the velocity head at the summit. If the suction level is higher than the discharge level and flow is by gravity, the absolute pressure head at the summit is found as above and $TH = 0$.

Whenever Z_1 in Figure 27 is so high that it exceeds the maximum siphon capability, a seal well is necessary to increase the pressure at the top of the siphon above vapor pressure. Note $Z_1 - Z_S$ represents an unrecoverable head and increases the pumping head. Water has a vapor pressure of 0.77 ft (23.5 cm) at 68°F (20°C) and theoretically a 33.23-ft-high (10.13-cm) siphon is possible with a 34-ft (10.36-m) water barometer. In practice, higher water temperatures and lower barometric pressures limit the height of siphons used in condenser cooling water systems to 26 to 28 ft (8 to 8.5 m). The siphon height can be found by using Eq. 12 and letting H_S equal the vapor pressure in feet (meters).

In addition to recovering head in systems such as condenser cooling water, thermal dilution, and levees, siphons are also used to prevent reverse flow after pumping is stopped by use of an automatic vacuum breaker located in the summit. Often siphons are used solely to eliminate the need for valves or flap gates.

In open-ended pumping systems, siphons can be primed by external means of air removal. Unless the siphon is primed initially upon starting, a pump must fill the system and provide a minimum flow to induce siphon action. During this filling period and until the siphon is primed, the siphon head curve must include this additional siphon filling head, which must be provided by the pump. Pumps in siphon systems are usually low-head, and they may not be capable of filling the system to the top of the siphon or of filling it with adequate flow. Low-head pumps are high-specific-speed and require more power at reduced flows than during normal pumping. Figure 28 illustrates the performance of a typical propeller pump when priming a siphon system and during normal operation.

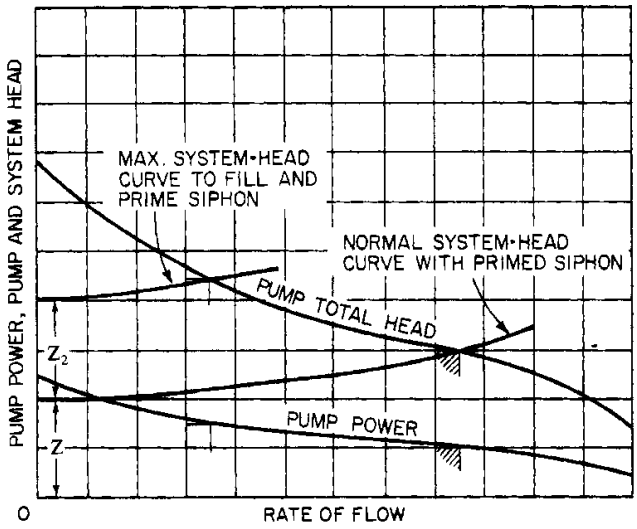


FIGURE 28 Transient system total head priming a siphon

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When a pump and driver are to be selected to prime a siphon system, it is necessary to estimate the pump head and the power required to produce the minimum flow needed to start the siphon.

The minimum flow required increases with the length and the diameter and decreases with the slope of the down-leg pipe.[2] Prior to the removal of all the air in the system, the pump is required to provide head to raise the liquid up to and over the siphon crest. Head above the crest is required to produce a minimum flow similar to flow over a broad-crested weir. This weir head may be an appreciable part of the total pump head if the pump is low-head and large-capacity. A conservative estimate of the pump head would include a full conduit above the siphon crest. In reality, the down leg must flow partially empty before it can flow full, and it is accurate enough to estimate that the depth of liquid above the siphon crest is at critical depth for the cross section. Table 1 can be used to estimate critical depth in circular pipes, and Figure 29 can be used to calculate the cross-sectional area of the filled pipe to determine the velocity at the siphon crest.

Until all the air is removed and all the piping becomes filled, the down leg is not part of the pumping system, and its frictional and minor losses are not to be added to the maximum system-head curve to fill and prime the siphon shown in Figure 28. The total head TH in feet (meters) to be produced by the pump in Figure 27 until the siphon is primed is

$$TH = Z_3 + h_{f(1-S)} + \frac{V_C^2}{2g} \tag{13}$$

where Z_3 = distance between suction level and centerline liquid at siphon crest, ft (m)

$h_{f(1-s)}$ = frictional and minor losses from 1 to S, ft (m)

$V^2_C/2g$ = velocity head at crest using actual liquid depth, approx. critical depth, ft (m)

Use of Eq. 13 permits plotting the maximum system-head curve to fill and prime the siphon for different flow rates. The pump priming flow is the intersection of the pump total head curve and this system-head curve. The pump selected must have a driver with power as shown in Figure 28 to prime the system during this transient condition.

For the pumping system shown in Figure 27, after the system is primed, the pump total head reduces to

$$TH = Z + h_{f(1-2)} \tag{14}$$

TABLE 1 Values for determining pipe-diameter ratio versus $(ft^3/s)/d^{5/2}$ in circular pipes

$\frac{D_{crit}}{d}$	0.00	0.01	0.02	0.03	0.04	0.05	0.06	0.07	0.08	0.09
0.0	...	0.0006	0.0025	0.0055	0.0098	0.0153	0.0220	0.0298	0.0389	0.0491
0.1	0.0605	0.0731	0.0868	0.1016	0.1176	0.1347	0.1530	0.1724	0.1928	0.2144
0.2	0.2371	0.2609	0.2857	0.3116	0.3386	0.3666	0.3957	0.4259	0.4571	0.4893
0.3	0.523	0.557	0.592	0.628	0.666	0.704	0.743	0.784	0.825	0.867
0.4	0.910	0.955	1.000	1.046	1.093	1.141	1.190	1.240	1.291	1.343
0.5	1.396	1.449	1.504	1.560	1.616	1.674	1.733	1.792	1.853	1.915
0.6	1.977	2.041	2.106	2.172	2.239	2.307	2.376	2.446	2.518	2.591
0.7	2.666	2.741	2.819	2.898	2.978	3.061	3.145	3.231	3.320	3.411
0.8	3.505	3.602	3.702	3.806	3.914	4.023	4.147	4.272	4.406	4.549
0.9	4.70	4.87	5.06	5.27	5.52	5.81	6.18	6.67	7.41	8.83

All tabulated values are in units of $(ft^3/s)/d^{5/2}$; d = diameter, ft; D_{crit} = critical depth, ft. For SI units, multiply $(m^3/h)/m^{5/2}$ by 5.03×10^{-4} to obtain $(ft^3/s)/ft^{5/2}$. Examples 2, 3, and 4 illustrate the use of this table.

Source: Reference 13.

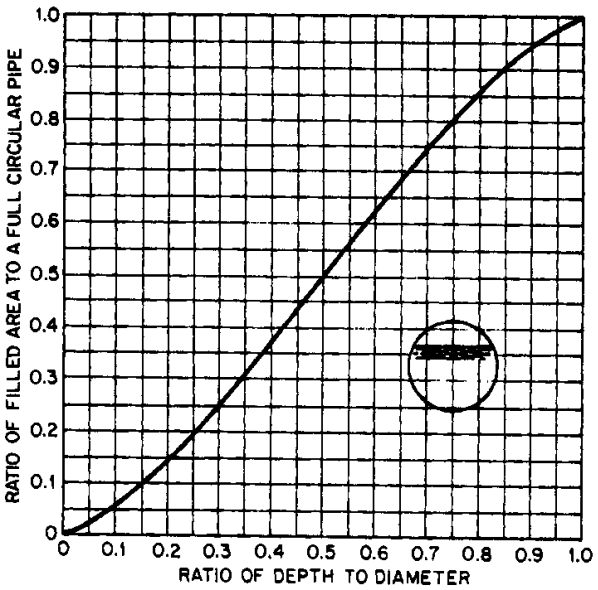


FIGURE 29 Area versus depth for a circular pipe

In SI units: $R = \frac{\rho VD}{\mu}$ (ρ in kg/m^3, V in m/s, D in m, μ in N·s/m^2)

1 meter = 3.28 feet

1 VD (V in m/s, D in m) = 129.2 VD'' (V in ft/s, D'' in inches)

where Z = distance between suction and seal well levels (or discharge pool level if no seal well is used), ft (m)

$h_{f(1-2)}$ = frictional and minor losses from 1 to 2 (or 3 if no seal well is used), including exit velocity head loss at 2 (or 3), ft (m)

Use of Eq. 14 permits plotting of the normal system-head curve with a primed siphon, shown in Figure 28, for different flow rates. The normal pump flow is the intersection of the pump total head curve and the normal system-head curve.

If the pump cannot provide sufficient flow to prime the siphon, or if the driver does not have adequate power, the system must be primed externally by a vacuum or jet pump. An auxiliary priming pump can also be used to continuously vent the system because it is necessary that this be done to maintain full siphon recovery. In some systems, the water pumped is saturated with air, and as the liquid flows through the system, the pressure is reduced (and in cooling systems the temperature is increased). Both these conditions cause the release of some of the entrained air. Air will accumulate at the top of the siphon and in the upper parts of the down leg. The siphon works on the principle that an increase in elevation in the up leg produces a decrease in pressure and an equal decrease in elevation in the down leg, resulting in recovery of this pressure. This cannot occur if the density of the liquid in the down leg is decreased as a result of the formation of air pockets. These air pockets also restrict the flow area. A release of entrained air and air leakage into the system through pipe joints and fittings will result in a centrifugal pump's delivering less than design flow, as the head will be higher than estimated. Also, high-specific-speed pumps with rising power curves toward shutoff can become overloaded. In order to maintain full head recovery, it is necessary to continuously vent the siphon at the top and at several points along the down leg, especially at the beginning of a change in slope.[3] These venting points can be manifolded together and connected to a single downward venting system.

Some piping systems may contain several up and down legs; that is, several siphons in series. Each down leg, as in a single siphon, is vulnerable to air or vapor binding. The likelihood of flow reduction, and conceivably in some cases complete flow shutoff, is increased in a multiple siphon system.[3] As shown in Figure 30, system static head is increased if

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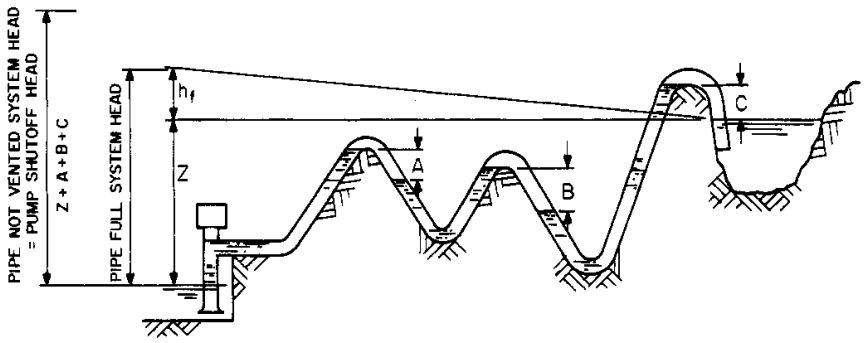


FIGURE 30 Multiple siphon system (Z = normal system static head when pipe is flowing full). Pump flow is stopped when normal static head plus sum of air pocket heights equals pump shutoff head.

proper venting is not provided at the top of each siphon. Normally the static head is the difference between outlet and inlet elevations. If air pockets exist, head cannot be recovered and the normal static head is increased by the sum of the heights of all the intermediate liquidless pockets. Flow will stop when the total static head equals the pump shutoff head.

The following examples illustrate the use of Eqs. 3, 9, 12, 13, and 14, Table 1, and Figure 29.

EXAMPLE 2 A pump is required to produce a flow of 70,000 gpm (15,900 m³/h) through the system shown in Figure 27. Calculate the system total head from point 1 to point 3 (no seal well) under the following conditions:

Specific gravity = 0.998 for 80°F (26.7°C) water

Barometric pressure = 29 in (73.7 cm) mercury abs (sp. gr. 13.6)

Suction and discharge water levels are equal, $Z = 0$

Z_1 = 40 ft (12.2 m)

$Hf_{(1-s)}$ = 3 ft (0.91 m) up-leg frictional head

$h_{f(S-3)}$ = 3.3 ft (1 m) down-leg frictional head, including exit loss

Pipe diameter = 48 in (121.9 cm) ID

Water vapor pressure = 0.507 lb/in² (3.5 kPa) abs at 80°F (26.7°C)

The maximum siphon height may be found from Eq. 12:

$$Z_s = H_B + h_{f(S-3)} - H_S + \frac{V_S^2}{2g}$$

From Eq. 3

H_B = barometric pressure head in feet (meters) of liquid pumped

In USCS units

$$\frac{\text{sp. gr.}_1}{\text{sp. gr.}_2} h_1 = \frac{13.6}{0.998} \times \frac{29}{12} = 32.9 \text{ ft abs}$$

$$H_S = \frac{p}{\gamma} = \frac{144 \times 0.507}{62.19} = 1.17 \text{ ft abs}$$

In SI units

$$\frac{\text{sp. gr.}_1}{\text{sp. gr.}_2} h_1 = \frac{13.6}{0.998} \times \frac{73.7}{100} = 10 \text{ m abs}$$

$$H_S = \frac{p}{\gamma} = \frac{3.5 \times 1000}{9769} = 0.358 \text{ m abs}$$

From Eqs. 9 and 12,

in USCS units $$V_S = \frac{\text{gpm}}{(\text{pipe ID in inches})^2} \times 0.408 = \frac{70{,}000}{48^2} \times 0.408 = 12.4 \text{ ft/s}$$

$$Z_S = 32.9 + 3.3 - 1.17 - \frac{12.4^2}{2 \times 32.17} = 32.63 \text{ ft}$$

in SI units $$V_S = \frac{\text{m}^3/\text{h}}{(\text{pipe ID in cm})^2} \times 3.54 = \frac{15{,}900}{121.9^2} \times 3.54 = 3.79 \text{ m/s}$$

$$Z_S = 10 + 1 - 0.358 - \frac{3.79^2}{2 \times 9.807} = 9.1 \text{ m}$$

Because the maximum height is exceeded ($Z_1 > Z_s$), siphon recovery is not possible. The system total head is therefore found from Eq. 13:

$$TH = Z_3 + h_{f(1-S)} + \frac{V_c^2}{2g}$$

The critical depth D_{crit} is found using Table 1:

In USCS units $$\text{ft}^3/\text{s} = \frac{70{,}000}{7.481 \times 60} = 156$$

$$\frac{\text{ft}^3/\text{s}}{d^{5/2}} = \frac{156}{4^{5/2}} = 4.88$$

$$\frac{D_{\text{crit}}}{d} = 0.901$$

$$D_{\text{crit}} = 0.901 \times 4 = 3.6 \text{ ft}$$

In SI units $$\frac{\text{m}^3/\text{h}}{d^{5/2}} = \frac{15{,}900}{1.219^{5/2}} = 9695$$

Convert to USCS units (see footnote to Table 1):

$$\frac{\text{ft}^3/\text{s}}{\text{ft}^{5/2}} = 9695 \times 5.03 \times 10^{-4} = 4.88$$

$$\frac{D_{\text{crit}}}{d} = 0.901$$

$$D_{\text{crit}} = 0.901 \times 1.219 = 1.1 \text{ m}$$

To calculate the water velocity at the siphon crest, determine the area of the filled pipe. From Figure 29, ratio of filled area to area of a full pipe is 0.95 for a depth-to-diameter ratio of 0.901:

In USCS units $$V_c = \frac{\text{gpm}}{0.95(\text{ID in inches})^2} \times 0.408 = \frac{70{,}000}{0.95 \times 48^2} \times 0.408 = 13.0 \text{ ft/s}$$

$$Z_3 = 40 - 4 + \frac{3.6}{2} = 37.8 \text{ ft}$$

From Eq. 13 $$TH = 37.8 + 3 + \frac{13.0^2}{2 \times 32.17} = 43.43 \text{ ft}$$

In SI units $$V_c = \frac{\text{m}^3/\text{h}}{0.95(ID \text{ in cm})^2} \times 3.54 = \frac{15{,}900}{0.95 \times 121.9^2} \times 3.54 = 3.99 \text{ m/s}$$

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$$Z_3 = 12.2 - 1.219 + \frac{1.1}{2} = 11.53 \text{ m}$$

From Eq. 13 $$TH = 11.53 + 0.91 + \frac{3.99^2}{2 \times 9.807} = 13.25 \text{ m}$$

EXAMPLE 3 Calculate the minimum total system head using conditions in Example 2 and a seal well, as shown in Figure 27. Use 2.8 ft (0.853 m) for the frictional head loss $h_{f(s-2)}$.

The maximum siphon height Z_s in Example 2 was found to be 32.63 ft (9.95 m). Therefore from Eq. 14 the total system head after priming is

$$TH = Z + h_{f(1-2)}$$

In USCS units $$h_{f(1-2)} = h_{f(1-S)} + h_{f(S-2)} = 3 + 2.8 = 5.8 \text{ ft}$$

$$Z = Z_1 - Z_S = 40 - 32.63 = 7.37 \text{ ft}$$

In SI units $$h_{f(1-2)} = h_{f(1-S)} + h_{f(S-2)} = 0.91 + 0.853 = 1.763 \text{ ft}$$

$$Z = Z_1 - Z_S = 12.2 - 9.95 = 2.25 \text{ m}$$

Note that the seal well elevation is above discharge level. Therefore

in USCS units $$TH = 7.37 + 5.8 = 13.17 \text{ ft}$$

in SI units $$TH = 2.25 + 1.763 = 4.01 \text{ m}$$

EXAMPLE 4 The dimensions of the down leg in Example 3 require a minimum velocity of 5 ft/s (1.52 m/s) flowing full to purge air from the system and start the siphon. Calculate the system head the pump must overcome to prime the siphon.

In USCS units $$\text{gpm} = \frac{V(\text{pipe } ID \text{ in inches})^2}{0.408} = \frac{5 \times 48^2}{0.408} = 28{,}200$$

$$\text{ft}^3/\text{s} = 28{,}200 \div 449 = 62.8$$

In SI units $$\text{m}^3/\text{h} = \frac{V(\text{pipe } ID \text{ in cm})^2}{3.54} = \frac{1.52 \times 121.9^2}{3.54} = 6400$$

The critical depth is found from Table 1:

In USCS units $$\frac{\text{ft}^3/\text{s}}{d^{5/2}} = \frac{62.8}{4^{5/2}} = 1.97$$

$$\frac{D_{\text{crit}}}{d} = 0.6$$

$$D_{\text{crit}} = 0.6 \times 4 = 2.4 \text{ ft}$$

In SI units $$\frac{\text{m}^3/\text{h}}{d^{5/2}} = \frac{6400}{1.219^{5/2}} = 3902$$

Convert to USCS units (see footnote to Table 1):

$$\frac{\text{ft}^3/\text{s}}{d^{5/2}} = 3902 \times 5.03 \times 10^{-4} = 1.97$$

$$\frac{D_{\text{crit}}}{d} = 0.6$$

$$D_{\text{crit}} = 0.6 \times 1.219 = 0.73 \text{ m}$$

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From Figure 29, the ratio of the filled area to the area of a full pipe is 0.625 for a depth-to-diameter ratio of 0.60:

In USCS units $$V_c = \frac{5}{0.625} = 8.0 \text{ ft/s}$$

In SI units $$V_c = \frac{1.52}{0.625} = 2.4 \text{ m/s}$$

From Eq. 13 $$TH = Z_3 + h_{f(1-S)} + \frac{V_c^2}{2g}$$

In USCS units $$Z_2 = Z_S - 4 + \frac{D_{crit}}{2} = 32.63 - 4 + \frac{2.4}{2} = 29.83 \text{ ft}$$

$$Z = 7.37 \text{ ft (from Example 3)}$$

$$h_{f(1-S)} \text{ at } 28{,}200 \text{ gpm} = \left(\frac{28{,}200}{70{,}000}\right)^2 \times 3 = 0.49 \text{ ft}$$

$$TH = (7.37 + 29.83) + 0.49 + \frac{8^2}{2 \times 32.17} = 38.68 \text{ ft}$$

In SI units $$Z_2 = Z_S - 1.219 + \frac{D_{crit}}{2} = 9.95 - 1.219 + \frac{0.73}{2} = 9.1 \text{ m}$$

$$Z = 2.25 \text{ m (from Example 3)}$$

$$h_{f(1-S)} \text{ at } 6400 \text{ m}^3/\text{h} = \left(\frac{6400}{15{,}900}\right)^2 \times 0.91 = 0.15 \text{ m}$$

$$TH = (2.25 + 9.1) + 0.15 + \frac{2.4^2}{2 \times 9.807} = 11.79 \text{ m}$$

If the system is not externally primed, the centrifugal pump selected must be able to deliver at least 28,000 gpm (6400 m³/h) at 38.68 ft (11.79 m) total head and must be provided with a driver having adequate power for this condition. After the system is primed, the pump must be capable of delivering at least 70,000 gpm (15,900 m³/h) at 13.17 ft (4.01 m) (see Figure 28).

HEAD LOSSES IN SYSTEM COMPONENTS

Pressure Pipes Resistance to flow through a pipe is caused by viscous shear stresses in the liquid and by turbulence at the pipe walls. *Laminar* flow occurs in a pipe when the average velocity is relatively low and the energy head is lost mainly as a result of viscosity. In laminar flow, liquid particles have no motion next to the pipe walls and flow occurs as a result of the movement of particles in parallel lines with velocity increasing toward the center. The movement of concentric cylinders past each other causes viscous shear stresses, more commonly called *friction*. As flow increases, the flow pattern changes, the average velocity becomes more uniform, and there is less viscous shear. As the laminar film decreases in thickness at the pipe walls and as the flow increases, the pipe roughness becomes important because it causes turbulence. *Turbulent* flow occurs when average pipe velocity is relatively high and energy head is lost predominantly because of turbulence caused by the wall roughness. The average velocity at which the flow changes from laminar to turbulent is not definite, and there is a critical zone in which either laminar or turbulent flow can occur.

Viscosity can be visualized as follows. If the space between two planar surfaces is filled with a liquid, a force will be required to move one surface at a constant velocity relative to the other. The velocity of the liquid will vary linearly between the surfaces. The ratio of the force per unit area, called *shear stress*, to the velocity per unit distance between surfaces, called *shear* or *deformation rate*, is a measure of a liquid's *dynamic* or *absolute viscosity*.

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Liquids such as water and mineral oil, which exhibit shear stresses proportional to shear rates, have a constant viscosity for a particular temperature and pressure and are called *Newtonian* or *true liquids.* In the normal pumping range, however, the viscosity of true liquids may be considered independent of pressure. For these liquids, the viscosity remains constant because the rate of deformation is directly proportional to the shearing stress. The viscosity and resistance to flow, however, increase with decreasing temperature.

Liquids such as molasses, grease, starch, paint, asphalt, and tar behave differently from Newtonian liquids. The viscosity of the former does not remain constant and their shear, or deformation, rate increases more than the stress increases. These liquids, called *thixotropic,* exhibit lower viscosity as they are agitated at a constant temperature.

Still other liquids, such as mineral slurries, show an increase in viscosity as the shear rate is increased and are called *dilatant.*

In USCS units, dynamic (absolute) viscosity is measured in pound-seconds per square foot or slugs per foot-second. In SI measure, the units are newton-seconds per square meter or pascal-seconds. Usually dynamic viscosity is measured in *poises* (1 P = 0.1 Pa · s) or in *centipoises* (1 cP = $\frac{1}{100}$ P):

$$1 \text{ lb} \cdot \text{s/ft}^2 = 47.8801 \text{ Pa} \cdot \text{s} = 47{,}880.1 \text{ cP}$$

The viscous property of a liquid is also sometimes expressed as *kinematic viscosity.* This is the dynamic viscosity divided by the mass density (specific weight/g). In USCS units, kinematic viscosity is measured in square feet per second. In SI measure, the units are square meters per second. Usually kinematic viscosity is measured in *stokes* (1 St = 0.0001 m^2/s) or in *centistokes* (1 cSt = $\frac{1}{100}$ St):

$$1 \text{ ft}^2/\text{s} = 0.0929034 \text{ m}^2/\text{s} = 92{,}903.4 \text{ cSt}$$

A common unit of kinematic viscosity in the United States is Saybolt seconds universal (SSU) for liquids of medium viscosity and Saybolt seconds Furol (SSF) for liquids of high viscosity. Viscosities measured in these units are determined by using an instrument that measures the length of time needed to discharge a standard volume of the sample. Water at 60°F (15.6°C) has a kinematic viscosity of approximately 31 SSU (1.0 cSt). For values of 70 cSt and above,

$$\text{cSt} = 0.216 \text{ SSU}$$
$$\text{SSU} = 10 \text{ SSF}$$

The dimensionless Reynolds number Re is used to describe the type of flow in a pipe flowing full and can be expressed as follows:

$$Re = \frac{VD}{\upsilon} = \frac{\rho VD}{\mu} \tag{15}$$

where V = average pipe velocity, ft/s (m/s)
D = inside pipe diameter, ft (m)
υ = liquid kinematic viscosity, ft^2/s (m^2/s)
ρ = liquid density, slugs/ft^3 (kg/m^3)
μ = liquid dynamic (or absolute) viscosity slug/ft · s (N · s/m^2)

Note: The dimensionless Reynolds number is the same in both USCS and SI units.

When the Reynolds number is 2000 or less, the flow is generally laminar, and when it is greater than 4000, the flow is generally turbulent. The Reynolds number for the flow of water in pipes is usually well above 4000, and therefore the flow is almost always turbulent.

The Darcy-Weisbach formula is the one most often used to calculate pipe friction. This formula recognizes that friction increases with pipe wall roughness, with wetted surface area, with velocity to a power, and with viscosity and decreases with pipe diameter to a power and with density. Specifically, the frictional head loss h_f in feet (meters) is

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$$h_f = f\frac{L}{D}\frac{V^2}{2g} \tag{16}$$

where f = friction factor
L = pipe length, ft (m)
D = inside pipe diameter, ft (m)
V = average pipe velocity, ft/s (m/s)
g = acceleration of gravity, 32.17 ft/s^2 (9.807 m/s^2)

For laminar flow, the friction factor f is equal to $64/Re$ and is independent of pipe wall roughness. For turbulent flow, f for all incompressible fluids can be determined from the well-known Moody diagram, shown in Figure 31. To determine f, it is required that the Reynolds number and the relative pipe roughness be known. Values of relative roughness (ϵ/D), where ϵ is a measure of pipe wall roughness height in feet (meters), can be obtained from Figure 32 for different pipe diameters and materials. Figure 32 also gives values for f for the flow of 60°F (15.6°C) water in rough pipes with complete turbulence. Values of kinematic viscosity and Reynolds numbers for a number of different liquids at various temperatures are given in Figure 33. The Reynolds numbers of 60°F (15.6°C) water for various velocities and pipe diameters may be found by using the VD'' scale in Figure 31.

There are many empirical formulas for calculating pipe friction for water flowing under turbulent conditions. The most widely used is the Hazen-Williams formula:

In USCS units $$V = 1.318Cr^{0.63}S^{0.54} \tag{17a}$$

In SI units $$V = 0.8492Cr^{0.63}S^{0.54} \tag{17b}$$

where V = average pipe velocity, ft/s (m/s)
C = friction factor for this formula, which depends on roughness only
r = hydraulic radius (liquid area divided by wetted perimeter) or $D/4$ for a full pipe, ft (m)
S = hydraulic gradient or frictional head loss per unit length of pipe, ft/ft (m/m)

The effect of age on a pipe should be taken into consideration when estimating the frictional loss. A lower C value should be used, depending on the expected life of the system. Table 2 gives recommended friction factors for new and old pipes. A value of C of 150 may be used for plastic pipe. Figure 34 is a nomogram that can be used in conjunction with Table 2 for a solution to the Hazen-Williams formula.

The frictional head loss in pressure pipes can be found by using either the Darcy-Weisbach formula (Eq. 16) or the Hazen-Williams formula (Eq. 17). Tables in the appendix give Darcy-Weisbach friction values for Schedule 40 new steel pipe carrying water. Tables are also provided for losses in old cast iron piping based on the Hazen-Williams formula with $C = 100$. In addition, values of C for various pipe materials, conditions, and years of service can also be found in the appendix.

The following examples illustrate how Figures 31, 32, and 33 and Table 2 may be used.

EXAMPLE 5 Calculate the Reynolds number for 175°F (79.4°C) kerosene flowing through 4-in (10.16-cm), Schedule 40, 3.426-in (8.70-cm) ID, seamless steel pipe at a velocity of 14.6 ft/s (4.45 m/s).

In USCS units $VD'' = 14.6 \times 3.426 = 50$ ft/s × in

In SI units $VD = 4.45 \times 0.087 = 0.387$ m/s × m $= 0.387 \times 129.2 = 50$ ft/s × in

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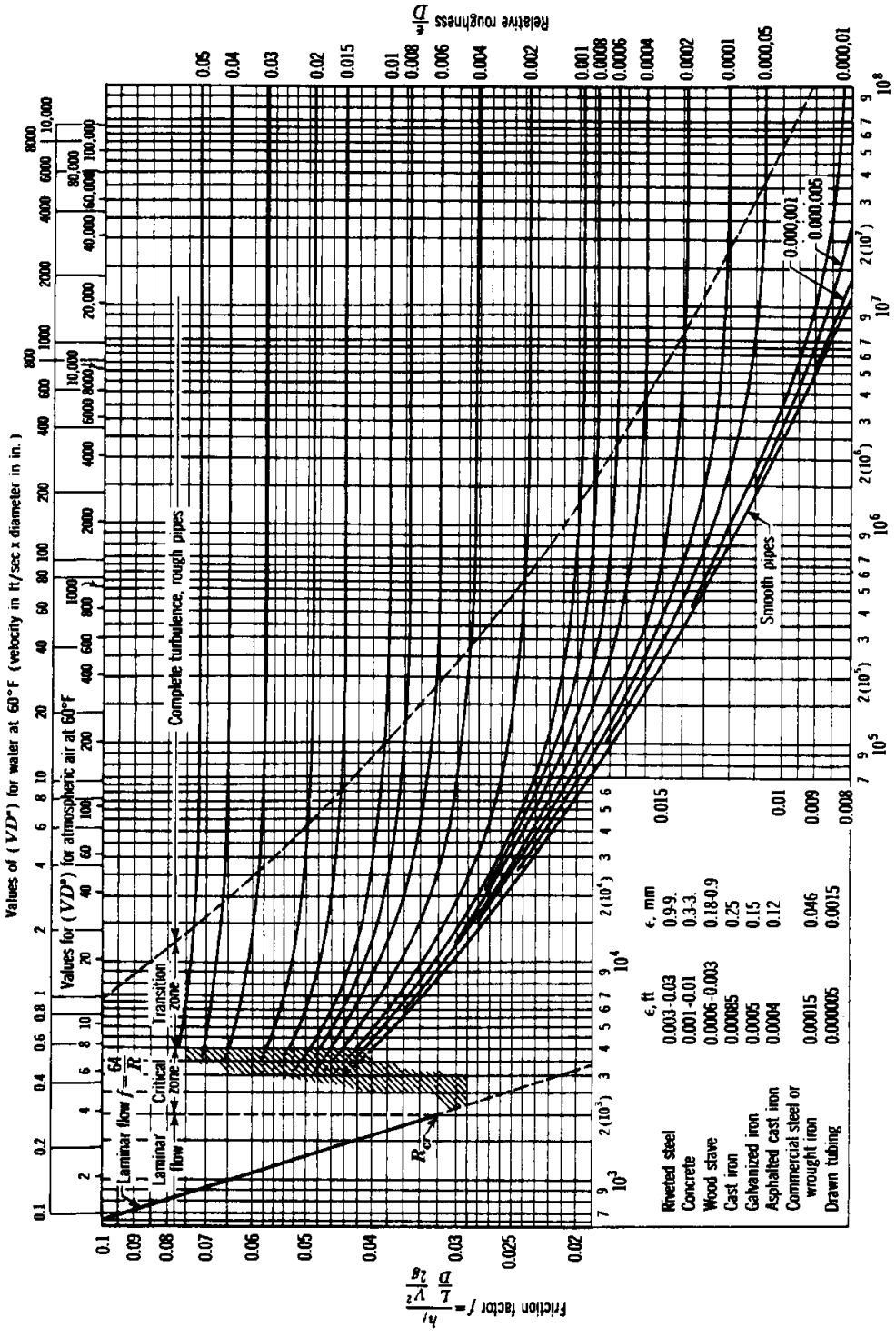


FIGURE 31 Moody diagram. (Reference 14) In SI units: $\mathbf{R} = \dfrac{\rho VD}{\mu}$ (ρ in kg /m^3, V in m/s, D in m, μ in N • s/m^2.) 1 meter = 3.28 ft; $1VD$ (V in m/s, D in m) = $129.2\,VD^*$ (V in ft/s, D^* in inches).

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TABLE 2 Values of friction factor C to be used with the Hazen-Williams formula in Figure 34

Type of pipe	Age	Size, in[a]	C
Cast iron	New	All sizes	130
	5 years old	12 and over	120
		8	119
		4	118
	10 years old	24 and over	113
		12	111
		4	107
	20 years old	24 and over	100
		12	96
		4	89
	30 years old	30 and over	
		16	87
		4	75
	40 years old	30 and over	83
		16	80
		4	64
	40 and over	77	
		24	74
		4	55
Welded steel	Any age, any size		Same as for cast iron pipe 5 years old
Riveted steel	Any age, any size		Same as cast iron pipe 10 years older
Wood-stave	Average value, regardless of age and size		120
Concrete or concrete-lined	Large sizes, good workmanship, steel forms		140
	Large sizes, good workmanship, wooden forms		120
	Centrifugally spun		135
Vitrified	In good condition		110

[a]In × 25.4 = mm

Source: Adapted From Reference 15.

Follow the tracer lines in Figure 33 and read directly:

$$Re = 3.5 \times 10^5$$

EXAMPLE 6 Calculate the frictional head loss for 100 ft (30.48 m) of 20-in (50.8-cm), Schedule 20, 19.350-in (49.15-cm) ID, seamless steel pipe for 109°F (42.8°C) water flowing at a rate of 11,500 gpm (2612 m³/h). Use the Darcy-Weisbach formula.

In USCS Units

$$V = \frac{\text{gpm}}{(\text{pipe } ID \text{ in inches})^2} \times 0.408 = \frac{11{,}500}{19.35^2} \times 0.408 = 12.53 \text{ ft/s}$$

$$VD'' = 12.53 \times 19.35 = 242 \text{ ft/s} \times \text{in}$$

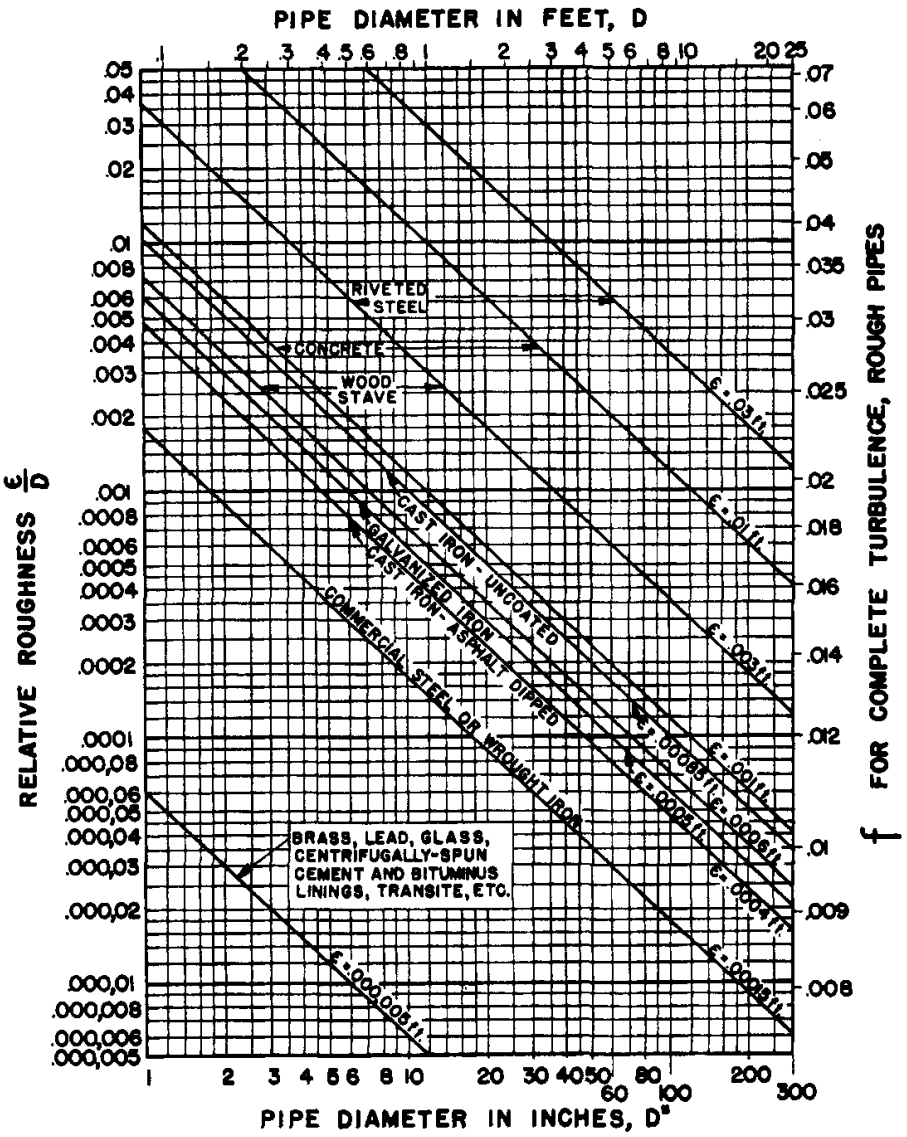


FIGURE 32 Relative roughness and friction factors for new, clean pipes for flow of 60°F (15.6°C) water (Hydraulic Institute Engineering Data Book, Reference 5) (1 meter 39.37 in = 3.28 ft)

In SI units

$$V = \frac{\text{m}^3/\text{h}}{(\text{pipe } ID \text{ in cm})} \times 3.54 = \frac{2612}{49.15^2} \times 3.54 = 3.83 \text{ m/s}$$

$$VD = 3.83 \times 0.4915 = 1.88 \text{ m/s} \times \text{m} = 1.88 \times 129.2 = 242 \text{ ft/s} \times \text{in}$$

From Figure 33 $Re = 3 \times 10^6$

From Figure 32 $\frac{\epsilon}{D} = 0.00009$

From Figure 31 $f = 0.012$

Using Eq. 16,

In USCS units $D = \frac{19.35}{12} = 1.61 \text{ ft}$

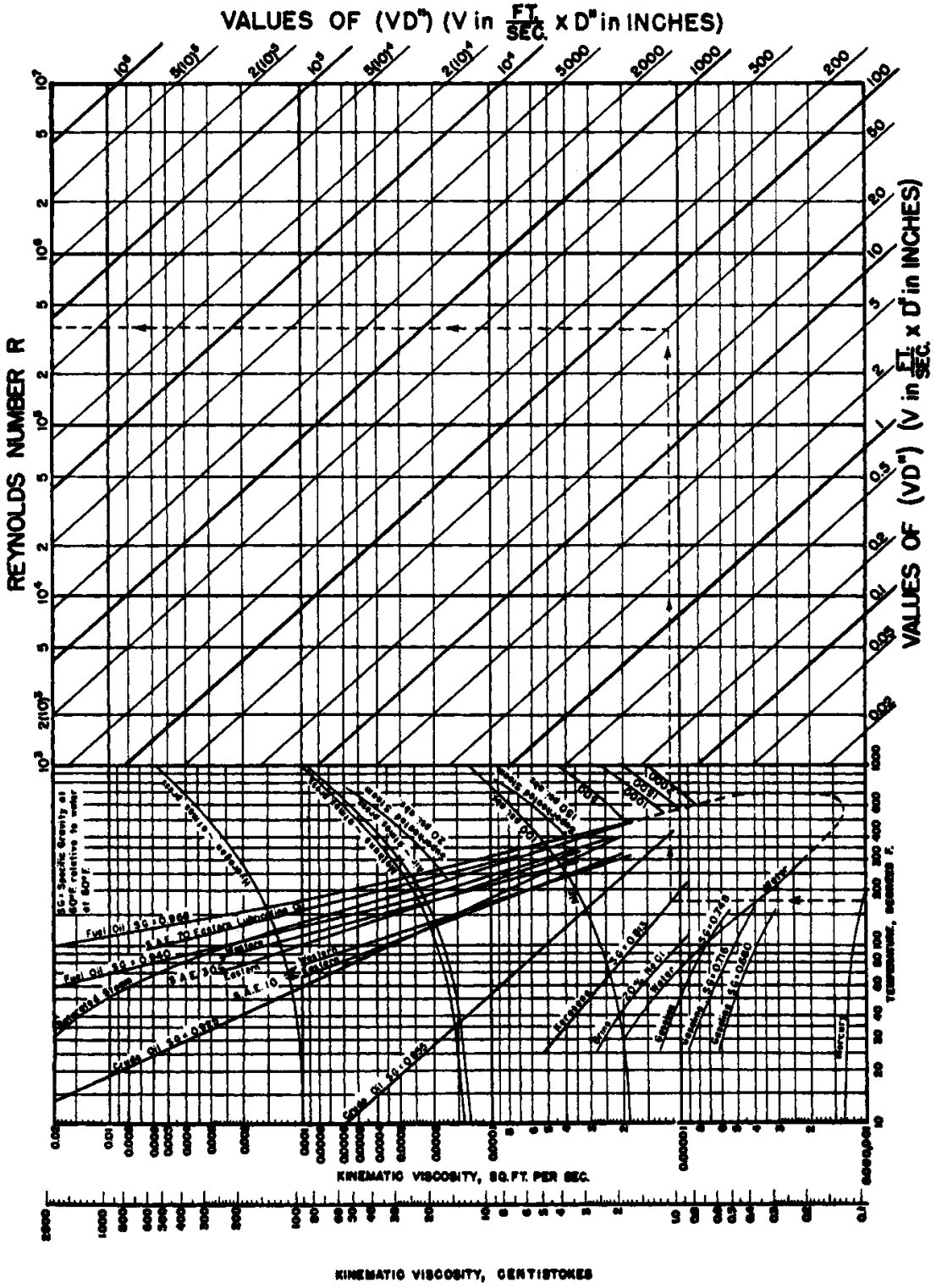


FIGURE 33 Kinematic viscosity and Reynolds number. (Hydraulic Institute Engineering Data Book, Reference 5) [1 ft²/s = 0.0929 m²/s; 1 cSt = 1.0 × 10^{-6} m²/s; 1*VD* (m/s × m) = 129.2 VD" (ft/s × in); °F = (°C + 18) 1.8]

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$$h_f = f\frac{L}{d}\frac{V^2}{2g} = 0.012\,\frac{100}{1.61} \times \frac{12.53^2}{2 \times 32.17} = 1.82 \text{ ft}$$

In SI units $$h_f = f\frac{L}{D}\frac{V^2}{2g} = 0.012\,\frac{30.48}{0.4915} \times \frac{3.83^2}{2 \times 9.807} = 0.556 \text{ m}$$

EXAMPLE 7 The flow in Example 6 is increased until complete turbulence results. Determine the friction factor f and flow.

From Figure 31, follow the relative roughness curve $\varepsilon/D = 0.00009$ to the beginning of the zone marked "complete turbulence, rough pipes" and read

$$f = 0.0119 \quad \text{at } Re = 2 \times 10^7$$

The problem may also be solved using Figure 32. Enter relative roughness $\varepsilon/D = 0.00009$ and read directly across to

$$f = 0.0119$$

An increase in Re from 3×10^6 to 2×10^7 would require an increase in flow to

in USCS units $$\frac{2 \times 10^7}{3 \times 10^6} \times 11{,}500 = 76{,}700 \text{ gpm}$$

in SI units $$\frac{2 \times 10^7}{3 \times 10^6} \times 2612 = 17{,}413 \text{ m}^3/\text{h}$$

EXAMPLE 8 The liquid in Example 6 is changed to water at 60°F (15.6°C). Determine Re, f, and the frictional head loss per 100 ft (100 m) of pipe.

$$VD'' = 242 \text{ ft/s} \times \text{in} \qquad \text{(as in Example 6)}$$

Because the liquid is 60°F (15.6°C) water, enter Figure 31 and read directly downward from VD'' to

$$Re = 1.8 \times 10^6$$

Where the line VD'' to Re crosses $\varepsilon/D = 0.00009$ in Figure 31, read

$$f = 0.013$$

Water at 60°F (15.6°C) is more viscous than 109°F (42.8°C) water, and this accounts for the fact that Re decreases and f increases. Using Eq. 16, it can be calculated that the frictional head loss increases to

in USCS units $$h_f = f\frac{L}{D}\frac{V^2}{2g} = 0.013\,\frac{100}{1.61} \times \frac{12.53^2}{2 \times 32.17} = 1.97 \text{ ft}$$

in SI units $$h_f = f\frac{L}{D}\frac{V^2}{2g} = 0.013\,\frac{100}{0.4915} \times \frac{3.83^2}{2 \times 9.807} = 1.97 \text{ m}$$

EXAMPLE 9 A 102-in (259-cm) ID welded steel pipe is to be used to convey water at a velocity of 11.9 ft/s (3.63 m/s). Calculate the expected loss of head due to friction per 1000 ft and per 1000 m of pipe after 20 years. Use the empirical Hazen-Williams formula.

From Table 2, $C = 100$.

In USCS units $$r = \frac{D}{4} = \frac{102}{(4 \times 12)} = 2.13 \text{ ft}$$

In SI units $$r = \frac{D}{4} = \frac{2.59}{4} = 0.648 \text{ m}$$

Substituting in Eq. 17,

in USCS units
$$S^{0.54} = \frac{V}{1.318Cr^{0.63}} = \frac{11.9}{1.318 \times 100 \times 2.13^{0.63}} = 0.0557$$
$$S = (0.0557)^{1/0.54} = 0.0048 \text{ ft/ft}$$
$$h_f = 1000 \times 0.0048 = 4.8 \text{ ft}$$

in SI units
$$S^{0.54} = \frac{V}{0.8492Cr^{0.63}} = \frac{3.63}{0.8492 \times 100 \times 0.648^{0.63}} = 0.0562$$
$$S = (0.0562)^{1/0.54} = 0.0048 \text{ m/m}$$
$$h_f = 1000 \times 0.0048 = 4.8 \text{ m}$$

The problem may also be solved by using Figure 34, following the trace lines:

$$h_f \approx 5 \text{ ft (m)}$$

Frictional Loss for Viscous Liquids Table 3 gives the frictional loss for viscous liquids flowing in new Schedule 40 steel pipe. Values of pressure loss are given for both laminar and turbulent flows.

For laminar flow, the pressure loss is directly proportional to the viscosity and the velocity of flow and inversely proportional to the pipe diameter to the fourth power. Therefore, for intermediate values of viscosity and flow, obtain the pressure loss by direct interpolation.

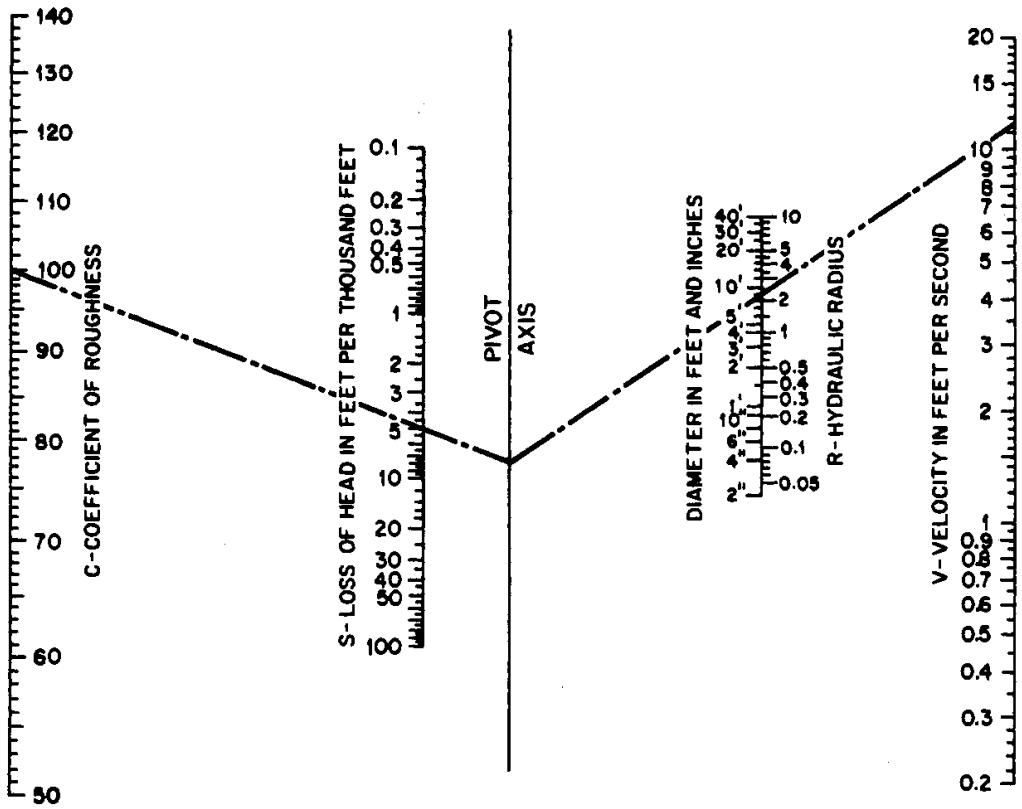


FIGURE 34 Nomogram for the solution of the the Hazen-Williams formula. Obtain values for C from Table 2 (Reference 15) (1 m/s = 3.28 ft/s; 1 m = 39.37 in)

For pipe sizes not shown, multiply the fourth power of the ratio of any tabulated diameter to the pipe diameter wanted by the tabulated loss shown. The flow rate and viscosity must be the same for both diameters.

For turbulent flow and for rates of flow and pipe sizes not tabulated, the following procedures may be followed. For the viscosity and pipe size required, an intermediate flow loss is found by selecting the pressure loss for the next lower flow and multiplying by the square of the ratio of actual to tabulated flow rates. For the viscosity and flow required, an intermediate pipe diameter flow loss is found by selecting the pressure loss for the next smaller diameter and multiplying by the fifth power of the ratio of tabulated to actual inside diameters.

The viscosity of various common liquids can be found in tables in the appendix.

Partially Full Pipes and Open Channels Another popular empirical equation applicable to the flow of water in pipes flowing full or partially full or in open channels is the Manning formula:

In USCS units
$$V = \frac{1.486}{n} r^{2/3} S^{1/2} \tag{18a}$$

In SI units
$$V = \frac{r^{2/3} S^{1/2}}{n} \tag{18b}$$

where V = average velocity, ft/s (m/s)
n = friction factor for this formula, which depends on roughness only
r = hydraulic radius (liquid area divided by wetted perimeter), ft (m)
S = hydraulic gradient or frictional head loss per unit length of conduit, ft/ft (m/m)

The Manning formula nomogram shown in Figure 35 can be used to determine the flow or frictional head loss in open or closed conduits. Note that the hydraulic gradient S in Figure 35 is plotted in feet per 100 ft of conduit length. Values of friction factor n are given in Table 4.

If the conduit is flowing partially full, computing the hydraulic radius is sometimes difficult. When the problem to be solved deals with a pipe that is not flowing full, Figure 36 may be used to obtain multipliers for correcting the flow and velocity of a full pipe to the values needed for the actual fill condition. If the flow in a partially full pipe is known and the frictional head loss is to be determined, Figure 36 is first used to correct the flow to what it would be if the pipe were full. Then Eq. 18 or Figure 35 is used to determine the frictional head loss (which is also the hydraulic gradient and the slope of the pipe). The problem is solved in reverse if the hydraulic gradient is known and the flow is to be determined.

For full or partially full flow in conduits that are not circular in cross section, an alternate solution to using Eq. 18 is to calculate an equivalent diameter equal to four times the hydraulic radius. If the conduit is extremely narrow and width is small relative to length (annular or elongated sections), the hydraulic radius is one-half the width of the section.[4] After the equivalent diameter has been determined, the problem may be solved by using the Darcy-Weisbach formula (Eq. 16).

The hydraulic gradient in a uniform open channel is synonymous with frictional head loss in a pressure pipe. The hydraulic gradient of an open channel or of a pipe flowing partially full is the slope of the free liquid surface. In the reach of the channel where the flow is uniform, the hydraulic gradient is parallel to the slope of the channel bottom. Figure 37 shows that, in a pressure pipe of uniform cross section, the slope of both the energy and hydraulic gradients is a measure of the frictional head loss per foot (meter) of pipe between points 1 and 2. Figure 38 illustrates the flow in an open channel of varying slope. Between points 1 and 2, the flow is uniform and the liquid surface (hydraulic gradient) and channel bottom are both parallel and their slope is the frictional head loss per foot (meter) of channel length.

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TABLE 3 Frictional loss for viscous liquids (Hydraulic Institute Engineering Data Book, Reference 5)*

gpm	Pipe size	Viscosity, SSU 100	200	300	400	500	1000	1500	2000
	½	11.2	23.6	35.3	47.1	59	118	177	236
3	¾	3.7	7.6	11.5	15.3	19.1	38.2	57	76
	1	1.4	2.9	4.4	5.8	7.3	14.5	21.8	29.1
	¾	6.1	12.7	19.1	25.5	31.9	61	96	127
5	1	2.3	4.9	7.3	9.7	12.1	21.2	36.3	48.5
	1¼	0.77	1.6	2.4	3.3	4.1	8.1	12.2	16.2
	¾	8.5	17.9	26.8	35.7	44.6	89	134	178
7	1	3.2	6.8	10.2	13.6	17	33.9	51	68
	1¼	1.1	2.3	3.4	4.5	5.7	11.4	17	22.7
	1	4.9	9.7	14.5	19.4	24.2	48.5	73	97
10	1¼	1.6	3.3	4.9	6.5	8.1	16.2	24.3	32.5
	1½	0.84	1.8	2.6	3.5	4.4	8.8	13.1	17.5
	1	11	14.5	21.8	29.1	36.3	73	109	145
15	1¼	2.8	4.9	7.3	9.7	12.2	24.3	36.5	48.7
	1½	1.3	2.6	3.9	5.3	6.6	13.1	197	26.3
	1	18	18	29.1	38.8	48.5	97	145	194
	1¼	4.9	6.4	9.7	13	16.2	32.5	48.7	65
20	1½	2.3	3.5	5.3	7	8.8	17.5	26.3	35
	2	0.64	1.3	1.9	2.6	3.2	6.4	9.6	12.9
	1½	3.5	4.4	6.6	8.8	11	21.9	32.8	43.8
25	2	1	1.6	2.4	3.2	4	8	12.1	16.1
	2½	0.4	0.79	1.2	1.6	2	4	5.9	7.9
	1½	5	5.3	7.9	10.5	13.1	26.3	39.4	53
30	2	1.4	1.9	2.9	3.9	4.8	9.6	14.5	19.3
	2½	0.6	0.95	1.4	1.9	2.4	4.7	7.1	9.5
	1½	8.5	9	10.5	14	17.5	35	53	70
40	2	2.5	2.5	3.9	5.1	6.4	12.9	19.3	25.7
	2½	1.1	1.3	1.9	2.5	3.2	6.3	9.5	12.6
	1½	12.5	14	14	17.5	21.9	43.8	66	88
50	2	3.7	4	4.8	6.4	8	16.1	24.1	32.1
	2½	1.6	1.7	2.4	3.2	4	7.9	11.8	15.8
	2	5	5.8	5.8	7.7	9.6	19.3	28.9	38.5
60	2½	2.2	2.4	2.8	3.8	4.7	9.5	14.2	19
	3	0.8	0.8	1.2	1.6	2	4	6	8
	2½	2.8	3.2	3.4	4.4	5.5	11.1	16.6	22.1
70	3	1	1.1	1.4	1.9	2.3	4.6	7	9.3
	4	0.27	0.31	0.47	0.63	0.78	1.6	2.4	3.1
	2½	3.6	4.2	4.2	5.1	6.3	12.6	19	25.3
80	3	1.3	1.4	1.6	2.1	2.7	5.3	8	10.6
	4	0.36	0.36	0.54	0.72	0.89	1.8	2.7	3.6
	2½	5.3	6.1	6.4	6.4	8	15.8	23.7	31.6
100	3	1.9	2.2	2.2	2.7	3.3	6.6	9.9	13.3
	4	0.52	0.57	0.67	0.89	1.1	2.2	3.4	4.5

← TURBULENT FLOW → ← LAMINAR FLOW →

*Loss in pounds per square inch per 100 ft of new Schedule 40 steel pipe based on specific gravity of 1.00 of that liquid. For commercial installations, it is recommended that 15% be added to the values in this table.

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TABLE 3 Frictional loss for viscous liquids (Hydraulic Institute Engineering Data Book, Reference 5)* (*Continued*)

Viscosity, SSU									
2500	3000	4000	5000	6000	7000	8000	9000	10,000	15,000
294	353	471	589	706	824	942	...	...	...
96	115	153	191	229	268	306	344	382	573
36.3	43.6	58	73	87	101	116	131	145	218
159	191	255	319	382	446	510	573	637	956
61	73	97	121	145	170	194	218	242	363
20.3	24.3	32.5	40.6	48.7	57	65	73	81	122
223	268	357	416	535	624	713	803	892	...
85	102	136	170	203	237	271	305	339	509
28.4	34.1	45.4	57	68	80	91	102	114	170
121	145	194	242	291	339	388	436	485	727
40.6	48.7	65	81	97	114	130	146	162	243
21.9	26.3	35	43.8	53	61	70	79	88	131
182	218	291	363	436	509	581	654	727	...
61	73	97	122	146	170	195	219	243	365
32.8	39.4	53	66	79	92	105	118	131	197
242	291	388	485	581	678	775	872	...	...
81	97	130	162	195	227	260	292	325	487
43.8	53	70	88	105	123	140	158	175	263
16.1	19.3	25.7	32.1	38.5	45	51	58	64	96
55	66	88	110	131	153	176	197	219	328
20.1	24.1	32.1	40.2	48.2	56	61	72	80	121
9.9	11.8	15.8	19.7	23.7	27.6	31.6	35.5	39.5	59
66	79	105	131	158	184	210	237	263	394
24.1	28.9	38.5	48.2	58	67	77	87	96	145
11.8	14.2	19	23.7	28.4	33.2	37.9	42.6	47.4	71
88	105	140	175	210	245	280	315	350	526
32.1	38.5	51	64	77	90	103	116	129	193
15.8	19	25.3	31.6	37.9	44.2	51	57	63	95
110	131	175	219	263	307	350	394	438	657
40.2	48.2	64	80	96	112	129	145	161	241
19.7	23.7	31.6	39.5	47.4	55	63	71	79	118
48.2	58	77	96	116	135	154	173	193	289
23.7	28.4	37.9	47.4	57	66	76	85	95	142
9.9	11.9	15.9	19.9	23.9	27.9	31.8	35.8	39.8	60
27.6	33.2	44.2	55	66	77	88	100	111	166
11.6	13.9	18.6	23.2	27.8	32.5	37.1	41.7	46.4	70
3.9	4.7	6.3	7.8	9.4	11	12.5	14.1	15.6	23.5
31.6	37.9	51	63	76	88	101	114	126	190
13.3	15.9	21.2	26.5	31.8	37.1	42.4	47.7	53	80
4.5	5.4	7.2	8.9	10.7	12.5	14.3	16.1	17.9	26.8
39.5	47.4	63	79	95	111	127	142	158	237
16.6	19.9	26.5	33.1	39.8	46.4	53	60	66	99
5.6	6.7	8.9	11.2	13.4	15.6	17.9	20.1	22.3	33.5

← LAMINAR FLOW →

*For a liquid having a specific gravity other than 1.00, mulitply the value from the table by the specific gravity. No allowance for aging of pipe is included.

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TABLE 3 Frictional loss for viscous liquids (Hydraulic Institute Engineering Data Book, Reference 5) (*Continued*)

gpm	Pipe size	Viscosity, SSU 100	200	300	400	500	1000	1500	2000
	3	2.7	3.1	3.2	3.2	4	8	11.9	15.9
120	4	0.73	0.81	0.81	1.1	1.3	2.7	4	5.4
	6	0.98	0.11	0.16	0.21	0.26	0.52	0.78	1.0
	3	3.4	4	4.3	4.3	4.6	9.3	13.9	18.6
140	4	0.95	1.1	1.1	1.3	1.6	3.1	4.7	6.3
	6	0.17	0.18	0.21	0.28	0.35	0.69	1.0	1.4
	3	4.4	5	5.7	5.7	5.7	10.6	15.9	21.2
160	4	1.2	1.4	1.4	1.4	1.8	3.6	5.4	7.2
	6	0.17	0.18	0.21	0.28	0.35	0.69	1.0	1.4
	3	5.3	6.3	7	7	7	11.9	17.9	23.9
180	4	1.5	1.8	1.8	1.8	2	4	6	8
	6	0.2	0.24	0.24	0.31	0.39	0.78	1.2	1.6
	3	6.5	7.7	8.8	8.8	8.8	13.3	19.9	26.5
200	4	1.8	2.2	2.2	2.2	2.2	4.5	6.7	8.9
	6	0.25	0.3	0.3	0.35	0.43	0.87	1.3	1.7
	4	2.6	3.2	3.5	3.5	3.5	5.6	8.4	11.2
250	6	0.36	0.43	0.45	0.45	0.51	1.1	1.6	2.2
	8	0.95	0.12	0.12	0.15	0.18	0.36	0.54	0.72
	4	3.7	4.3	5	5	5	6.7	10.1	13.4
300	6	0.5	0.6	0.65	0.65	0.65	1.3	2	2.6
	8	0.13	0.17	0.17	0.18	0.22	0.43	0.65	0.87
	6	0.82	1	1.1	1.2	1.2	1.7	2.6	3.5
400	8	0.23	0.27	0.29	0.29	0.29	0.58	0.87	1.2
	10	0.08	0.09	0.1	0.1	0.12	0.23	0.35	0.47
	6	1.2	1.5	1.6	1.8	1.8	2.2	3.2	4.3
500	8	0.33	0.39	0.44	0.47	0.47	0.72	1.1	1.5
	10	0.11	0.14	0.15	0.15	0.15	0.29	0.44	0.58
	6	1.8	2.2	2.3	2.4	2.6	2.7	3.9	5.2
600	8	0.47	0.57	0.62	0.67	0.67	0.87	1.3	1.7
	10	0.16	0.18	0.2	0.22	0.22	0.35	0.52	0.07
	6	2.3	2.7	3	3.2	3.5	3.6	4.6	6.1
700	8	0.6	0.74	0.82	0.89	0.93	1	1.5	2
	10	0.2	0.25	0.27	0.3	0.3	0.41	0.61	0.82
	6	2.8	3.5	3.7	4	4.2	4.8	5.2	6.9
800	8	0.78	0.94	1	1.1	1.2	1.2	1.7	2.3
	10	0.26	0.3	0.34	0.38	0.4	0.47	0.7	0.92
	6	3.5	4.3	4.6	5.0	5.2	6	6	7.8
900	8	0.95	1.1	1.3	1.4	1.5	1.5	2	2.6
	10	0.32	0.37	0.43	0.46	0.5	0.52	0.79	1.1
	8	1.1	1.4	1.5	1.6	1.8	1.9	2.2	2.9
1000	10	0.38	0.45	0.5	0.55	0.6	0.6	0.87	1.2
	12	0.17	0.2	0.22	0.24	0.25	0.29	0.43	0.58

← TURBULENT FLOW → ← LAMINAR FLOW →

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TABLE 3 Frictional loss for viscous liquids (Hydraulic Institute Engineering Data Book, Reference 5) (*Continued*)

Viscosity, SSU									
2500	3000	4000	5000	6000	7000	8000	9000	10,000	15,000
19.9	23.9	31.8	39.8	47.7	56	61	72	80	119
6.7	8	10.7	13.4	16.1	18.8	21.4	24.1	26.8	40.2
1.3	1.6	2.1	2.6	3.1	3.6	1.2	4.7	5.2	7.8
23.2	27.8	37.1	46.4	56	65	74	84	93	139
7.8	9.4	12.5	15.6	18.8	21.9	25	28.2	31.3	46.9
1.5	1.8	2.4	3.0	3.6	4.2	4.9	5.5	6.0	9.1
26.5	31.8	42.4	53	64	74	85	95	106	159
8.9	10.7	14.3	17.9	21.5	25	28.6	32.2	35.7	54
1.7	2.1	2.8	3.5	4.2	4.9	5.5	6.2	6.9	10.4
29.8	35.8	47.7	60	72	84	95	107	119	179
10.1	12.1	16.1	20.1	24.1	28.1	32.2	36.2	40.2	60
2	2.3	3.1	3.9	4.7	5.5	6.2	7	7.8	11.7
33.1	39.8	53	66	80	93	106	119	133	199
11.2	13.4	17.9	22.3	26.8	31.3	35.7	40.2	44.7	67
2.2	2.6	3.5	4.3	5.2	6.1	6.9	7.8	8.7	13
14	16.8	22.3	27.9	33.5	39.1	44.7	50	56	84
2.7	3.3	4.3	5.4	6.5	7.6	8.7	9.8	10.8	16.3
0.9	1.1	1.5	1.8	2.2	2.5	2.9	3.3	3.6	5.4
16.8	20.1	26.8	33.5	40.2	47	54	60	67	101
3.3	3.9	5.2	6.5	7.8	9.1	10.4	11.7	13	19.5
1.0	1.3	1.7	2.2	2.6	3	3.5	3.9	4.3	6.5
4.3	5.2	6.9	8.7	10.4	12.1	13.9	15.6	17.3	26
1.5	1.7	2.3	2.9	3.5	4.1	4.6	5.2	5.8	8.7
0.58	0.7	0.93	1.2	1.4	1.6	1.9	2.1	2.3	3.5
5.4	6.5	8.7	10.8	13	15.2	17.3	19.5	21.7	32.5
1.8	2.2	2.9	3.6	4.3	5.1	5.8	6.5	7.2	10.8
0.73	0.87	1.2	1.5	1.8	2	2.3	2.6	2.9	4.4
6.5	7.8	10.4	13	16	18.2	20.8	23.4	26	39
2.2	2.6	3.5	4.3	5.2	6.1	6.9	7.8	8.7	13
0.87	1.1	1.4	1.8	2.1	2.4	2.8	3.3	3.5	5.2
7.6	9.1	12.1	15.2	18.4	21.2	24.3	27.3	30.3	45.5
2.5	3	4.1	5.1	6.1	7.1	8.1	9.1	10.1	15.2
1	1.2	1.6	2	2.4	2.9	3.3	3.7	4.1	6.1
8.7	10.4	13.9	17.3	20.8	24.3	27.7	31.2	34.7	52
2.9	3.5	4.6	5.8	6.9	8.1	9.3	10.4	11.6	17.3
1.2	1.4	1.9	2.3	2.8	3.3	3.7	4.2	4.7	7
9.8	11.7	15.6	19.5	23.4	27.3	31.2	35.1	39	58.5
3.3	3.9	5.2	6.5	7.8	9.1	10.4	11.7	13	19.5
1.3	1.6	2.1	2.6	3.1	3.7	4.2	4.7	5.2	7.9
3.6	4.3	5.8	7.2	8.7	10.1	11.6	13	14.5	21.7
1.5	1.8	2.3	2.9	3.5	4.1	4.7	5.2	5.8	8.7
0.72	0.87	1.2	1.5	1.7	2	2.3	2.6	2.9	4.3

← LAMINAR FLOW →

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TABLE 3 Frictional loss for viscous liquids (Hydraulic Institute Engineering Data Book, Reference 5) (*Continued*)

gpm	Pipe size	Viscosity, SSU 20,000	25,000	30,000	40,000	50,000	60,000
	2	19.3	24.1	28.9	38.5	48.2	58
3	2½	9.5	11.8	14.2	19	23.7	28.4
	3	4	5	6	8	9.9	11.9
	2	32	40	48.2	64	80	96
5	2½	15.8	19.7	23.7	31.6	39.5	47.4
	3	6.6	8.3	9.9	13.3	16.6	9.9
	2	45	56	67	90	112	135
7	2½	22.1	27.6	33.2	44.2	55	66
	3	9.3	10.6	13.9	18.6	23.2	27.8
	2½	31.6	39.5	47.4	63	79	95
10	3	13.3	16.6	19.9	26.5	33.1	39.8
	4	4.5	5.6	6.7	8.9	11.2	13.4
	2½	47.4	59	71	95	118	142
15	3	19.9	24.9	29.8	39.8	49.7	60
	4	6.7	8.4	10.1	13.4	16.8	20.1
	3	26.5	33.1	39.8	53	66	80
20	4	8.9	11.2	13.4	17.9	22.3	26.8
	6	1.7	2.2	2.6	3.5	4.3	5.2
	3	33.1	41.4	49.7	66	83	99
25	4	11.2	14	16.8	22.3	27.9	33.5
	6	2.2	2.7	3.3	4.3	5.4	6.5
	3	39.8	49.7	60	80	99	119
30	4	13.4	16.8	20.1	26.8	33.5	40.2
	6	2.6	3.3	3.9	5.2	6.5	7.8
	3	53	66	80	106	133	160
40	4	17.9	22.3	26.8	35.7	44.7	54
	6	3.5	4.3	5.2	6.9	8.7	10.4
	4	22.3	27.9	33.5	44.7	56	67
50	6	4.3	5.4	6.5	8.7	10.8	13
	8	1.5	1.8	2.7	2.9	3.6	4.3
	4	26.8	33.5	40.2	54	67	80
60	6	5.2	6.5	7.8	10.4	13	16
	8	1.7	2.2	2.6	3.5	4.3	5.2
	4	31.3	39.1	46.9	63	78	94
70	6	6.1	7.6	9.1	12.1	15.2	18.4
	8	2	2.5	3	4.1	5.1	6.1
	6	6.9	8.7	10.4	13.9	17.3	20.8
80	8	2.3	2.9	3.5	4.6	5.8	6.9
	10	0.93	1.2	1.4	1.9	2.3	2.8
	6	7.8	9.8	11.7	15.6	19.5	23.4
90	8	2.6	3.3	3.9	5.2	6.5	7.8
	10	1.1	1.3	1.6	2.1	2.6	3.1

← LAMINAR FLOW →

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TABLE 3 Frictional loss for viscous liquids (Hydraulic Institute Engineering Data Book, Reference 5) (*Continued*)

Viscosity, SSU								
70,000	80,000	90,000	100,000	125,000	150,000	175,000	200,000	500,000
67	77	87	96	120	145	169	193	482
332	37.9	42.6	47.4	59	71	83	95	237
13.9	15.9	17.9	19.9	24.9	29.8	34.8	39.8	99
112	129	145	161	200	241	281	321	803
55	63	71	79	99	118	138	158	395
23.2	26.5	29.8	33	41.4	49.7	58	66	166
157	180	202	225	281	337	393	450	. . .
77	88	100	111	138	166	194	221	553
32.5	37.1	40.7	46.4	58	70	81	93	232
111	126	142	158	197	237	276	316	790
46.4	53	60	66	83	99	116	133	331
15.6	17.9	20.1	22.3	27.9	33.5	39.1	44.7	112
166	190	213	237	296	355	415	474	. . .
70	80	89	99	124	149	174	199	497
23.5	26.8	30.2	33.5	41.9	50	59	67	168
93	106	119	133	166	199	232	265	663
31.3	35.7	40.2	44.7	56	67	78	89	223
6.1	6.9	7.8	8.7	10.8	13	15.2	17.3	43.3
116	133	149	166	207	49	290	331	828
39.1	44.7	50	56	70	84	98	112	279
7.6	8.7	9.8	10.8	13.5	16.3	19	21.7	54
139	159	179	199	249	298	348	398	. . .
46.9	54	60	67	84	101	117	134	335
9.1	10.4	11.7	13	16.3	19.5	22.7	26	65
186	212	239	265	331	398	464	532	. . .
63	72	80	89	112	134	156	179	447
12.1	13.9	15.6	17.3	21.7	26	30.3	34.7	87
78	89	101	112	140	168	196	223	559
15.2	17.3	19.5	21.7	27.1	32.5	37.9	43.3	107
5.1	5.8	6.5	7.2	9	10.8	12.6	14.5	36.1
94	107	121	134	168	201	235	268	670
18.2	20.8	23.4	26	32.5	39	45.5	52	130
6.1	6.9	7.8	8.7	10.8	13	15.2	17.3	43.4
110	125	141	156	196	235	274	313	782
21.2	24.3	27.3	30.3	37.9	45.5	53	61	152
7.1	8.1	9.1	10.1	12.6	15.2	17.7	20.2	51
24.3	27.7	31.2	34.7	43.3	52	61	69	173
8.1	9.3	10.4	11.6	14.5	17.3	20.2	23.1	58
3.3	3.7	4.2	4.7	5.8	7	8.2	9.3	23.3
27.3	31.2	35.1	39	48.7	59	68	78	195
9.1	10.4	11.7	13	16.3	19.5	22.8	26	65
3.7	4.2	4.7	5.2	6.6	7.9	9.2	10.5	26.2

← LAMINAR FLOW →

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TABLE 3 Frictional loss for viscous liquids (Hydraulic Institute Engineering Data Book, Reference 5) (*Continued*)

gpm	Pipe size	Viscosity, SSU 20,000	25,000	30,000	40,000	50,000	60,000
	6	8.7	10.8	13	17.3	21.7	26
100	8	2.9	3.6	4.3	5.8	7.2	8.7
	10	1.2	1.5	1.8	2.3	2.9	3.5
	6	10.4	13	15.6	20.8	26	31.2
120	8	3.5	4.3	5.2	6.9	8.7	10.4
	10	1.4	1.8	2.1	2.8	3.5	4.2
	6	12.1	15.2	18.2	24.3	30.3	36.4
140	8	4	5.1	6.1	8.1	10.1	12.1
	10	1.7	2	2.4	3.3	4.1	4.9
	6	13.9	17.3	20.8	27.7	34.7	41.6
160	8	4.6	5.8	6.9	9.3	11.6	13.8
	10	1.9	2.3	2.8	3.7	4.7	5.6
	6	15.6	19.5	23.4	31.2	39	46.8
180	8	5.2	6.5	7.8	10.4	13	15.6
	10	2.1	2.6	3.1	4.2	5.2	6.3
	8	5.8	7.2	8.7	11.6	14.5	17.3
200	10	2.3	2.9	3.5	4.7	5.8	7
	12	1.2	1.5	1.7	2.3	2.9	3.5
	8	7.2	9	10.8	14.5	18.1	21.7
250	10	2.9	3.6	4.4	5.8	7.3	8.7
	12	1.5	1.8	2.2	2.9	3.6	4.3
	8	8.7	10.8	13	17.3	21.7	26
300	10	3.5	4.4	5.2	7	8.7	10.5
	12	1.7	2.2	2.6	3.5	4.3	5.2
	8	11.6	14.5	17.3	23	28.9	34.7
400	10	4.7	5.8	7	9.3	11.6	14
	12	2.3	2.9	3.5	4.6	5.8	7
	8	14.5	18.1	21.7	28.9	36.1	43.4
500	10	5.8	7.3	8.7	11.6	14.6	17.5
	12	2.9	3.6	4.3	5.8	7.2	8.7
	8	17.3	21.7	26	34.7	43.4	52
600	10	7	8.7	10.5	14	17.5	21
	12	3.5	4.3	5.2	7	8.7	10.4
	8	20.2	25.3	30.3	40.5	51	61
700	10	8.2	10.2	12.2	16.3	20.4	24.4
	12	4.1	5.1	6.1	8.1	10.1	12.2
	8	23.1	28.9	34.7	46.2	58	69
800	10	9.3	11.6	14	18.6	23.3	27.9
	12	4.6	5.8	7	9.3	11.6	13.9
	8	26	32.5	39	52	65	78
900	10	10.5	13.1	15.7	21	26.2	31.4
	12	5.2	6.5	7.8	10.4	13	15.6
	8	28.9	36.1	43.4	58	72	87
1000	10	11.6	14.6	17.5	23.3	29.1	34.9
	12	5.8	7.2	8.7	11.6	14.5	17.4

TABLE 3 Frictional loss for viscous liquids (Hydraulic Institute Engineering Data Book, Reference 5) (*Continued*)

Viscosity, SSU								
70,000	80,000	90,000	100,000	125,000	150,000	175,000	200,000	500,000
30.3	34.7	39	43.3	54	65	76	87	217
10.1	11.6	13	14.5	18.1	21.7	25.3	28.9	72
4.2	4.7	5.2	5.8	7.3	8.7	10.2	11.6	29.1
36.4	41.6	46.8	52	65	78	91	104	260
12.1	13.9	15.6	17.3	21.7	26	30.4	34.7	87
4.9	5.6	6.3	7	8.7	10.5	12.2	14	34.9
42.5	48.5	55	61	76	91	106	121	303
14.2	16.2	18.2	20.2	25.3	30.4	35.4	40.5	101
5.7	6.5	7.3	8.1	10.2	12.2	14.3	16.3	40.7
48.5	56	62	69	87	104	121	139	347
16.2	18.5	20.8	23.1	28.9	34.7	40.5	46.2	116
6.5	7.5	8.4	9.3	11.6	14	16.3	18.6	46.6
55	62	70	78	98	117	137	156	390
18.2	20.8	23.4	26	32.5	39	45.5	52	130
7.3	8.4	9.4	10.5	13.1	15.7	18.3	21	52
20.2	23.1	26	28.9	36.1	43.4	51	58	145
8.2	9.3	10.5	11.6	14.6	17.5	20.4	23.3	58
4.1	4.6	5.2	5.8	7.2	8.7	10.1	11.6	28.9
25.3	28.9	32.5	36.1	45.2	54	63	72	181
10.2	11.6	13.1	14.6	18.2	21.8	25.5	29.1	73
5.1	5.8	6.5	7.2	9	10.9	12.7	14.5	36.2
30.4	34.7	39	43.4	54	65	76	87	217
12.2	14	15.7	17.5	21.8	26.2	30.6	34.9	87
6.1	7	7.8	8.7	10.9	13	15.2	17.4	43.4
40.5	46.2	52	58	72	87	101	116	289
16.3	18.6	21	23.3	29.6	34.9	40.7	46.6	116
8.1	9.3	10.4	11.6	14.5	17.4	20.3	23.2	58
51	58	65	72	90	108	126	145	361
20.4	23.3	26.2	29.1	36.4	43.7	51	58	146
10.1	11.6	13	14.5	18.1	21.7	25.3	28.9	72
61	69	78	87	107	130	152	173	434
24.4	27.9	31.4	34.9	43.7	52	61	70	175
12.2	13.9	15.6	17.4	21.7	26.1	30.4	34.7	87
71	81	91	101	126	152	177	202	506
28.5	32.6	36.7	40.7	51	61	71	82	204
14.2	16.2	18.2	20.3	25.3	30.4	35.5	40.5	101
81	93	104	116	145	173	202	231	578
32.6	37.3	41.9	46.6	58	70	82	93	233
16.2	18.5	20.8	23.1	28.9	34.7	40.5	46.3	116
91	104	117	130	163	195	228	260	650
36.7	41.9	47.1	52	66	79	92	105	262
18.2	20.8	23.4	26.1	32.6	39.1	45.6	52	130
101	116	130	145	181	217	253	289	723
40.7	46.6	52	58	72	87	102	116	291
20.3	23.2	26.1	28.9	36.2	43.4	51	58	145

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LAMINAR FLOW

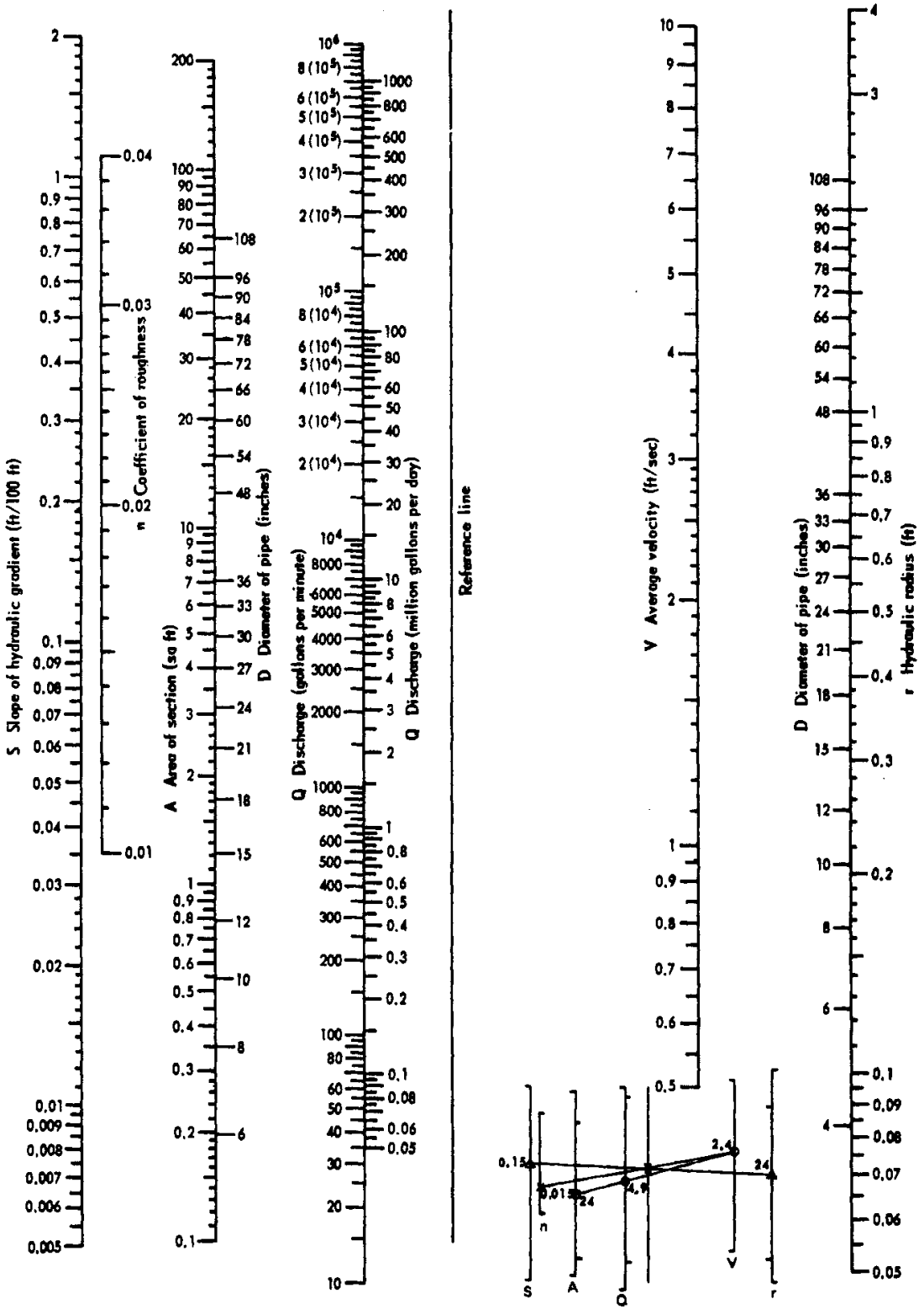


FIGURE 35 Nomogram for the solution of the Manning formula. Obtain values of n from Table 4. To solve for S, align A with Q and read V, align V and n intersecting reference line, align r with reference line intersection, and read S (1 m = 3.28 ft; 1 m = 39.37 in; 1 m^2 = 1550 in^2; 1 m^3/h = 4.4 gpm; 1 m/s = 3.28 ft/s; 1 m/m = 1 ft/ft).

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TABLE 4 Values of friction factor n to be used with the Manning Formula in Figure 35

Surface	Rest	Good	Fair	Bad
Uncoated cast iron pipe	0.012	0.013	0.014	0.015
Coated cast iron pipe	0.011	0.012[a]	0.013[a]	
Commercial wrought iron pipe, black	0.012	0.013	0.014	0.015
Commercial wrought iron pipe, galvanized	0.013	0.014	0.015	0.017
Smooth brass and glass pipe	0.009	0.010	0.011	0.013
Smooth lockbar and welded OD pipe	0.010	0.011[a]	0.013[a]	
Riveted and spiral steel pipe	0.013	0.015[a]	0.017[a]	
Vitrified sewer pipe	0.010	0.013[a]	0.015	0.017
	0.011			
Common clay drainage tile	0.011	0.012[a]	0.014[a]	0.017
Glazed brickwork	0.011	0.012	0.013[a]	0.015
Brick in cement mortar; brick sewers	0.012	0.013	0.015[a]	0.017
Neat cement surfaces	0.010	0.011	0.012	0.013
Cement mortar surfaces	0.011	0.012	0.013[a]	0.015
Concrete pipe	0.012	0.013	0.015[a]	0.016
Wood-stave pipe	0.010	0.011	0.012	0.013
Plank flumes:				
Planed	0.010	0.012[a]	0.013	0.014
Unplaned	0.011	0.013[a]	0.014	0.015
With battens	0.012	0.015[a]	0.016	
Concrete-lined channels	0.012	0.014[a]	0.016[a]	0.018
Cement-rubble surface	0.017	0.020	0.025	0.030
Dry-rubble surface	0.025	0.030	0.033	0.035
Dressed-ashlar surface	0.013	0.014	0.015	0.017
Semicircular metal flumes, smooth	0.011	0.012	0.013	0.015
Semicircular metal flumes, corrugated	0.0225	0.025	0.0275	0.0.30
Canals and ditches:				
Earth, straight, and uniform	0.017	0.020	0.0225[a]	0.025
Rock cuts, smooth, and uniform	0.025	0.030	0.033[a]	0.035
Rock cuts, jagged, and irregular	0.035	0.040	0.045	
Winding sluggish canals	0.0225	0.025[a]	0.0275	0.030
Dredged earth channels	0.025	0.0275[a]	0.030	0.033
Canals with rough stony beds, weeds on earth banks	0.025	0.030	0.035[a]	0.040
Earth bottom, rubble sides	0.028	0.030[a]	0.033[a]	0.035
Natural stream channels:				
(1) Clean, straight bank, full stage, no rifts, or deep pools	0.025	0.0275	0.030	0.033
(2) Same as (1), but some weeds and stones	0.030	0.033	0.035	0.040
(3) Winding, some pools and shoals, clean	0.033	0.035	0.040	0.045
(4) Same as (3), lower stages, more ineffective slope and sections	0.040	0.045	0.050	0.055
(5) Same as (3), some weeds and stones	0.035	0.040	0.045	0.050
(6) Same as (4), stony sections	0.045	0.050	0.055	0.060
(7) Sluggish river reaches, rather weedy or with very deep pools	0.050	0.060	0.070	0.080
(8) Very weedy reaches	0.075	0.100	0.125	0.150

[a]Values commonly used in designing.

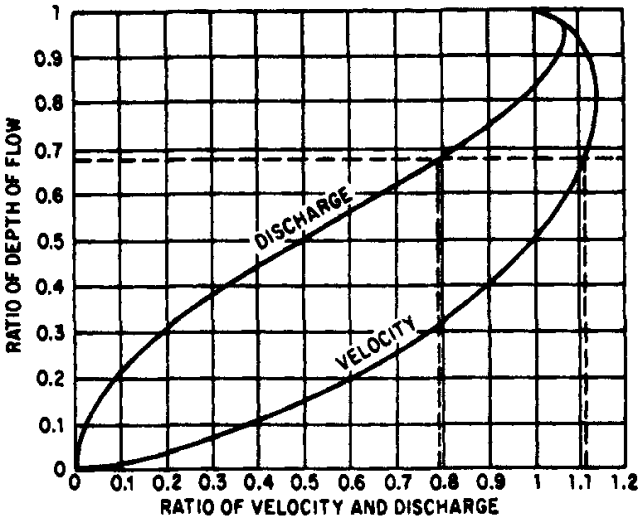


FIGURE 36 Discharge velocity of a partially full circular pipe versus that of a full pipe

A rearrangement of Eq. 18 gives

in USCS units
$$S = h_f = \left(\frac{Vn}{1.486r^{2/3}}\right)^2 \tag{19a}$$

in SI units
$$S = h_f = \left(\frac{Vn}{r^{2/3}}\right)^2 \tag{19b}$$

The following examples illustrate the application of the Manning formula (Eq. 19), Figures 35 and 36, and Table 4 to the solution of problems involving the flow of water in open channels.

EXAMPLE 10 The flow through a 24-in (61.0-cm) ID commercial wrought iron pipe in fair condition is 4.9 mgd (772.7 m^3/h). Determine the loss of head as a result of friction in feet per 100 ft (meters per 100 m) of pipe and the slope required to maintain a full, uniformly flowing pipe.

In USCS units
$$V = \frac{\text{gpm}}{(\text{pipe ID in inches})^2} \times 0.408 = \frac{4.9 \times 10^6 \times 0.408}{24 \times 60 \times 24^2} = 2.41 \text{ ft/s}$$

In SI units
$$V = \frac{\text{m}^3/\text{h}}{(\text{pipe ID in cm})^2} \times 3.54 = \frac{772.7}{60.96^2} \times 3.54 = 0.736 \text{ m/s}$$

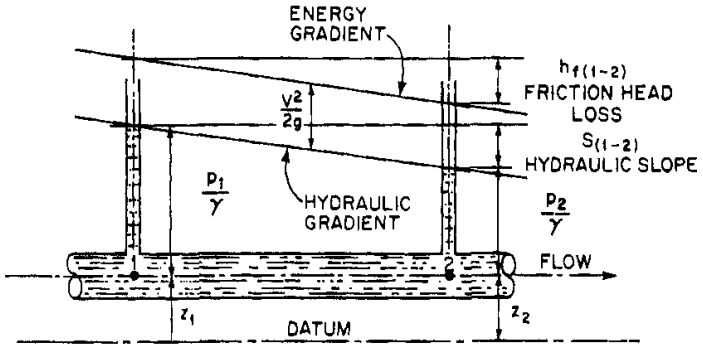


FIGURE 37 Slopes of energy and hydraulic gradients measure frictional head loss ft/ft (m/m) of pipe length

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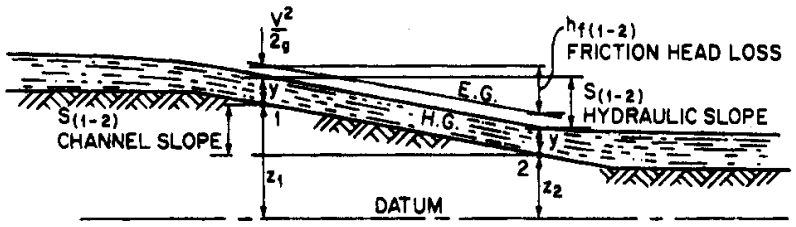


FIGURE 38 In an open channel with uniform flow, slopes of channel bottom, energy, and hydraulic gradients are the same as frictional head loss ft/ft (m/m) of channel length

From Table 4, $n = 0.015$.

In USCS units

$$r = D/4 = 2/4 = 0.5 \text{ ft} \qquad \text{(hydraulic radius)}$$

$$S = h_f = \left(\frac{Vn}{1.486r^{2/3}}\right)^2 = \left(\frac{2.41 \times 0.015}{1.486 \times 0.5^{2/3}}\right)^2 \qquad \text{(from Eq. 19a)}$$

$$= 0.0015 \text{ ft/ft}$$

$$100 \times 0.0015 = 0.15 \text{ ft/100 ft} \qquad \text{(slope and frictional head)}$$

In SI units

$$r = D/4 = 0.6096/4 = 0.1524 \text{ m} \qquad \text{(hydraulic radius)}$$

$$S = h_f = \left(\frac{Vn}{r^{2/3}}\right)^2 = \left(\frac{0.736 \times 0.015}{0.1524^{2/3}}\right)^2 \qquad \text{(from Eq. 19b)}$$

$$= 0.0015 \text{ m/m}$$

$$100 \times 0.0015 = 0.15 \text{ m/100 m} \qquad \text{(slope and frictional head)}$$

The problem may also be solved by using Figure 35 and following the trace lines:

$$S = h_f = 0.15 \text{ ft/100 ft (m/100 m)}$$

EXAMPLE 11 Determine what the flow and velocity would be if the pipe in Example 10 were flowing two-thirds full and were laid on the same slope.

Follow the trace lines in Figure 36 and note that the multipliers for discharge and velocity are 0.79 and 1.11, respectively. Therefore

in USCS units

$$\text{Flow} = 0.79 \times 4.9 = 3.87 \text{ mgd}$$

$$\text{Velocity} = 1.11 \times 2.41 = 2.68 \text{ ft/s}$$

in SI units

$$\text{Flow} = 0.79 \times 772.7 = 610 \text{ m}^3\text{/h}$$

$$\text{Velocity} = 1.11 \times 0.736 = 0.817 \text{ m/s}$$

Pipe Fittings Invariably, a system containing piping will have connections that change the size or direction of the conduit. These fittings add frictional losses, called *minor losses,* to the system head. Fitting losses are generally the result of changes in velocity or direction. A decreasing velocity results in more loss in head than an increasing velocity, as the former causes energy-dissipating eddies. Experimental results have

indicated that minor losses vary approximately as the square of the velocity through the fittings.

VALVES AND STANDARD FITTINGS The resistance to flow through valves and fittings may be found in References 4, 5, 17, and other sources. Losses are usually expressed in terms of a *resistance coefficient K* and the average velocity head in a pipe having the same diameter as the valve or fitting. The frictional resistance h in feet (meters) is found from the equation

$$h = K\frac{V^2}{2g} \tag{20}$$

where K = resistance coefficient, which depends on design and size of valve or fitting
V = average velocity in pipe of corresponding internal diameter, ft/s (m/s)
g = acceleration of gravity, 32.17 ft/s^2 (9.807 m/s^2)

A comparison of the Darcy-Weisbach equation (Eq. 16) with Eq. 20 suggests that K equals $f(L/D)$, where L is the equivalent length of pipe in feet (meters) and D is the inside pipe diameter in feet (meters), to produce the same head loss in a straight pipe as through a valve or fitting. The friction in an "equivalent length of pipe" has been another method used to estimate head loss through valves and fittings. Values of the ratio L/D have been experimentally determined. This ratio multiplied by the inside diameter of a pipe of specified schedule for the valve or fitting being considered gives the equivalent length of pipe to use to calculate the head lost.

Loss of head in straight pipe depends on the friction factor or Reynolds number. However, with valves and fittings, head is lost primarily because of change in direction of flow, change in cross section, and obstructions in the flow path. For this reason, the resistance coefficient is practically constant for a particular shape of valve or fitting for all flow conditions, including laminar flow. The resistance coefficient would theoretically be constant for all sizes of a particular design of valve or fitting except that all sizes are not geometrically similar. The Crane Company has reported the results of tests that show that the resistance coefficient for a number of lines of valves and fittings decreases with increasing size at flow conditions of equal friction factor and that the equivalent length L/D tends to be constant for the various sizes at the same flow conditions.

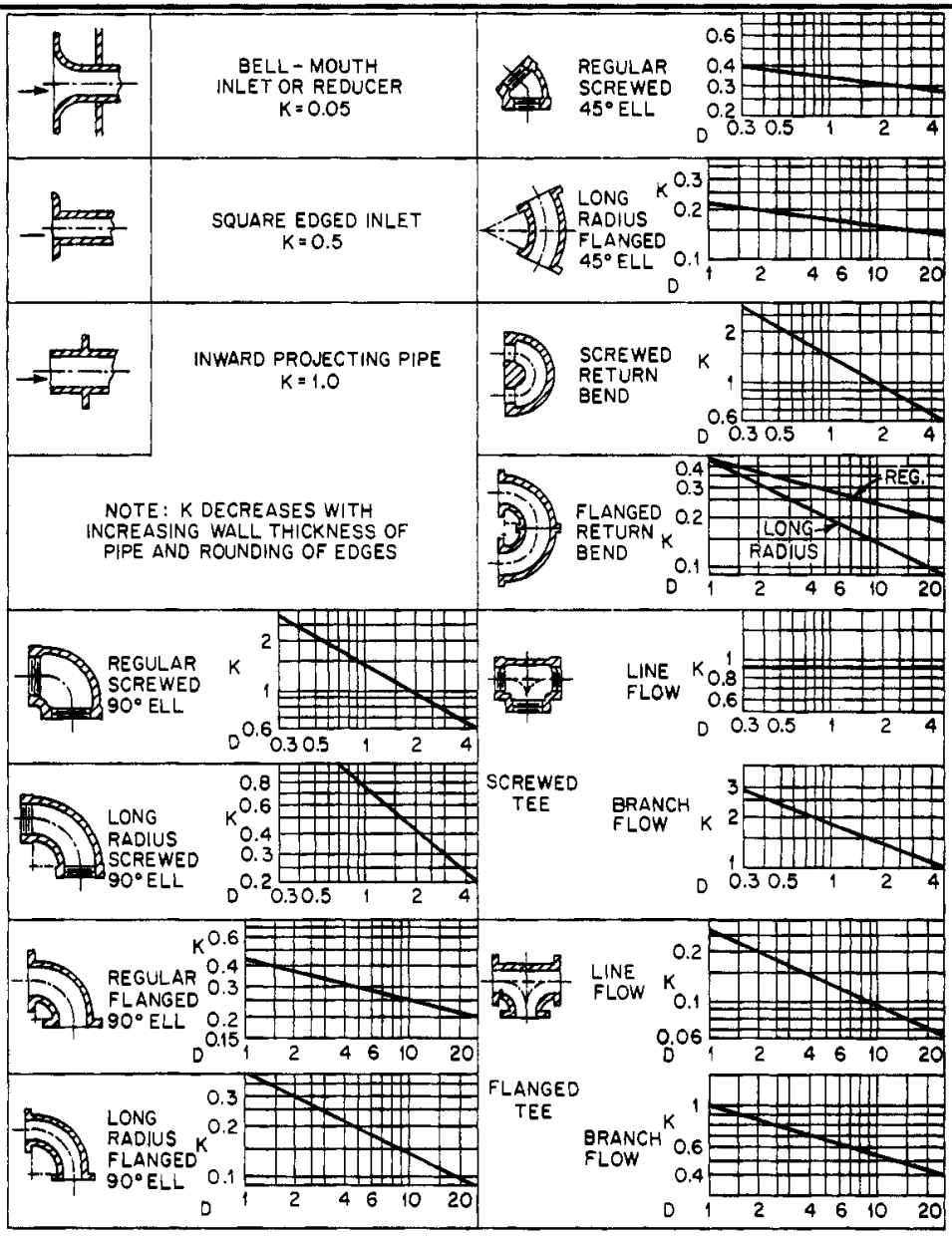
When available, the K factor furnished by the valve or fitting manufacturer should always be used rather than the value from a general listing. The Hydraulic Institute lists losses in terms of K through valves and fittings (Tables 5A to 5C); these losses vary with size of the valve or fitting but are independent of friction factor. The Crane Company provides a similar listing of K values (Tables 6A to 6E). The latter listing of flow coefficients is associated with the velocity head $V^2/2g$ that would occur through the internal diameter of the schedule pipes for the various ANSI classes of valves and fittings shown in Table 6e. If the connecting pipe is of a different size or schedule, either use the velocity for the pipe shown in Table 6a or use the actual pipe velocity head and correct the resistance coefficient obtained from this table by the multiplier

$$\left(\frac{\text{Actual pipe } ID}{\text{Standard pipe } ID}\right)^4$$

Tables 6 are based on the use of an equivalent length constant for complete turbulent flow for each valve or fitting shown. This constant is shown as the multiplier of the friction factor f_T for the corresponding clean commercial steel pipe with completely turbulent flow. The product $(L/D)(f_T)$ is the coefficient K. The friction factors are given in Table 6a for different pipe sizes, or they can be obtained from Figure 31 or 32. If the valve or fitting has a sudden or gradual contraction, enlargement, or change in direction of flow, appropriate formulas for these conditions are given for the determination of K. If flow is laminar, valve and fitting resistance coefficients are obtained from Table 6a based on completely turbulent

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TABLE 5A Resistance coefficients for K for valves and fittings



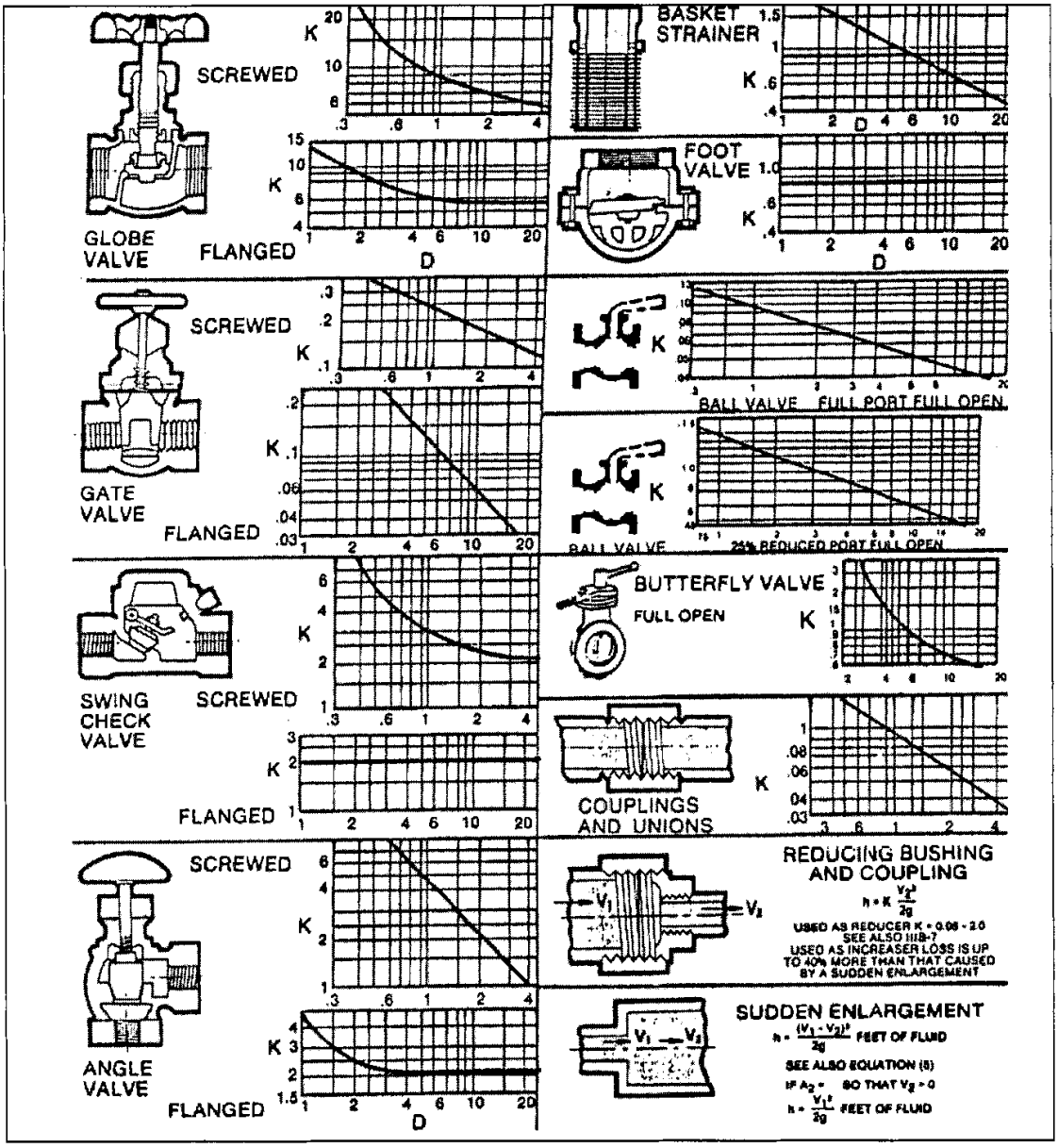
NOTE: D = nominal iron pipe size in inches (in × 25.4 = mm). (Hydraulic Institute Engineering Data Book, Reference 5)

flow, but the pipe frictional loss is calculated using the laminar friction factor $f = 64/Re$ instead of f_T. Also shown in Tables 6 for check valves is the minimum pipe velocity required for full disk lift for the coefficient of resistance listed ($\bar{V}$ = liquid specific volume in cubic feet per pound).

Prior to the 15th printing (1976) of the Crane Company Technical Paper 410, and as shown in the first edition of this text, valve and fitting losses were calculated using an equivalent length of pipe rather than the coefficient K. The Crane Company states that this conceptual change regarding the values of equivalent length L/D and resistance coefficient K for valves and fittings relative to the friction factor in pipes has a relatively minor

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TABLE 5B Resistance coefficients K for valves and fittings



NOTE: D = nominal iron pipe size in inches (in × 25.4 = mm). For velocities below 15 ft/s (4.6 m/s), check valves and foot valves will be only partially open and will exhibit higher values of K than shown.

$$h = K\frac{V^2}{2g}, \text{ ft (m) of fluid}$$

(Hydraulic Institute Engineering Data Book, Reference 5)

effect on most problems dealing with turbulent flow and avoids a significant overstatement of pressure drop in the laminar zone.

Valve Flow Coefficient The loss of head through valves, particularly control valves, is often expressed in terms of a *flow coefficient* C_v in USCS units (K_v in SI units). The flow of water in gallons per minute (cubic meters per hour) at 60°F (15.6°C) that will pass through a valve with a 1-lb/in^2 (1-bar) pressure drop is defined as the flow coefficient for a particular valve opening. Because loss of head h is a measure of energy loss per unit weight (force) and because $h = p/\gamma$ and head loss varies directly with the square of the flow through a certain fixed opening, the formulas for flow coefficient are

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TABLE 5C Approximate variation for K listed in Tables 5a and 5b

	Fitting	Range of variation, %
90° elbow	Regular screwed	±20 above 2-in size[a]
	Regular screwed	±40 below 2-in size
	Long radius, screwed	±25
	Regular flanged	±35
	Long radius, flanged	±30
45° elbow	Regular screwed	±10
	Long radius, flanged	±10
180° bend	Regular screwed	±25
	Regular flanged	±35
	Long radius, flanged	±30
T	Screwed, line or branch flow	±25
	Flanged, line or branch flow	±35
Globe valve	Screwed	±25
	Flanged	±25
Gate valve	Screwed	±25
	Flanged	±50
Check valve[b]	Screwed	±30
	Flanged	+200 / −80
Sleeve check valve	——	Multiply flanged values by 0.2 to 0.5
Tilting check valve	——	Multiply flanged values by 0.13 to 0.19
Drainage gate check	——	Multiply flanged values by 0.03 to 0.07
Angle valve	Screwed	±20
	Flanged	±50
Basket strainer	——	±50
Foot valve[a]	——	±50
Couplings	——	±50
Unions	——	±50
Reducers	——	±50

[a]In × 25.4 = mm.

[b]For velocities below 15 ft/s (4.6 m/s), check valves and foot valves will be only partially open and will exhibit higher values of K than shown.

(Hydraulic Institute Engineering Data Book, Reference 5)

In USCS units
$$C_v = \text{gpm}\sqrt{\frac{\text{sp. gr.}}{\text{lb/in}^2}} \tag{21a}$$

also
$$C_v = 29.9d^2/\sqrt{K} \tag{22a}$$

In SI units
$$K_v = \text{m}^3/\text{h}\sqrt{\frac{\text{sp. gr.}}{\text{bar}}} \tag{21b}$$

also
$$K_v = 0.04d^2/\sqrt{K} \tag{22b}$$

where d= internal diameter of pipe corresponding to K and as shown in Table 6E, in (mm) and 1 bar = 100 kPa. The conversion from the SI flow coefficient to the USCS flow coefficient is

TABLE 6A Resistance coefficient K for valves and fittings

PIPE FRICTION DATA FOR CLEAN COMMERCIAL STEEL PIPE WITH FLOW IN ZONE OF COMPLETE TURBULENCE

Nominal Size	½"	¾"	1"	1¼"	1½"	2"	2½, 3"	4"	5"	6"	8-10"	12-16"	18-24"
Friction Factor (f_T)	.027	.025	.023	.022	.021	.019	.018	.017	.016	.015	.014	.013	.012

FORMULAS FOR CALCULATING "K" FACTORS FOR VALVES AND FITTINGS WITH REDUCED PORT

• **Formula 1**

$$K_2 = \frac{0.8 \sin\frac{\theta}{2}(1-\beta^2)}{\beta^4} = \frac{K_1}{\beta^4}$$

• **Formula 2**

$$K_2 = \frac{0.5\,(1-\beta^2)\sqrt{\sin\frac{\theta}{2}}}{\beta^4} = \frac{K_1}{\beta^4}$$

• **Formula 3**

$$K_2 = \frac{2.6 \sin\frac{\theta}{2}(1-\beta^2)^2}{\beta^4} = \frac{K_1}{\beta^4}$$

• **Formula 4**

$$K_2 = \frac{(1-\beta^2)^2}{\beta^4} = \frac{K_1}{\beta^4}$$

• **Formula 5**

$$K_2 = \frac{K_1}{\beta^4} + \text{Formula 1} + \text{Formula 3}$$

$$K_2 = \frac{K_1 + \sin\frac{\theta}{2}[0.8\,(1-\beta^2) + 2.6\,(1-\beta^2)^2]}{\beta^4}$$

• **Formula 6**

$$K_2 = \frac{K_1}{\beta^4} + \text{Formula 2} + \text{Formula 4}$$

$$K_2 = \frac{K_1 + 0.5\sqrt{\sin\frac{\theta}{2}}(1-\beta^2) + (1-\beta^2)^2}{\beta^4}$$

• **Formula 7**

$$K_2 = \frac{K_1}{\beta^4} + \beta\,(\text{Formula 2} + \text{Formula 4}) \text{ when } \theta = 180°$$

$$K_2 = \frac{K_1 + \beta\left[0.5\,(1-\beta^2) + (1-\beta^2)^2\right]}{\beta^4}$$

NOTES:

$$\beta = \frac{d_1}{d_2}$$

$$\beta^2 = \left(\frac{d_1}{d_2}\right)^2 = \frac{a_1}{a_2}$$

Subscript 1 defines dimensions and coefficients with reference to the smaller diameter.
Subscript 2 refers to the larger diameter.

SUDDEN AND GRADUAL CONTRACTION



If: $\theta \leq 45°$ . . . . . K_2 = Formula 1
$45° < \theta \leq 180°$ . . . K_2 = Formula 2

SUDDEN AND GRADUAL ENLARGEMENT



If: $\theta \leq 45°$ . . . . . K_2 = Formula 3
$45° < \theta \leq 180°$ . . . K_2 = Formula 4

NOTE: K is based on use of schedule pipe as listed in Table 6e. In × 25.4 = mm.
SOURCE: Reference 4.

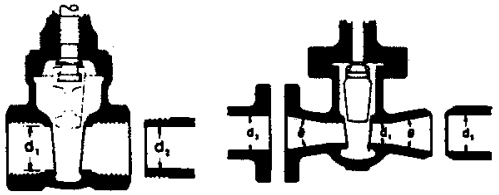
$$C_v = 1.156K_v$$

EXAMPLE 12 A pumping system consists of 20 ft (6.1 m) of 2-in (51-mm) suction pipe and 300 ft (91.5 m) of 1½-in (38-mm) discharge pipe, both Schedule 40 new steel. Also included are a bell mouth inlet, a 90° short radius (SR) suction elbow, a full port suction

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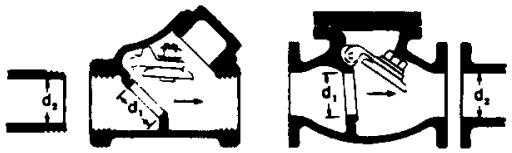
TABLE 6B Resistance coefficient K for valves and fittings

GATE VALVES
Wedge Disc, Double Disc, or Plug Type



If: $\beta = 1$, $\theta = 0$ . . . . . $K_1 = 8\,f_T$
$\beta < 1$ and $\theta \leq 45°$ . . . . . K_2 = Formula 5
$\beta < 1$ and $45° < \theta \leq 180°$ . . . K_2 = Formula 6

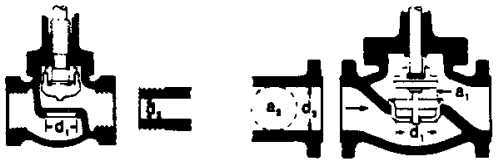
SWING CHECK VALVES



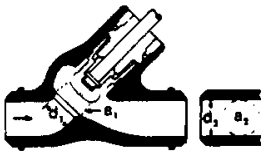
$K = 100\,f_T$
Minimum pipe velocity (fps) for full disc lift $= 35\sqrt{V}$

$K = 50\,f_T$
Minimum pipe velocity (fps) for full disc lift $= 48\sqrt{V}$

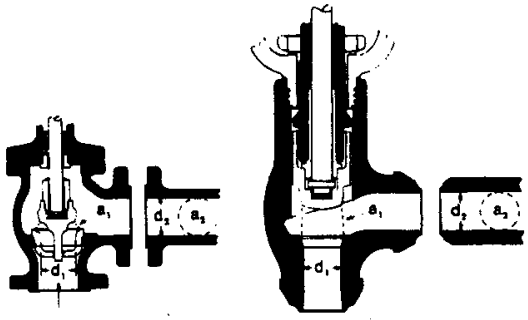
GLOBE AND ANGLE VALVES



If: $\beta = 1 \ldots K_1 = 340\,f_T$



If: $\beta = 1 \ldots K_1 = 55\,f_T$



If: $\beta = 1 \ldots K_1 = 150\,f_T$

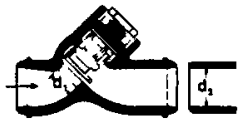
If: $\beta = 1 \ldots K_1 = 55\,f_T$

All globe and angle valves, whether reduced seat or throttled,
If: $\beta < 1 \ldots K_2$ = Formula 7

LIFT CHECK VALVES

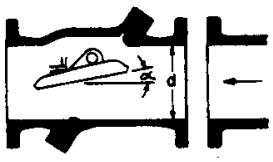


If: $\beta = 1 \ldots K_1 = 600\,f_T$
$\beta < 1 \ldots K_2$ = Formula 7
Minimum pipe velocity (fps) for full disc lift $= 40\,\beta^2\sqrt{V}$



If: $\beta = 1 \ldots K_1 = 55\,f_T$
$\beta < 1 \ldots K_2$ = Formula 7
Minimum pipe velocity (fps) for full disc lift $= 140\,\beta^2\sqrt{V}$

TILTING DISC CHECK VALVES



	$\alpha = 5°$	$\alpha = 15°$
Sizes 2 to 8" . . . $K =$	$40\,f_T$	$120\,f_T$
Sizes 10 to 14" . . . $K =$	$30\,f_T$	$90\,f_T$
Sizes 16 to 48" . . . $K =$	$20\,f_T$	$60\,f_T$
Minimum pipe velocity (fps) for full disc lift =	$80\sqrt{V}$	$30\sqrt{V}$

gate valve of Class 150 steel, a full port discharge gate valve of Class 400 steel, and a swing check valve. The valves and fittings are screw-connected and the same size as the connecting pipe.

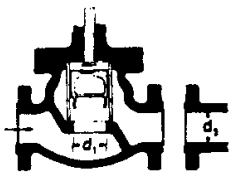
Determine the pipe, valve, and fitting losses when 60°F (15.6°C) oil having a specific gravity of 0.855 is pumped at a rate of 60 gpm (13.6 m³/h). Use resistance coefficients from Tables 5.

The inner diameter of the suction pipe is 2.067 in (52.5mm), and from Figure 32. $\epsilon/D = 0.00087$. From Eq. 9,

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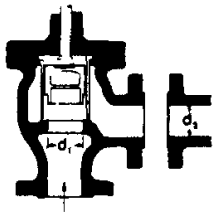
TABLE 6C Resistance coefficient K for valves and fittings

STOP-CHECK VALVES (Globe and Angle Types)



If:
$\beta = 1 \ldots K_1 = 400\, f_T$
$\beta < 1 \ldots K_2 =$ Formula 7

Minimum pipe velocity for full disc lift
$= 55\, \beta^2 \sqrt{V}$



If:
$\beta = 1 \ldots K_1 = 200\, f_T$
$\beta < 1 \ldots K_2 =$ Formula 7

Minimum pipe velocity for full disc lift
$= 75\, \beta^2 \sqrt{V}$

FOOT VALVES WITH STRAINER

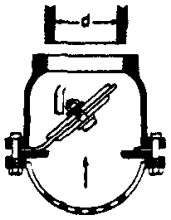
Poppet Disc



$K = 420 f_T$

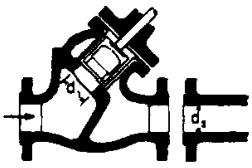
Minimum pipe velocity (fps) for full disc lift
$= 15 \sqrt{V}$

Hinged Disc

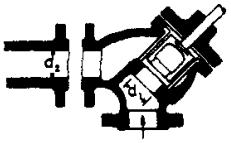


$K = 75 f_T$

Minimum pipe velocity (fps) for full disc lift
$= 35 \sqrt{V}$



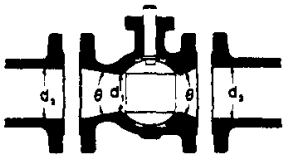
If:
$\beta = 1 \ldots K_1 = 300\, f_T$
$\beta < 1 \ldots K_2 =$ Formula 7



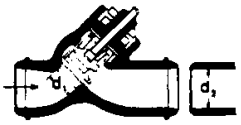
If:
$\beta = 1 \ldots K_1 = 350\, f_T$
$\beta < 1 \ldots K_2 =$ Formula 7

Minimum pipe velocity (fps) for full disc lift
$= 60\, \beta^2 \sqrt{V}$

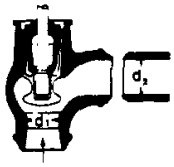
BALL VALVES



If: $\beta = 1, \theta = 0 \ldots K_1 = 3\, f_T$
$\beta < 1$ and $\theta \leq 45° \ldots K_2 =$ Formula 5
$\beta < 1$ and $45° < \theta \leq 180° \ldots K_2 =$ Formula 6



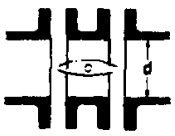
If:
$\beta = 1 \ldots K_1 = 55\, f_T$
$\beta < 1 \ldots K_2 =$ Formula 7



If:
$\beta = 1 \ldots K_1 = 55\, f_T$
$\beta < 1 \ldots K_2 =$ Formula 7

Minimum pipe velocity (fps) for full disc lift
$= 140\, \beta^2 \sqrt{V}$

BUTTERFLY VALVES



Sizes 2 to 8" ... $K = 45\, f_T$
Sizes 10 to 14" ... $K = 35\, f_T$
Sizes 16 to 24" ... $K = 25\, f_T$

In USCS units,
$$V = \frac{\text{gpm}}{(\text{pipe ID in inches})^2} \times 0.408 = \frac{60}{2.067^2} \times 0.408 = 5.73 \text{ ft}$$
$$VD'' = 5.73 \times 2.067 = 11.8 \text{ ft/s} \times \text{in}$$

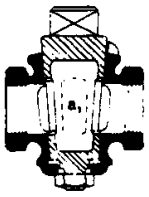
In SI units
$$V = \frac{\text{m}^3/\text{h}}{(\text{pipe ID in cm})^2} \times 3.54 = \frac{13.6}{5.25^2} \times 3.54 = 1.75 \text{ m/s}$$

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TABLE 6D Resistance coefficient K for valves and fittings

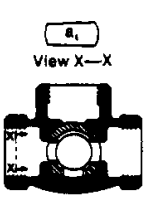
PLUG VALVES AND COCKS

Straight-Way

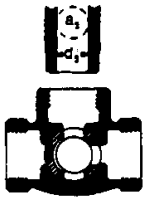


If: $\beta = 1$, $K_1 = 18\, f_T$

3-Way



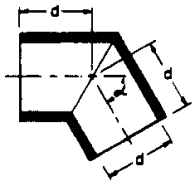
If: $\beta = 1$, $K_1 = 30\, f_T$



If: $\beta = 1$, $K_1 = 90\, f_T$

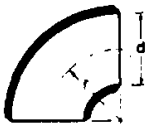
If: $\beta < 1 \ldots K_2$ = Formula 6

MITRE BENDS



α	K
0°	2 f_T
15°	4 f_T
30°	8 f_T
45°	15 f_T
60°	25 f_T
75°	40 f_T
90°	60 f_T

90° PIPE BENDS AND FLANGED OR BUTT-WELDING 90° ELBOWS



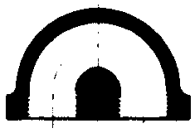
r/d	K	r/d	K
1	20 f_T	8	24 f_T
1.5	14 f_T	10	30 f_T
2	12 f_T	12	34 f_T
3	12 f_T	14	38 f_T
4	14 f_T	16	42 f_T
6	17 f_T	20	50 f_T

The resistance coefficient, K_B, for pipe bends other than 90° may be determined as follows:

$$K_B = (n - 1)\left(0.25\, \pi\, f_T \frac{r}{d} + 0.5\, K\right) + K$$

n = number of 90° bends
K = resistance coefficient for one 90° bend (per table)

CLOSE PATTERN RETURN BENDS



$K = 50\, f_T$

STANDARD ELBOWS

90°



$K = 30\, f_T$

45°



$K = 16\, f_T$

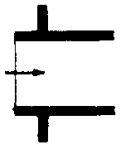
STANDARD TEES



Flow thru run . . . . . $K = 20\, f_T$
Flow thru branch . . . . $K = 60\, f_T$

PIPE ENTRANCE

Inward Projecting

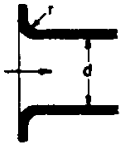


$K = 0.78$

r/d	K
0.00*	0.5
0.02	0.28
0.04	0.24
0.06	0.15
0.10	0.09
0.15 & up	0.04

*Sharp-edged

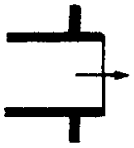
Flush



For K, see table

PIPE EXIT

Projecting



$K = 1.0$

Sharp-Edged



$K = 1.0$

Rounded



$K = 1.0$

$$VD = 1.75 \times 0.0525 = 0.0919 \text{ m/s} \times \text{m}$$

$$(VD'' = 0.0919 \times 129.2 = 11.8 \text{ ft/s} \times \text{in})$$

From Figure 33, $Re = 1 \times 10^4$, and from Figure 31, $f = 0.031$. From Eq. 16

in USCS units $$h_{fs} = 0.031 \frac{20 \times 12}{2.067} \times \frac{5.73^2}{2 \times 32.17} = 1.84 \text{ ft}$$

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TABLE 6E Pipe schedule for different classes of valves and fittings associated with K factors used in Tables 6A to 6D

Class	Schedule
300 and lower	40
400 and 600	80
900	120
1500	160
2500 (sizes $\frac{1}{2}$ to 6 in)[a]	XXS
2500 (sizes 8 in and up)[a]	160

[a]In × 25.4 = mm.
Source: Reference 4.

In SI units $$h_{fs} = 0.031 \frac{6.1}{0.0525} \times \frac{1.75^2}{2 \times 9.807} = 0.56 \text{ m}$$

The inner diameter of the discharge pipe is 1.610 in (40.89 mm), and from Figure 32, $e/D = 0.0011$. From Eq. 9

in USCS units $$V = \frac{60}{1.601^2} \times 0.408 = 9.44 \text{ ft/s}$$
$$VD'' = 9.44 \times 1.601 = 15.2 \text{ ft/s} \times \text{in}$$

in SI units $$V = \frac{13.6}{4.089^2} \times 3.54 = 2.88 \text{ m/s}$$
$$VD = 2.88 \times 0.04089 = 0.118 \text{ m/s} \times \text{m}$$
$$(VD'' = 0.118 \times 129.2 = 15.2 \text{ ft/s} \times \text{in})$$

From Figure 33, $Re = 1.5 \times 10^4$, and from Figure 31, $f = 0.030$. From Eq. 16

in USCS units $$h_{fd} = 0.030 \frac{300 \times 12}{1.610} \times \frac{9.44^2}{2 \times 32.17} = 9.29 \text{ ft}$$

in SI units $$h_{fd} = 0.030 \frac{91.5}{0.04089} \times \frac{2.88^2}{2 \times 9.807} = 28.39 \text{ m}$$

The valve and fitting losses from Tables 5 and Eq. 20 are 2-in (51-mm) bellmouth, $K = 0.05$:

In USCS units $$h_{f1} = 0.05 \frac{5.73^2}{2 \times 32.17} = 0.026 \text{ ft}$$

In SI units $$h_{f1} = 0.05 \frac{1.75^2}{2 \times 9.807} = 0.0078 \text{ m}$$

2-in (51-mm) SR 90° elbow, $K = 0.95 \pm 30\%$

In USCS units $$h_{f2} = 0.95 \frac{5.73^2}{2 \times 32.17} = 0.48 \pm 0.14 \text{ ft}$$

In SI units $$h_{f2} = 0.95 \frac{1.75^2}{2 \times 9.807} = 0.15 \pm 0.044 \text{ m}$$

2-in (51-mm) gate valve, $K = 0.16 \pm 25\%$

In USCS units $$h_{f3} = 0.16 \frac{5.73^2}{2 \times 32.17} = 0.082 \pm 0.021 \text{ ft}$$

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In SI units $h_{f3} = 0.16 \dfrac{1.75^2}{2 \times 9.807} = 0.0250 \pm 0.0062$ m

$1\frac{1}{2}$-in (38-mm) gate valve, $K = 0.19 \pm 25\%$

In USCS units $h_{f4} = 0.19 \dfrac{9.44^2}{2 \times 32.17} = 2.63 \pm 0.066$ ft

In SI units $h_{f4} = 0.19 \dfrac{2.88^2}{2 \times 9.807} = 0.0803 \pm 0.02$ m

$1\frac{1}{2}$-in (38-mm) swing check valve, $K = 2.5 \pm 30\%$

In USCS units $h_{f5} = 2.5 \dfrac{9.44^2}{2 \times 32.17} = 3.46 \pm 1.0$ ft

In SI units $h_{f5} = 2.5 \dfrac{2.88^2}{2 \times 9.807} = 1.06 \pm 0.32$ m

The total pipe, valve, and fitting losses are

In USCS units
$$\Sigma h_f = h_{fs} + h_{fd} + h_{f1} + h_{f2} + h_{f3} + h_{f4} + h_{f5}$$
$$= 1.84 + 92.9 + 0.026 + 0.48 + 0.082 + 0.263 + 3.46 = 99.05 \text{ ft}$$
$$\text{Total variation} = \pm(0.14 + 0.021 + 0.066 + 1.0) = \pm 1.23 \text{ ft}$$

In SI units
$$\Sigma h_f = h_{fs} + h_{fd} + h_{f1} + h_{f2} + h_{f3} + h_{f4} + h_{f5}$$
$$= 0.56 + 28.39 + 0.0078 + 0.044 + 0.0250 + 0.0803 + 1.06 = 30.17 \text{ m}$$
$$\text{Total variation} = \pm(0.044 + 0.0062 + 0.02 + 0.32) = \pm 0.39 \text{ m}$$

EXAMPLE 13 Solve Example 12 using resistance coefficients from Tables 6.

Suction pipe:

In USCS units $h_{fs} = 1.84$ ft (same as in Example 12)

In SI units $h_{fs} = 0.56$ m (same as in Example 12)

Discharge pipe,

In USCS units $h_{fd} = 92.9$ ft (same as in Example 12)

In SI units $h_{fd} = 28.39$ m (same as in Example 12)

Valve and fitting losses from Tables 6 and Eq. 20: 2-in (51-mm) bellmouth, $K = 0.04$

In USCS units $h_{f1} = 0.04 \dfrac{5.73^2}{2 \times 32.17} = 0.020$ ft

In SI units $h_{f1} = 0.04 \dfrac{1.75^2}{2 \times 9.807} = 0.0062$ m

2-in (51-mm) SR 90° elbow, $K = 30 f_T$

$f_T = 0.019$ (from Table 6A)

$K = 30 \times 0.019 = 0.57$

In USCS units $h_{f2} = 0.57 \dfrac{5.73^2}{2 \times 32.17} = 0.29$ ft

In SI units $h_{f2} = 0.57 \dfrac{1.57^2}{2 \times 9.807} = 0.089$ m

2-in (51-mm) gate valve, $\beta = 1, \theta = 0, k = 8f_T$ $\left(\beta = \dfrac{d_1}{d_2} = 1 \text{ from Table 6A}\right)$

$K = 8 \times 0.019 = 0.15$

In USCS units $h_{f3} = 0.15 \dfrac{5.73^2}{2 \times 32.17} = 0.077$ ft

In SI units $h_{f3} = 0.15 \dfrac{1.75^2}{2 \times 9.807} = 0.023$ m

1-in (38-mm) gate valve, $\beta = 1, \theta = 0, K = 8f_T$ for Schedule 80 (from Table 6E)

$K = 8f_T (1.10/1.500)^4 = 10.62f_T$ for Schedule 40

$f_T = 0.021$ (from Table 6A)

$K = 10.62 \times 0.021 = 0.22$

In USCS units $h_{f4} = 0.22 \dfrac{9.44^2}{2 \times 32.17} = 0.30$ ft

In SI units $h_{f4} = 0.22 \dfrac{2.88^2}{2 \times 9.807} = 0.093$ m

$1\frac{1}{2}$-in (38-mm) swing check valve, $K = 100f_T$ (from Table 6B)

Minimum pipe velocity for full disk lift $= 35\sqrt{V} = 35\sqrt{0.0189} = 4.81$ ft/s < 9.44 ft/s

$K = 100 \times 0.021 = 2.1$

In USCS units $h_{f5} = 2.1 \dfrac{9.44^2}{2 \times 32.17} = 2.91$ ft

In SI units $h_{f5} = 2.1 \dfrac{2.88^2}{2 \times 9.807} = 0.89$ m

Total pipe, valve, and fitting losses:

In USCS units
$$\begin{aligned} \Sigma h_f &= h_{fs} + h_{fd} + h_{f1} + h_{f2} + h_{f3} + h_{f4} + h_{f5} \\ &= 1.84 + 92.9 + 0.020 + 0.29 + 0.077 + 0.30 + 2.91 \\ &= 98.34 \text{ ft} \end{aligned}$$

In SI units
$$\begin{aligned} \Sigma h_f &= h_{fs} + h_{fd} + h_{f1} + h_{f2} + h_{f3} + h_{f4} + h_{f5} \\ &= 0.56 + 28.39 + 0.062 + 0.089 + 0.023 + 0.093 + 0.89 \\ &= 30.05 \text{ m} \end{aligned}$$

INCREASERS The head lost when there is a sudden increase in pipe diameter, with velocity changing from V_1 to V_2 in the direction of flow, can be calculated analytically. Computed results have been confirmed experimentally to be true to within ±3%. The head loss is expressed as shown, with K computed to be equal to unity:

$$h = K \frac{(V_1 - V_2)^2}{2g} = K\left[1 - \left(\frac{D_1}{D_2}\right)^2\right]^2 \frac{V_1^2}{2g} = K\left[\left(\frac{D_2}{D_1}\right)^2 - 1\right]^2 \frac{V_2^2}{2g} \tag{23}$$

The value of K is also approximately equal to unity if a pipe discharges into a relatively large reservoir. This indicates that all the kinetic energy $V_1^2/2g$ is lost and V_2 equals zero.

The loss of head for a gradual increase in pipe diameter when the flow is through a diffuser can be found from Figure 39 and Table 6A. The diffuser converts some of the kinetic

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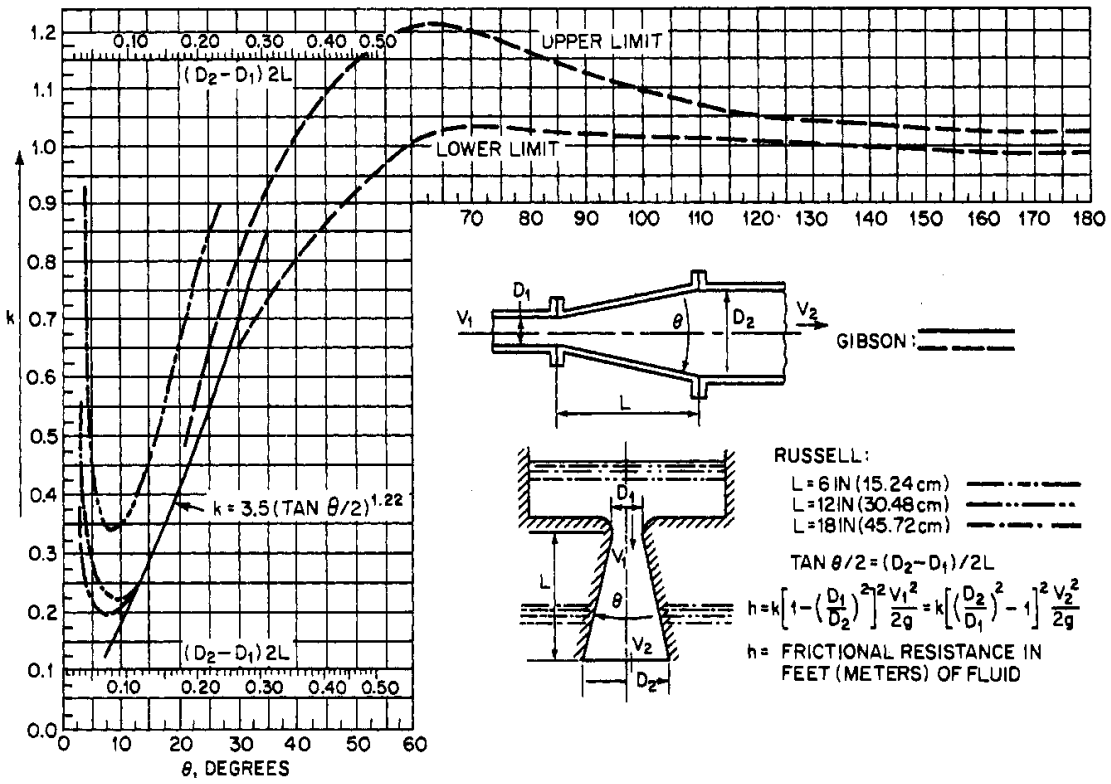


FIGURE 39 Resistance coefficients for increasers and diffusers (D = in or ft [m]; V = ft/s [m/s]) (Hydraulic Institute Engineering Data Book, Reference 5)

energy to pressure. Values for the coefficient used with Eq. 23 for calculating head loss are shown in Figure 39. The optimum total angle appears to be 7.5°. Angles greater than this result in shorter diffusers and less friction, but separation and turbulence occur. For angles greater than 50°, it is preferable to use a sudden enlargement.

REDUCERS Figure 40 and Table 6A give values of the resistance coefficient to be used for sudden reducers.

BENDS Figure 41 may be used to determine the resistance coefficient for 90° pipe bends of uniform diameter. Figure 42 gives resistance coefficients for bends that are less than 90° and can be used for surfaces having moderate roughness such as clean steel and cast iron. Figures 41 and 42 are not recommended for elbows with R/D below 1. Tables 6D and 7 give values of resistance coefficients for miter bends.

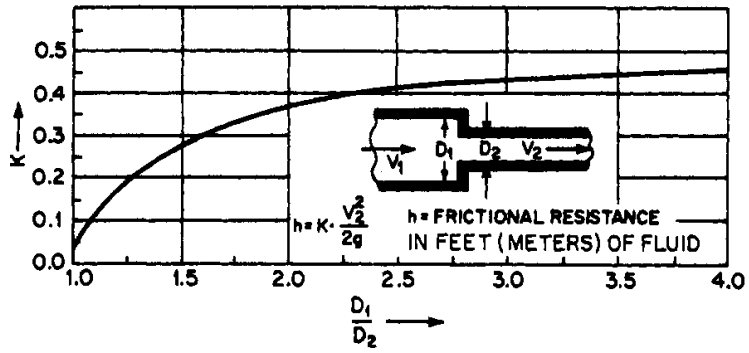


FIGURE 40 Resistance coefficients for reducers (Hydraulic Institute Engineering Data Book, Reference 5)

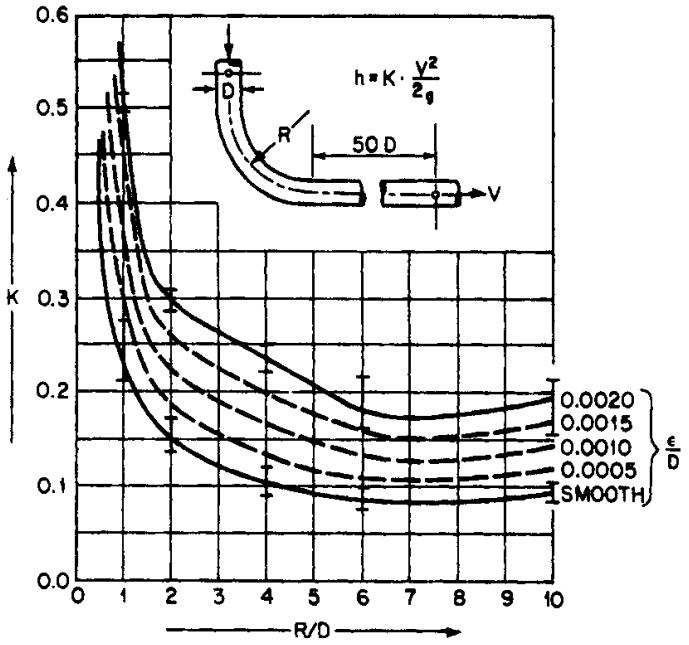


FIGURE 41 Resistance coefficients for 90° bends of uniform diameter (Hydraulic Institute Engineering Data Book, Reference 5)

PUMP SUCTION ELBOWS Figures 43 and 44 illustrate two typical rectangular to round reducing suction elbows. Elbows of this configuration are sometimes used under dry-pit vertical volute pumps. These elbows are formed in concrete and are designed to require a minimum height, thus permitting a higher pump setting with reduced excavation. Figure 43 shows a long-radius elbow, and Figure 44 a short-radius elbow. The resulting velocity distribution into the impeller eye and the loss of head are shown for these two designs.

METERS Orifice, nozzle, and venturi meters (Figures 45–47) are used to measure rate of flow. These meters, however, introduce additional loss of head into the pumping system. Each of these meters is designed to create a pressure differential through the primary element. The magnitude of the pressure differential depends on the velocity and density of the liquid and the design of the element. The primary element restricts the area of flow, increases the velocity, and decreases the pressure. An expanding section following the

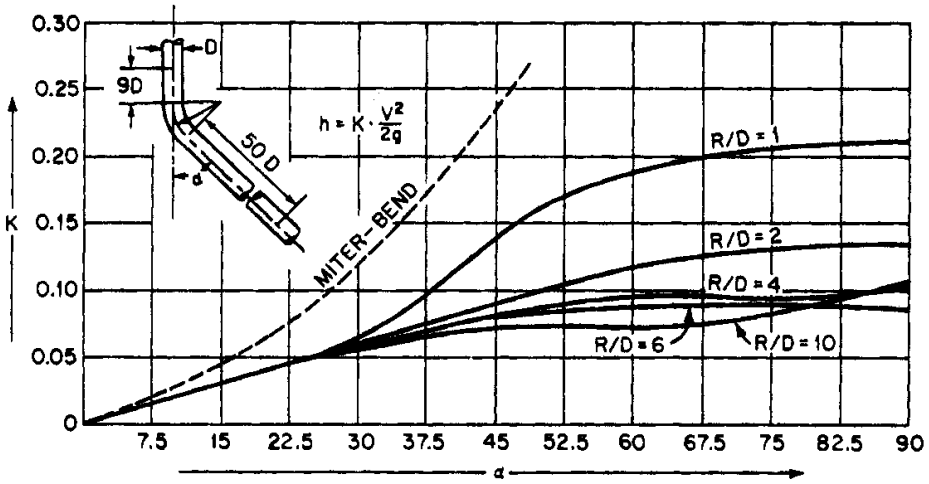
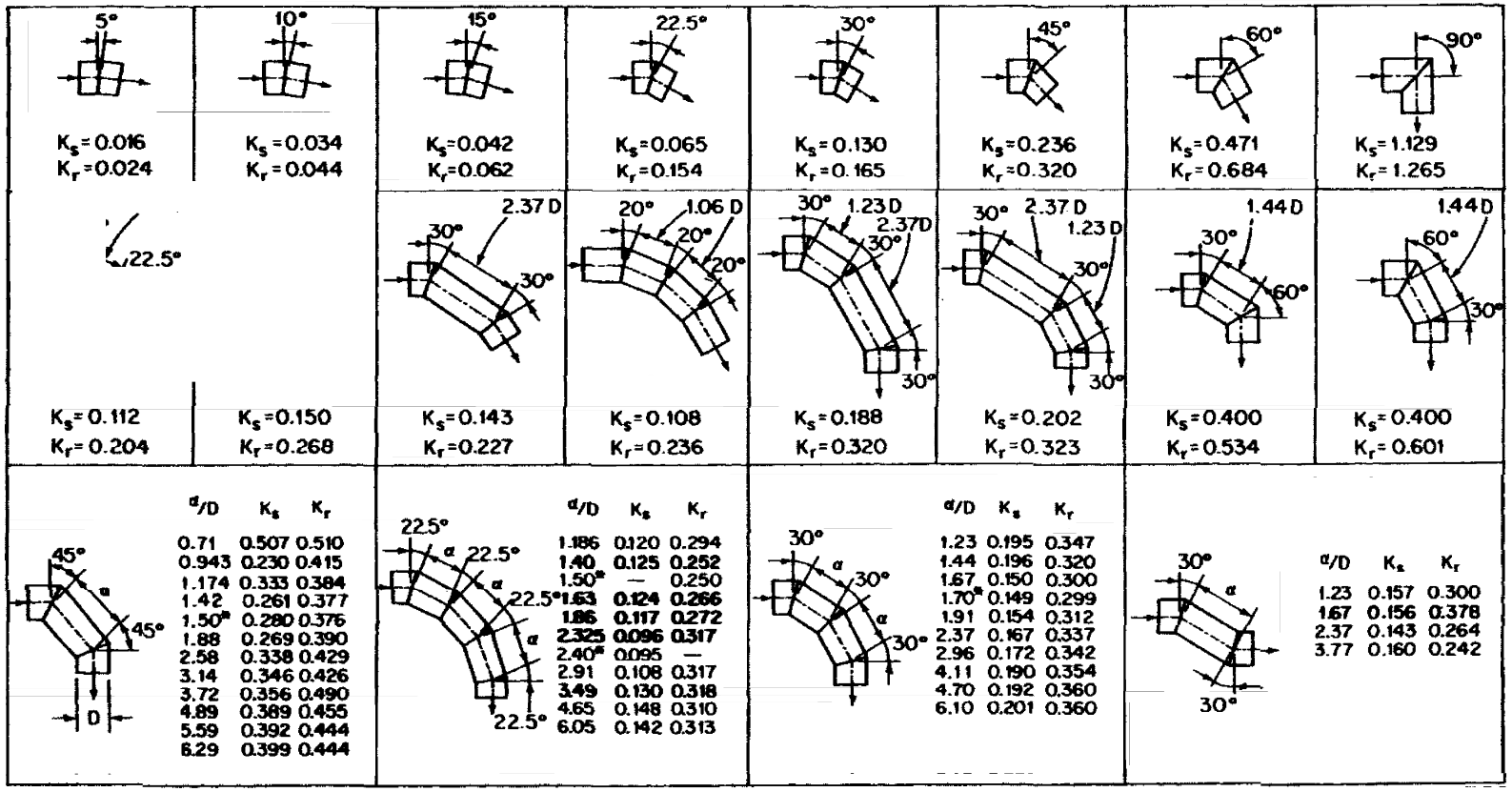


FIGURE 42 Resistance coefficients for bends of uniform diameter and smooth surface at Reynolds number $\approx 2.25 \times 10^5$ (Hydraulic Institute Engineering Data Book, Reference 5)

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TABLE 7 Resistance coefficients for miter bends at reynolds number $\approx 2.25 \times 10^5$



d/D	K_s	K_r
0.71	0.507	0.510
0.943	0.230	0.415
1.174	0.333	0.384
1.42	0.261	0.377
1.50*	0.280	0.376
1.88	0.269	0.390
2.58	0.338	0.429
3.14	0.346	0.426
3.72	0.356	0.490
4.89	0.389	0.455
5.59	0.392	0.444
6.29	0.399	0.444

d/D	K_s	K_r
1.186	0.120	0.294
1.40	0.125	0.252
1.50*	—	0.250
1.53	0.124	0.266
1.86	0.117	0.272
2.325	0.096	0.317
2.40*	0.095	—
2.91	0.108	0.317
3.49	0.130	0.318
4.65	0.148	0.310
6.05	0.142	0.313

d/D	K_s	K_r
1.23	0.195	0.347
1.44	0.196	0.320
1.67	0.150	0.300
1.70*	0.149	0.299
1.91	0.154	0.312
2.37	0.167	0.337
2.96	0.172	0.342
4.11	0.190	0.354
4.70	0.192	0.360
6.10	0.201	0.360

d/D	K_s	K_r
1.23	0.157	0.300
1.67	0.156	0.378
2.37	0.143	0.264
3.77	0.160	0.242

K_s = RESISTANCE COEFFICIENT FOR SMOOTH SURFACE
K_r = RESISTANCE COEFFICIENT FOR ROUGH SURFACE, $\frac{\epsilon}{D} = 0.0022$

*OPTIMUM VALUE OF α INTERPOLATED

Source: Hydraulic Institute Engineering Data Book, Reference 5

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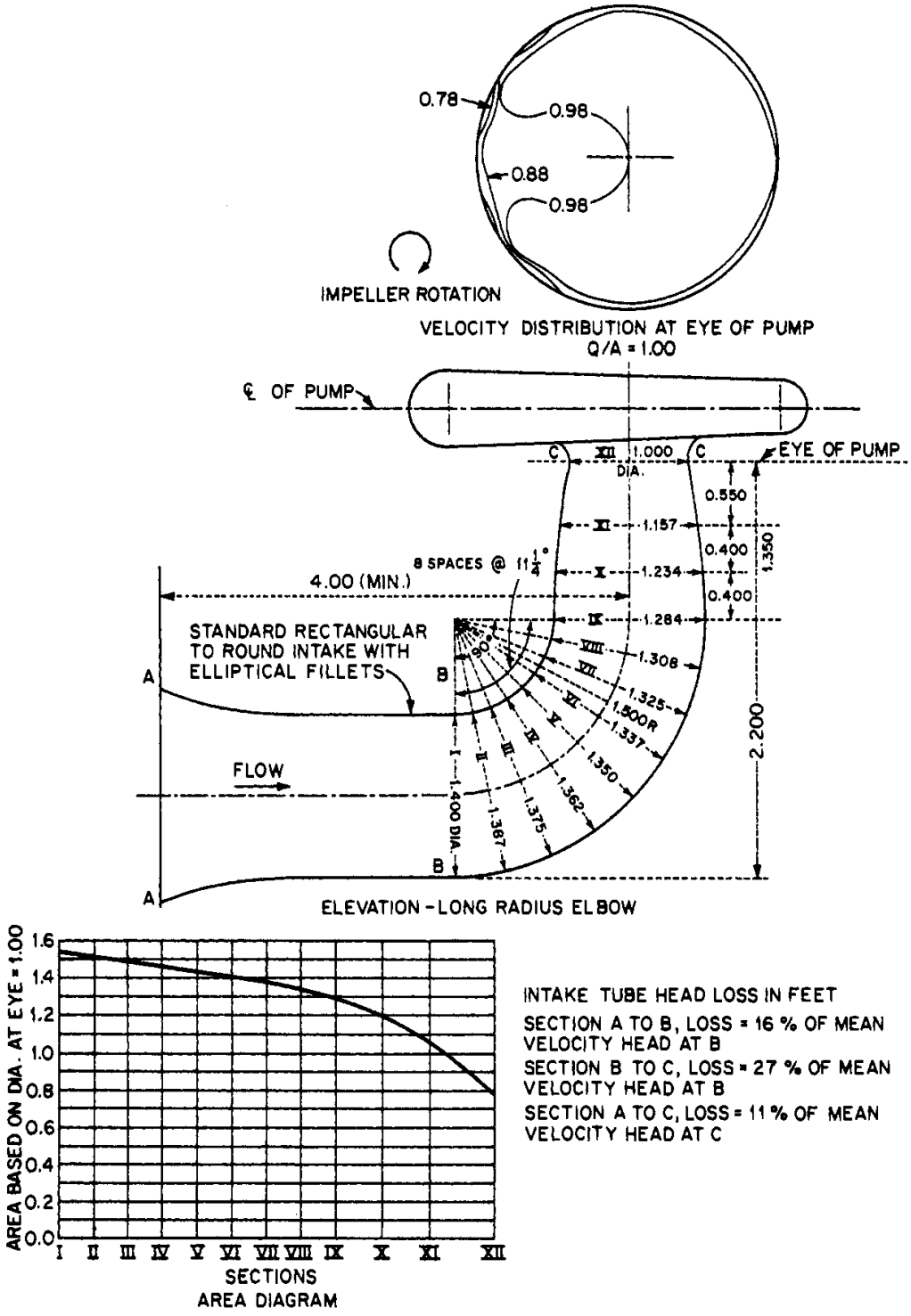


FIGURE 43 Head loss in a long-radius pump suction elbow (in × 25.4 = mm) (Reference 16)

primary element provides pressure head recovery and determines the meter efficiency. The pressure differential between inlet and throat taps measure rate of flow; the pressure differential between inlet and outlet taps measures the meter head loss (an outlet tap is not usually provided). Of the three types, venturi meters offer the least resistance to flow, and orifice meters the most.

When meters are designed and pressure taps located as recommended,[6] Figures 48–50 may be used to estimate the overall pressure loss. In these figures, the loss of pressure is expressed as a percentage of the differential pressure measured at the appropriate taps.

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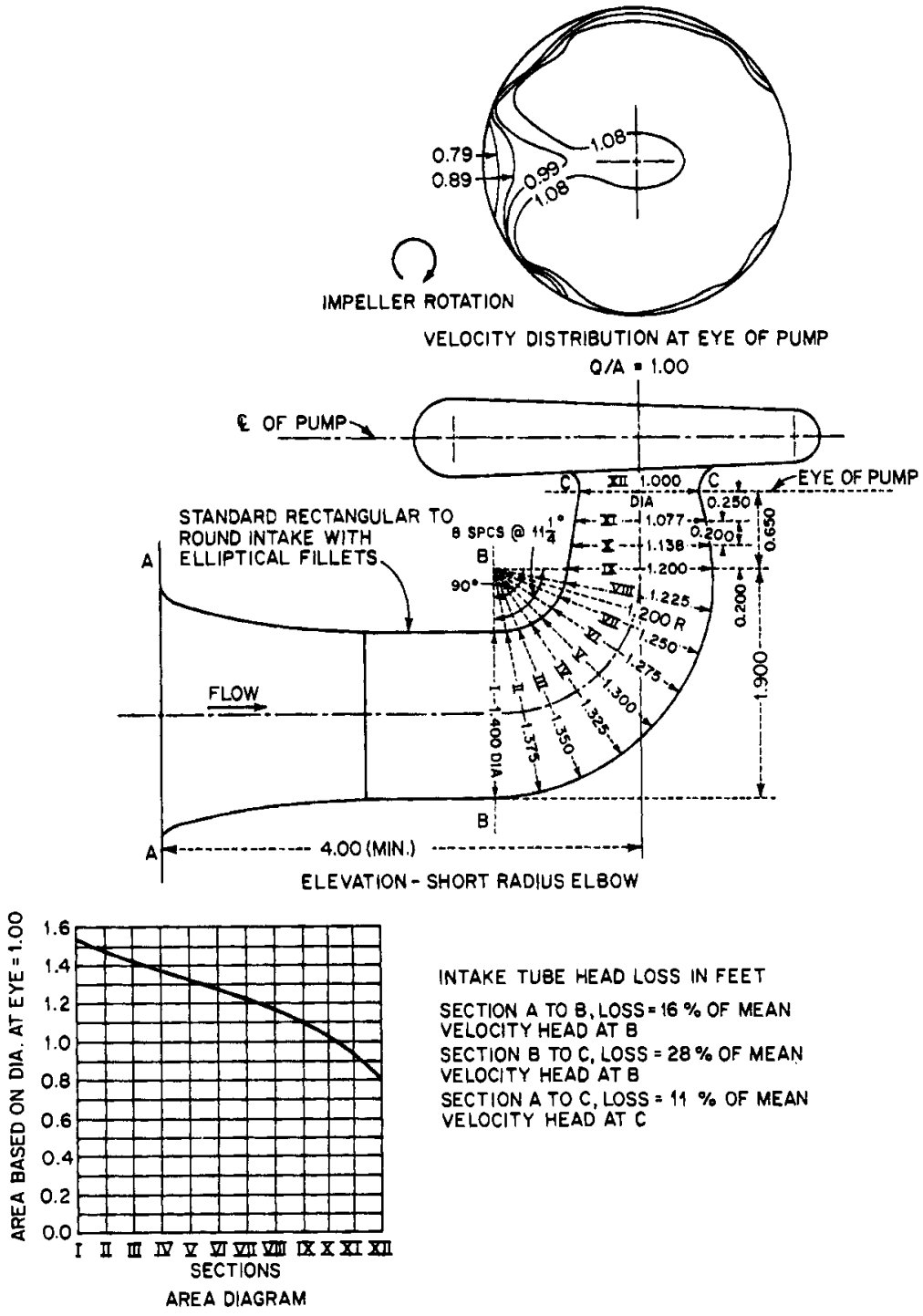


FIGURE 44 Head loss in a short-radius pump suction elbow (in × 25.4 = mm) (Reference 16)

and values are given for various sizes of meters. This loss of pressure is also the meter total head, or energy loss, because there is no change in velocity head if the pipe inside diameters are the same at the various measuring points. The meter loss of head should be in units of feet (meters) of liquid pumped if other system losses are expressed this way. Reference 6 should be consulted for information concerning formulas and coefficients for calculating differential pressure versus rate of flow.

Screens, Perforated Plates, and Bar Racks Obstructions to the flow of liquid in the form of multiple orifices uniformly distributed across an open or closed conduit may be used to remove solids, throttle flow, and produce or reduce turbulence. They may be used

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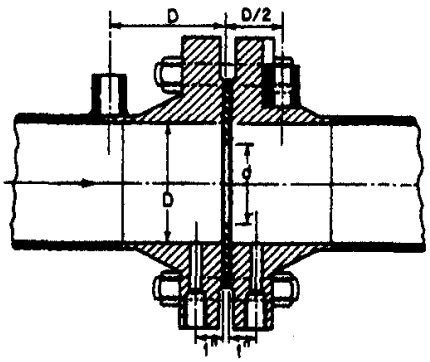


FIGURE 45 Thin-plate, square-edged orifice meter, showing alternate locations of pressure taps (Reference 6)

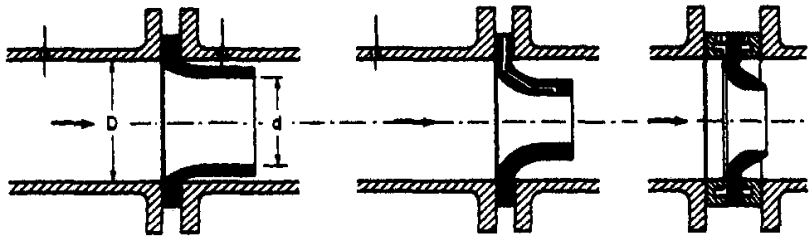


FIGURE 46 Shapes of flow nozzle meters and locations of pressure taps (Reference 6)

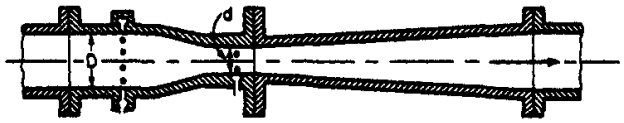


FIGURE 47 Herschel-type venturi meter, showing locations of pressure taps (Reference 6)

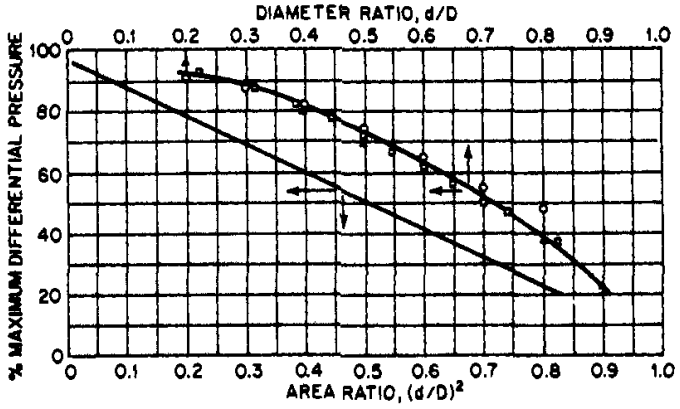


FIGURE 48 *Overall pressure loss across thin-plate orifices (Reference 6)*

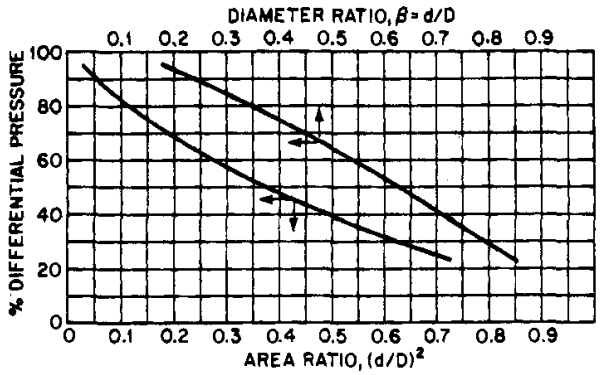


FIGURE 49 Overall pressure loss across flow nozzles (Reference 6)

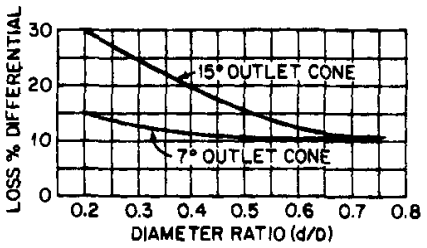


FIGURE 50 Overall pressure loss across venturi tubes (Reference 6)

upstream or downstream from a pump, depending on their purpose, and they therefore introduce a loss of head that must be accounted for. When an obstruction is placed upstream from a pump, a significant reduction in suction pressure and NPSH available can result.

The loss of head results from an increase in velocity at the entrance to the openings, friction, and the sudden decrease in velocity following the expansion of the numerous liquid jets. The total head loss is a function of the ratio of the total area of the openings to the area of the conduit before the obstruction, the thickness of the obstruction, the Reynolds number, and the velocities. Various investigators have determined values for resistance coefficients that can be multiplied by the approach velocity head to obtain the loss through these obstructions. According to Idel'chik,[7] loss of head h in feet (meters) may be calculated from the equation

$$h = K_1 \frac{V_1^2}{2g} \tag{24}$$

where K_1 = resistance coefficient
V_1 = average velocity in the conduit approaching the obstruction, ft/s (m/s)
g = acceleration of gravity, 32.17 ft/s² (9.807 m/s²)

ROUND-WIRE MESH SCREENS For flow having Reynolds numbers equal to or greater than 400, the resistance coefficient for flow through a round-wire, plain square mesh screen (Figure 51a) may be estimated as a function of percentage of open area using the equations

in USCS units
$$Re = \frac{V_o W_d}{\upsilon 12} \geq 400 \tag{25a}$$

in SI units
$$Re = \frac{V_o W_d}{\upsilon 1000} \geq 400 \tag{25b}$$

$$K_1 = k_0\left(1 - \frac{A_r}{100}\right) + \left(\frac{100}{A_r} - 1\right)^2 \tag{26}$$

$$A_r = 100\left(\frac{S}{S + W_d}\right)^2 \tag{27}$$

$$A_r = 100(1 - MW_d)^2 \tag{28}$$

where Re = screen Reynolds number referred to wire diameter
V_o = velocity through area of rectangular opening = $100V_1/A_r$, ft/s (m/s)

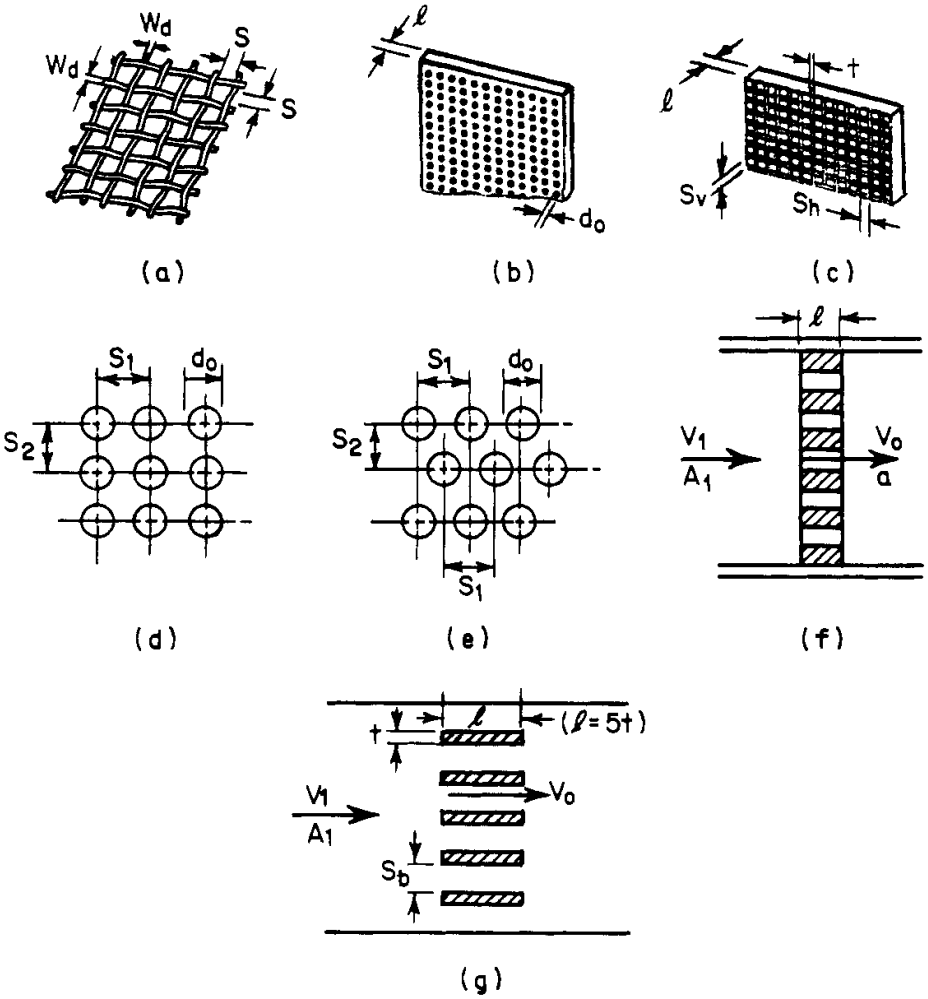


FIGURE 51 Explanation of terms used in Eq. 25–28 and 30–37 and Tables 8 and 9 for calculating resistance coefficient K_1 (a) round-wire, plain square, mesh screen, (b) round-hole perforated plate, (c) rectangular-grid perforated plate, (d) perforated plate, holes in vertical columns, (e) perforated plate, staggered holes, (f) perforated plate, cross section, (g) rectangular bar rack, cross section

W_d = screen wire diameter, in (mm)
v = kinematic viscosity, ft²/s (m²/s)
k_0 = 1.0 for new, perfectly clean screens to 1.3 for normal screens
A_r = percentage of open area
S = square space between wires, in (mm)
M = mesh of screen or number of wires per in (mm)

Using water in a flow range of Reynolds numbers of approximately 60 to 800, Padmanabhan and Vigander[8] experimented with 1.3- to 12.0-mesh screens, 51.4 to 56% open area, and found their results to be comparable with those of other investigators. Values of the coefficient of resistance K_1, from the Padmanabhan-Vigander tests and others, vary from 2 decreasing asymptotically to 1 with increase in Reynolds number (Eq. 25) up to approximately 1000 for 47 to 56% open area. Smaller percentage open area values have larger coefficients.

Armour and Cannon,[9] using nitrogen as the fluid, tested various types of fine woven wire screens and derived equations for pressure drop in terms of a friction factor and the screen Reynolds number. Approach velocities ranged from 0.1 to 30 ft/s (0.03 to 9 m/s),

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TABLE 8 Plain square screen geometry

Screen mesh size M, in^{-1} (mm^{-1})	Wire diameter W_d, in $\times 10^{-4}$ (mm $\times 10^{-4}$)	Open area A_r, %	Screen constant C_1, $ft^{-1} \times 10^3$ ($m^{-1} \times 10^3$)	Screen constant C_2
30 × 30 (1.18 × 1.18)	9.45 (240)	94.4	2.553 (8.376)	1.553
150 × 150 (5.91 × 5.91)	2.36 (59.9)	93.0	8.267 (27.12)	2.594
250 × 250 (9.84 × 9.84)	1.69 (42.9)	91.7	16.21 (53.18)	2.934
400 × 400 (15.7 × 15.7)	1.00 (25.4)	92.2	30.68 (100.7)	2.680

Source: Reference 9.

resulting in Reynolds numbers from 350 to 275,000. To simply calculations, screen geometry constants C_1 = (surface area to unit volume ratio)^2 × (pore diameter) and C_2 = (screen thickness) ÷ (void fraction)^2 × (pore diameter) have been added to the reference authors' equations to obtain the following expression for screen head loss h in feet (meters):

$$h = C_2 \frac{V_1}{g}(8.61\upsilon C_1 + 0.52V_1) \tag{29}$$

where V_1 = average velocity in the conduit approaching the screen, ft/s (m/s)

Table 8 lists C_1 and C_2 values for a sample of plain square screens tested by Armour and Cannon.

PERFORATED PLATES AND BAR RACKS For flow having Reynolds numbers equal to or greater than 10^5, the resistance coefficients for flow through thick, square-edge perforated plates with round (Figure 51b) or rectangular (Figure 51c) openings and racks with rectangular cross-section bars (length = 5 times thickness; Figure 51g) may be calculated using Eq. 24, Table 9, and the following equations:

$$Re = \frac{V_o D_h}{\upsilon} \geq 10^5 \tag{30}$$

In USCS units

$$D_h = \frac{a}{3p} \quad \text{or} \quad D_h = \frac{d_o}{12} \tag{31a}$$

In SI units

$$D_h = 0.004\frac{a}{\rho} \quad \text{or} \quad D_h = \frac{d_o}{1000} \tag{31b}$$

for plates with round holes

$$A_r = 100\left(\frac{0.785 d_o^2}{S_1 S_2}\right) \tag{32}$$

for plates with single hole in center

$$A_r = 100\left(\frac{d_o}{d_1}\right)^2 \tag{33}$$

for plates with square openings ($S = S_h = S_v$)

$$A_r = 100\left(\frac{S}{S + t}\right)^2 \tag{34}$$

for plates with rectangular openings

$$A_r = 100\left[\frac{S_h S_v}{(S_h + t)(S_v + t)}\right] \tag{35}$$

for any plate or bar rack

$$A_r = 100\frac{A_o}{A_1} \tag{36}$$

for single vertical or horizontal bar racks

$$A_r = 100\left(\frac{S_b}{S_b + t}\right) \tag{37}$$

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TABLE 9 Values of resistance coefficient K_1 for perforated plates and bar racks

	$A_r/100$ *or* A_o/A_1															
l/Dh	0.02	0.04	0.06	0.08	0.10	0.15	0.20	0.25	0.30	0.40	0.50	0.60	0.70	0.80	0.90	1.0
0	7000	1670	730	400	245	96.0	51.5	30.0	18.2	8.25	4.00	2.00	0.97	0.42	0.13	0
0.2	6600	1600	687	374	230	94.0	48.0	28.0	17.4	7.70	3.75	1.87	0.91	0.40	0.13	0.01
0.4	6310	1530	660	356	221	89.0	46.0	26.5	16.6	7.40	3.60	1.80	0.88	0.39	0.13	0.01
0.6	5700	1380	590	322	199	81.0	42.0	24.0	15.0	6.60	3.20	1.60	0.80	0.36	0.13	0.01
0.8	4680	1130	486	264	164	66.0	34.0	19.6	12.2	5.50	2.70	1.34	0.66	0.31	0.12	0.02
1.0	4260	1030	443	240	149	60.0	31.0	17.8	11.1	5.00	2.40	1.20	0.61	0.29	0.11	0.02
1.4	3930	950	408	221	137	55.6	28.4	16.4	10.3	4.60	2.25	1.15	0.58	0.28	0.11	0.03
2.0	3770	910	391	212	134	53.0	27.4	15.8	9.90	4.40	2.20	1.13	0.58	0.28	0.12	0.04
3.0	3765	913	392	214	132	53.5	27.5	15.9	10.0	4.50	2.24	1.17	0.61	0.31	0.15	0.06
4.0	3775	930	400	215	132	53.8	27.7	16.2	10.0	4.60	2.25	1.20	0.64	0.35	0.16	0.08
5.0	3850	936	400	220	133	55.5	28.5	16.5	10.5	4.75	2.40	1.28	0.69	0.37	0.19	0.10
6.0	3870	940	400	222	133	55.8	28.5	16.6	10.5	4.80	2.42	1.32	0.70	0.40	0.21	0.12
7.0	4000	950	405	230	135	55.9	29.0	17.0	10.9	5.00	2.50	1.38	0.74	0.43	0.23	0.14
8.0	4000	965	410	236	137	56.0	30.0	17.2	11.1	5.10	2.58	1.45	0.80	0.45	0.25	0.16
9.0	4080	985	420	240	140	57.0	30.0	17.4	11.4	5.30	2.62	1.50	0.82	0.50	0.28	0.18
10	4110	1000	430	245	146	59.7	31.0	18.2	11.5	5.40	2.80	1.57	0.89	0.53	0.32	0.20

Note: l = thickness of perforated plate or length of bars, ft (m); D_h, A_r, A_o, and A_1 are as defined following Eq. 37.
Source: Reference 7.

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where Re = Reynolds number referred to hydraulic diameter
V_O = velocity through area of opening, ft/s (m/s)
D_h = hydraulic diameter (= diameter if openings are round holes), ft (m)
v = kinematic viscosity, ft²/s (m²/s)
a = area of single opening, in² (mm²)
p = perimeter of single opening, in (mm)
d_O = diameter of hole, in (mm)
A_r = percentage of open area
S_1 = horizontal spacing of holes, in (mm)
S_2 = vertical spacing of holes, in (mm)
d_1 = diameter of approach, in (mm)
S_h = horizontal apace between vertical bars, in (mm)
S_v = vertical space between horizontal bars, in (mm)
t = thickness of plate laths or bars, in (mm)
A_O = total area of openings, ft² (m²)
A_1 = total area of approach, ft² (m²)
S_b = space between single vertical or horizontal bars, in (mm)

The loss of head through perforated plates may also be calculated by using an *orifice coefficient C,* as suggested by Smith and Van Winkle[10] and Kolodzie and Van Winkle.[11] Test results using air and other gases with equilateral-triangle pitch perforated plates are shown in Table 10 for Reynolds numbers 400 to 20,000. Using single-orifice relations, the following expression equates flow rate to pressure drop:

$$\omega = CA_o \sqrt{\frac{2g\Delta P}{\gamma\left[1 - \left(\frac{A_r}{100}\right)^2\right]}} \tag{38}$$

where ω = flow rate, ft³/s (m³/s)
C = orifice coefficient from Table 10
A_O = total area of openings, ft² (m²)
g = acceleration of gravity, 32.17 ft/s² (9.807 m/s²)
ΔP = pressure drop, lb/ft² (N/m²)
γ = specific weight (force) of liquid, lb/ft³ (N/m³)
A_r = percentage of open area

The resistance coefficient K_1 and the orifice coefficient C may be interchanged in Eqs. 24 and 38 for loss through perforated plates:

$$C = \sqrt{\frac{1 - \left(\frac{A_r}{100}\right)^2}{K_1 - \left(\frac{A_r}{100}\right)^2}} \tag{39}$$

$$K_1 = \frac{1 - \left(\frac{A_r}{100}\right)^2}{C^2\left(\frac{A_r}{100}\right)^2} \tag{40}$$

TABLE 10 Orifice coefficients C for equilateral-triangle pitch perforated plates

		l/D_h								
Pitch/D_h	Re	0.33	0.43	0.50	0.52	0.65	0.75	0.80	1.0	2.0
2.0	400–4000	0.74–0.69	0.75–0.71	0.76–0.74	—	0.79–0.85	0.82–0.86	0.83–0.87	0.84–0.89	0.77–0.92
2.0	4000–20,000	0.69	0.71	0.74	0.77	0.85	0.86	0.87	0.89	0.92
3.0	400–4000	0.70–0.67	0.72–0.68	0.73–0.72	—	0.75–0.81	0.76–0.82	0.77–0.84	0.78–0.85	0.70–0.87
3.0	4000–20,000	0.67	0.68	0.72	0.74	0.81	0.82	0.84	0.85	0.87
4.0	400–4000	0.71–0.66	0.72–0.67	0.72–0.69	—	0.73–0.77	0.74–0.79	0.75–0.81	0.76–0.53	0.69–0.84
4.0	4000–20,000	0.66	0.67	0.69	0.72	0.77	0.79	0.81	0.88	0.84
5.0	400–4000	0.68–0.65	0.69–0.66	0.7–0.68	—	0.72–0.76	0.73–0.77	0.74–0.78	0.75–0.81	0.65–0.82
5.0	4000–20,000	0.65	0.66	0.68	0.71	0.76	0.77	0.78	0.81	0.82

Note: Pitch/D_h = pitch-to-hole-diameter ratio; l/D_h = plate-thickness-to-hole diameter ratio; Re = Reynolds number
Source: References 10 and 11.

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Throttling Orifices In addition to measuring flow, orifices can be used to (a) reduce flow by adding artificial resistance to increase system head, (b) dissipate energy to provide a desired pressure reduction, and (c) create a high-velocity jet. Orifices for these purposes are called *throttling orifices.* In order to maintain a desired minimum flow to prevent damage to a centrifugal pump, a throttling orifice, or a series of throttling orifices, can be used in the bypass system. The orifice provides additional bypass resistance to maintain the required bypass flow.

A throttling orifice can be fabricated by drilling a hole in a metal plate (or through bar stock) that, when inserted between flanges in a pipe (or threaded to pipe), will create the desired loss of head at the design flow. The resistance coefficient K_1 may be calculated as if the throttling device were a single hole in a perforated plate, using Table 9 and Eqs. 30, 31, and 33, and the loss of head calculated using Eq. 24.

Energy is dissipated through a throttling orifice because pressure head is converted to velocity head and this conversion is followed by an inefficient pressure head recovery. Conditions may exist at the orifice *vena contracta* that could cause vaporization of the high-velocity, low-pressure liquid jet. Care must be taken in selecting the orifice size to avoid excessive cavitation noise or choke flow. An orifice cavitation index used to check the orifice selection is described by Tung and Mikasinovic,[12] who also discuss the use of orifices in series to avoid cavitation.

Although orifices are used to meter flow, accuracy requires that they be fabricated to standard proportions and that pressure taps be precisely located (Reference 6 and Figure 45). A distinction should be made between *meter differential pressure*, which is used to measure flow and is the difference in pressure at the upstream and downstream *vena contracta* taps, and *meter loss of head* calculated using the resistance coefficient K_1, which is the total overall loss of energy as measured at the upstream tap and past the downstream *vena contracta* tap. The loss of head through a standard orifice meter should be calculated as discussed previously under Meters.

Water Meters and Backflow Preventors Tables 11 and 12 from Reference 17 provide indicative values of losses through these devices.

PUMP FLOW, HEAD, AND POWER IN VARYING TEMPERATURE SYSTEMS

In a pumping system where the weight of liquid pumped is constant, the volumetric flow rate will vary through system components having different temperatures. An example would be the condensate and feedwater system in a steam power plant. The following equation may be used to calculate volumetric flow rate using the specific gravity corresponding to the temperature of the liquid at the location in the system where flow is required:

In USCS units
$$\text{gpm} = \frac{\text{lb/h}}{500\ (\text{sp. gr.})} \tag{41a}$$

In SI units
$$\text{m}^3/\text{h} = \frac{\text{kg/h}}{998\ (\text{sp. gr.})} \tag{41b}$$

When calculating the total head required of a pump or pumps to overcome total system component losses, the actual volumetric flow rate, and temperature through each component must be used because head loss is a function of velocity and viscosity. Information provided in this chapter permits computing pipe, valve, and fitting losses in ft · lb/lb or ft (N · m/N or m) of liquid passing through the component. If the pump is at a location in the system where the temperature is different than at the locations where the head losses are calculated, the total head to be produced by the pump cannot be found by simply adding the individual component heads.

The pump head required to produce a specific increase in pressure varies inversely with specific gravity. Therefore, to calculate pump total head, either the component total

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TABLE 11 Pressure losses for turbine-type water meters

Flow Rate, gpm	Pressure Loss, psi, when Meter Size is:						
	2 in	3 in	4 in	6 in	8 in	10 in	12 in
100	0.8	0.8	0.5				
150	3.5	3.5	1.5				
200	6.5	6.5	3.0				
250	10.0	10.0	4.5				
300		14.0	5.5				
350			7.0				
400			8.0	0.5			
450			9.0	1.0			
500			10.5	1.2	0.5	0.5	
600				2.0	1.0	0.6	
700				3.0	1.1	0.7	
800				4.0	1.4	0.8	
900				5.0	1.7	1.0	
1000				6.0	2.0	1.1	0.6
1500				13.0	4.5	2.0	1.0
2000					8.0	3.0	1.6
2500					13.0	4.0	3.0
3000						6.0	4.5
3500						8.5	6.0
4000						11.0	7.7
5000							11.0
6000							18.0

in × 25.4 mm gpm × .227 = m^3/h psi × .069 = bar

Source: Reference 17

heads must be converted to pump equivalent heads and added together, or all component losses must be expressed in pressure units so the total of these pressures can be converted to an equivalent pump head at the pump temperature. From Eqs. 2 and 3, the pump equivalent head is

$$h_2 = \frac{\gamma_1}{\gamma_2} h_1$$

or

$$h_2 = \frac{\text{sp. gr.}_1}{\text{sp. gr.}_2} h_1$$

where the subscript 1 denotes the component and the subscript 2 denotes the pump. From Eq. 6, individual component head losses can be converted to lb/ft^2 (N/m^2) using

$$p_1 = h_1 \gamma_1$$

TABLE 12 Pressure losses for backflow preventers

Flow Rate, gpm	Pressure Loss, psi, when Preventer Size is:					
	2 in	3 in	4 in	6 in	8 in	10 in
100	3.0					
150	5.5	1.5				
200		2.2	3.0			
300		4.0	2.0			
400			2.5	3.5		
500			3.0	2.5		
600			4.0	2.4	4.5	
700			5.0	2.6	3.6	
800				2.8	3.0	
900				3.0	2.8	
1000				3.3	2.5	4.0
1200				3.7	2.0	3.5
1400				4.3	2.0	2.9
1600				5.0	2.4	2.7
1800					2.6	2.6
2000					2.8	2.5
2500					3.5	2.5
3000						3.0
3500						3.6
4000						4.0
4500						5.0

in × 25.4 mm gpm × .227 = m^3/h psi × .069 = bar

Source: Reference 17

Also for Eq. 6,

$$TH = \frac{P_\Delta}{\gamma_2} = \frac{\Sigma_{p1}}{\gamma_2}$$

from which total component losses in lb/ft^2 (N/m^2) can be converted to an equivalent total pump head in feet (meters) of liquid to be produced at the pumping temperature.

In a varying temperature system, the positive or negative static head required to raise or lower the liquid pumped is not simply a difference in elevation. A pump must produce pressure in a pipe to raise liquid; the pressure required is proportional to the specific weight (force) of the liquid. The static head required at the pump should be found by expressing the suction and discharge elevation heads Z as pressures at the pump suction and discharge connections (corrected to the reference datum plane, if it is not at the pump centerline elevation) and using actual specific weights (forces) along the pipe. This differential pressure, in lb/ft^2 (N/m^2), is then converted to an equivalent static head using the specific weight (force) or specific gravity at the pump in the above appropriate equations.

When designing a pumping system, there may be several locations for placing a pump to produce a specified flow rate in lb/h (kg/h) and an increase in pressure in lb/ft^2 (N/m^2). If the temperature of the liquid varies at the different pump locations being considered (for example, before or after a feedwater heater in a steam power plant), the pump total head in feet (meters) and the volumetric flow rate in gpm (m^3/h) will vary. Although it is true

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that pump power is proportional to the product of volumetric flow × head × specific gravity, higher pumping temperature (lower specific gravity) will nevertheless result in higher pumping power. For the same conditions of weight (or mass) flow and differential pressure, pump power varies inversely with specific gravity because of the following relationships:

$$\text{Pump power} \propto \text{volumetric flow} \times \text{total head} \times \text{sp. gr.}$$

$$\text{Volumetric flow} \propto \frac{\text{weight or mass flow}}{\text{sp. gr.}}$$

$$\text{Total head} \propto \frac{\text{pressure}}{\text{sp. gr.}}$$

therefore,

$$\text{Pump power} \propto \frac{\text{weight or mass flow}}{\text{sp. gr.}} \times \frac{\text{pressure}}{\text{sp. gr.}} \times \text{sp. gr.}$$

then,

$$\text{Pump power} \propto \frac{(\text{weight or mass flow}) \times \text{pressure}}{\text{sp. gr.}}$$

Following are formulas for calculating pump input power in brake horsepower or brake kilowatts:

In USCS units

$$\text{bhp} = \frac{\text{gpm} \times TH \times \text{sp. gr.}}{3960 \times \text{pump eff.}} \tag{42a}$$

$$= \frac{\text{lb/h} \times \text{lb/in}^2}{858{,}600 \times \text{pump eff.} \times \text{sp. gr.}} \tag{43a}$$

In SI units

$$\text{bkW} = \frac{\text{m}^3\text{/h} \times \text{m} \times \text{sp. gr.}}{367.7 \times \text{pump eff.}} \tag{42b}$$

$$= \frac{\text{kg/h} \times \text{kPa}}{3{,}593{,}000 \times \text{pump eff.} \times \text{sp. gr.}} \tag{43b}$$

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SECTION 11.2
BRANCH-LINE PUMPING SYSTEMS

J. P. MESSINA

In some systems, the liquid leaving the pump or pumps will divide into a network of pipes. If the pump is of the centrifugal type, the total pump flow is dependent on the combined system resistance. The total pump flow and flow through each branch can be determined by the following methods. (Review "Pump Total Head and System-Head Curves" in Section 11.1.)

BRANCHES IN CLOSED-LOOP SYSTEMS

Figure 1 illustrates a pump and network of piping consisting of three parallel branches in series with common supply and return headers. Junction points 1 and 2 need not be at the same elevation (provided the liquid density remains constant and the pipes flow full and free of vapor) because, in a closed-loop system, the net change in elevation is zero. Figure 2 shows the system total-head curves for each branch line and header considered independent of the others. These curves are constructed for several flow rates by adding the frictional resistances of the pipes, fittings, and head losses through the equipment serviced from point 1 to point 2. Curves *A, B, C,* and *D* therefore represent the variation in system resistance in feet (meters) versus flow through each branch and header.

If the valves are open in all branches, the total system resistance, total pump flow, and individual branch flows are found by the following method. First observe that (a) the total flow must be equal to the sum of the branch flows, (b) the head loss or pressure drop across each branch from junction 1 to junction 2 is identical, and (c) the flow divides to produce these identical head losses. Therefore, at several head points, add together the flow through each branch and obtain curve $A + B + C$. *Header D* is in series with branches *A, B,* and *C*, and their system heads are added together for several flow conditions to obtain

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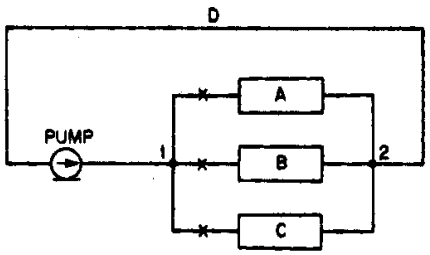


FIGURE 1 Closed-loop pumping system with branch lines

curve $(A + B + C) + D$. On curve E, the head-capacity characteristics of a centrifugal pump, point X represents the pump flow because at this point the system total head and pump total head are equal. Point Y_{1-2} represents the total head across points 1 and 2, and this head determines the flow through each branch; consequently points a, b, and c give individual branch flows. Curve F represents the head-capacity characteristics of a positive displacement pump (constant capacity) that would produce the same flow conditions.

If valve A is open and valves B and C are closed, Figure 3 shows the construction of the curves required to determine pump flow point X'. Obviously the pump flow and branch A flow are the same. Note that the total flow of point X is less than when all valves are open as a result of an increase in system head. If all valves were open and the total flow was obtained by a positive displacement pump having a constant capacity curve F, closing valves B and C would not change the flow. The system head would, however, increase to point X'' and the head would be greater than for a centrifugal pump having curve E.

Also shown in Figure 3 are the system total-head curves for different combinations of open valves A, B, and C and the resulting flow caused by a pump having characteristic curve E. For these various valve combinations, the head differential across the junction points is found by subtracting the head of the curve D from the system total head for the condition investigated, for example, point Y'_{1-2} for only valve A open. The intersection of a horizontal line through point Y'_{1-2} and the individual branch curves gives the branch flow, as illustrated in Figure 2.

BRANCHES IN OPEN-ENDED SYSTEMS

Figure 4 illustrates a pump supplying three branch lines that are open-ended and terminate at different elevations. Figure 5 shows the system total-head curves for each branch

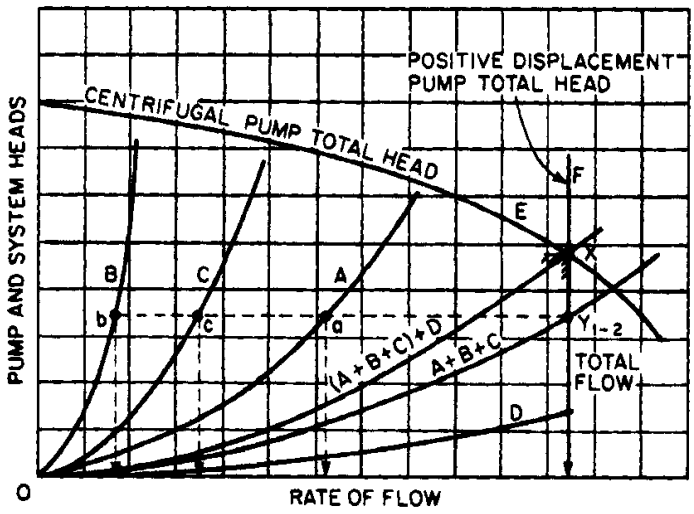


FIGURE 2 System-head curves for pump and branch lines shown in Figure 1 with all valves open

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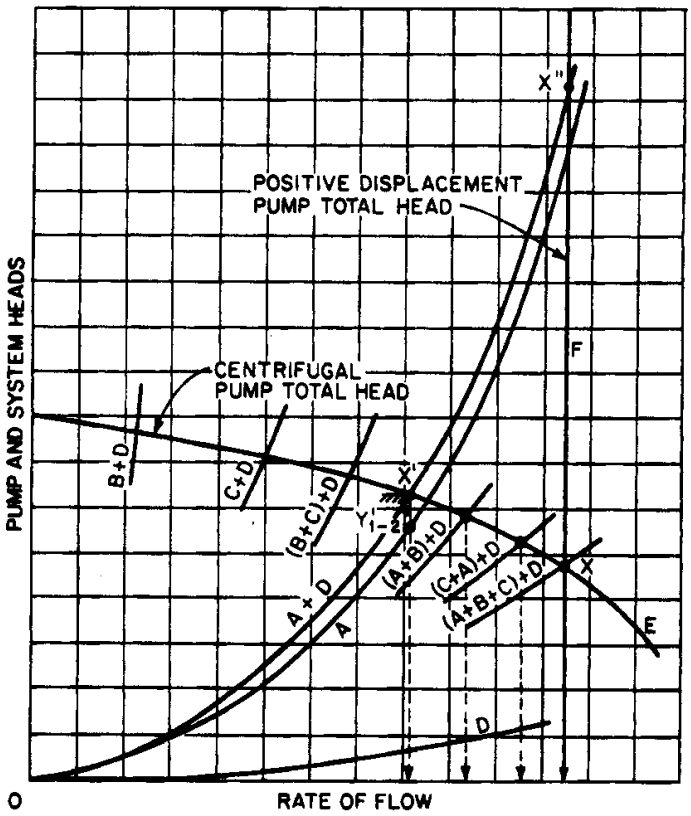


FIGURE 3 System-head curves for pump and branch line shown in Figure 1 with different combinations of open valves

line and main supply line considered independently of each other. These curves are constructed by starting at elevation heads Z_A, Z_B, Z_C, and Z_{DA} at zero flow. To each of these heads is added the frictional resistances in each line for several flow rates. Frictional losses from the suction tank to junction 1 are included in curve D. Curves A, B, C, and D therefore represent the variation in system resistance in feet (meters) versus flow through each branch and supply line. Note that Z_D is negative because in line D there is a decrease in elevation to point 1.

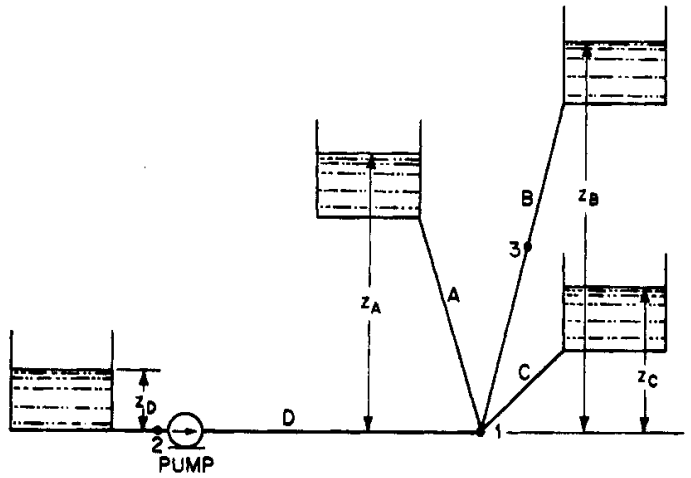


FIGURE 4 Open-ended pumping system with branch lines

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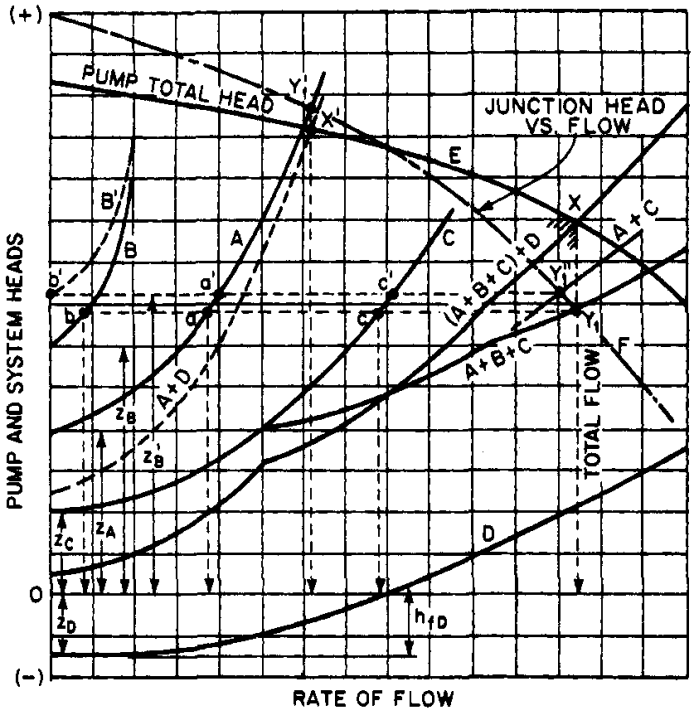


FIGURE 5 System-head curves for pump and branch lines shown in Figure 4

The total head at the junction is the head Z_D in the suction tank measured above point 1 plus the pump total head less the frictional head loss h_{fD} in line D, and it varies with flow, as illustrated by curve F.

The total system resistance, total pump flow, and individual branch flows are found by the following method. First observe that (a) the total flow must be equal to the sum of the branch flows, (b) the frictional resistance plus the elevation head measured relative junction 1 for each branch is identical, and (c) the flow divides to produce these identical total branch heads. Therefore, at several head points, add together the flow of each branch to obtain curve $A + B + C$. Supply line D is in series with branches A, B, and C, and their system heads are added algebraically for several flow conditions to obtain curve $(A + B + C) + D$. If curve E is the head-capacity characteristics of a centrifugal pump, point X represents the pump flow because at this point the system total head and pump total head are equal. Point Y_1 represents the total head at junction 1, and this head determines the flow through each branch; consequently points a, b, and c give individual branch flows.

In this example the pump discharges to all tanks, but this should not be assumed. There is a limiting liquid level elevation for each tank, and, if this level is exceeded, flow will be from the tank into the junction. Therefore, it is possible for the lower-level tanks to be fed by the higher-level tank and the pump. The limit for the liquid elevation in tank B is Z'_B, and it is found from the intersection of curve $A + C$ with curve F, point Y''_1. The flow in branches A and C is at rates a' and c' when there is no flow in branch B. This is also a condition similar to closing a valve in branch B.

If elevation Z_B is greater than previously found limiting height Z'_B flow in branches A and C is determined in the following manner. Construct a curve for junction head versus flow by adding heads and flows that result when the pump and suction tank are in a series with each other and tank B (less line losses) is in parallel with the pump and suction tank. The intersection of this curve with curve $A + C$ will give the junction head required to determine the individual flows from the pump and tank and the flows to tanks A and C (not illustrated).

If flow to branches B and C is shut off, Figure 5 illustrates the construction of the curves required to determine the pump flow point X' and junction head point Y'_1.

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CENTRIFUGAL PUMP BYPASS

Bypass orifices around centrifugal pumps are often used to maintain a minimum flow recommended by the pump manufacturer because of one or more of the following reasons:

- To limit the temperature rise to prevent seizing and cavitation
- To reduce shaft and bearing loads
- To prevent excessive recirculation in the impeller and casing
- To prevent overloading of driver if pump power increases with decrease in flow

Figure 6 illustrates a system that under certain conditions reduces pump flow below the recommended minimum. The pump delivers its flow to either tank *A* or tank *B*. Figure 7 shows the separate system-head curves for flow to tank *A* and for flow to tank *B*. Curve *E* is the head-capacity characteristics of the centrifugal pump. Individual flow rates to each tank are shown as Q_A and Q_B. The recommended minimum flow is Q_R, which is greater than Q_B by the amount shown. In order to maintain the minimum flow, a bypass orifice with necessary pipe, valves, and fittings is required to pass flow Q_c at total head H_R is when the pump discharges to tank *B* only.

Figure 8, (curve *C*) shows the construction necessary to determine the required bypass head versus flow characteristics of the orifice and pipe. The bypass system-head curve *C* includes the pipe, valve, and fitting losses from the pump connection between the suction tank and the end of the bypass piping below the suction water level. These losses must be deducted from the total bypass losses to determine the required orifice head.

Figure 9 illustrates the resultant pump flow with the bypass in operation. Curve *C* is added to curve *B* to obtain curve $B + C$ by combining flows through each system at the same heads. Note that flow through the piping from the suction tank to junction 1 is the total from both systems. Therefore, the combined system-head curve $B + C$ should take this into consideration. Similarly, curve *C* is added to curve *A* to obtain curve $A + C$. Note that when the flow is directed to tank *B* with the bypass open, pump flow is increased from Q_B to Q_R and tank flow is decreased from Q_B to Q'_B. When the flow is directed to tank *A* with the bypass open, pump flow is increased from Q_A to Q_p and tank flow is decreased from Q_A to Q'_A.

If it is desired that there be no reduction in flow or that there be no waste of pumping power when flow is to tank *A*, the bypass can be closed either manually or automatically. If pump flow is monitored, this measurement can be used to open, close, or modulate the bypass valve automatically to maintain desired flow. Refer to Section 11.4 for more detailed information.

If operating procedures require that the pump occasionally be run with a closed valve (at the pump discharge or at tanks *A* and *B*), the bypass line must be designed to recirculate all of the minimum required pump flow Q_R, dissipating head H_R, shown as curve *C'* and Figures 8 and 9.

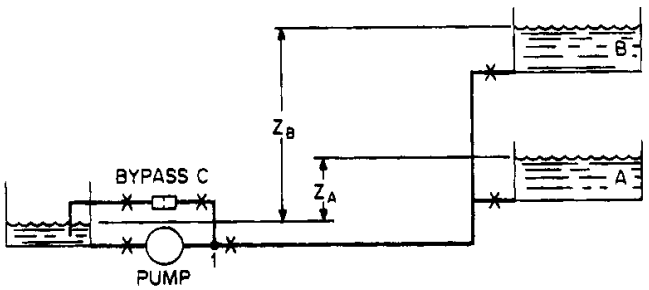


FIGURE 6 Pump with bypass to maintain minimum flow

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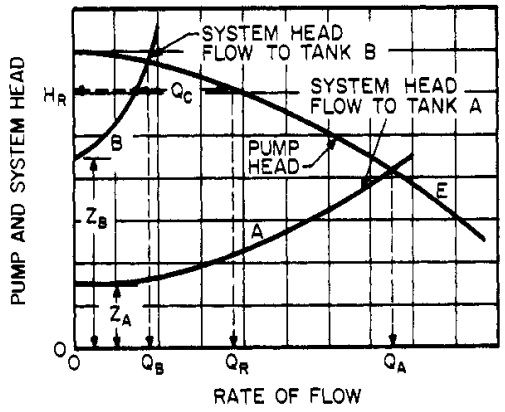


FIGURE 7 System-head curves for pump and tanks shown in Figure 6 with bypass valve closed

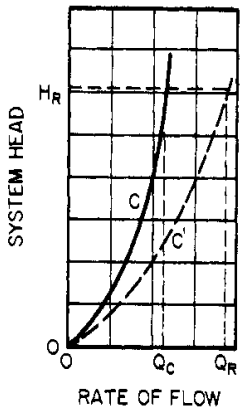


FIGURE 8 Bypass orifice system requirements

FLOW CONTROL THROUGH BRANCHES

The flow through branches *A, B,* and *C* in Figures 1 and 4 is dependent on the individual branch characteristics. When parallel branches are connected to a pump, the resulting division of flow may not satisfy the requirements of the individual lines. If it is desired that the flow to each branch meet or exceed specified individual line requirements, it is necessary only to select a pump to provide the maximum head required by any one branch. In those branches where this head is more than required, the flow will be greater than the desired amount. A throttling valve or other flow-restricting device may be used to reduce the flow in these branches to the desired quantity. If the flow is controlled in this manner, the pump need be selected to produce only the minimum total system flow at a total head required to satisfy the branch needing the highest head at junction 1 in Figures 1 and 4.

The flow through a branch is sometimes dictated by the requirement of a component elsewhere in the system. For example, in the system shown in Figure 10, the required flow through component *V* could be greater than the sum of the required flows through components *A, B,* and *C*. An additional branch line and control valve *F* around components *A, B,* and *C* may be used to bypass the additional flow needed by component *D*. Individual component throttling valves may also be used, if needed, to adjust the flow in each branch.

The following example illustrates how flow through branches may be controlled and how pump total head is calculated.

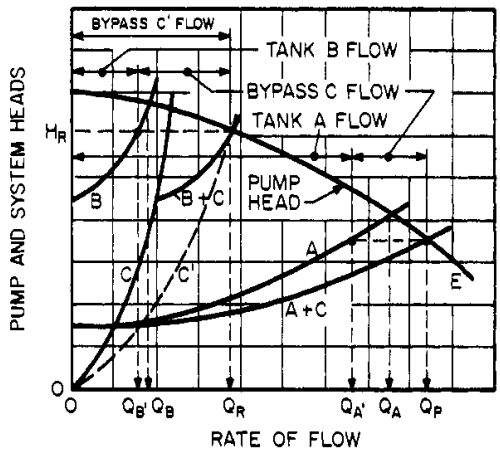


FIGURE 9 System-head curves for pump and tanks shown in Figure 6 with bypass valve open

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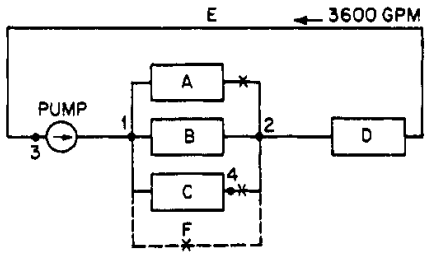


FIGURE 10 Example of a branch-flow pumping system

EXAMPLE A pump is required to circulate water at a rate of 3600 gpm (817 m^3/h) through the system shown in Figure 10. The head versus flow characteristics of the system components *A, B,* C, *D,* and *E* (system pipe and fittings) are shown in Figure 11. The branch pipe and fitting losses from point 1 to point 2 are included in the total heads for components *A, B,* and *C*. Determine

1. The pump total head required and the individual flows through components *A, B,* and *C*
2. The pump total head required if the flow through components *A, B,* and *C* need be only 800, 700, and 1000 gpm (182, 159, and 227 m^3/h), respectively, and a bypass *F* is installed

Calculate the individual throttling valve head drops to achieve a controlled branch flow system.

Solution

1. Head versus flow curves *A, B,* and *C* of Figure 11 are added together in parallel, giving curve *(A + B + C)*. Curves (A + *B* + *C*), *D*, and *E* are added together in

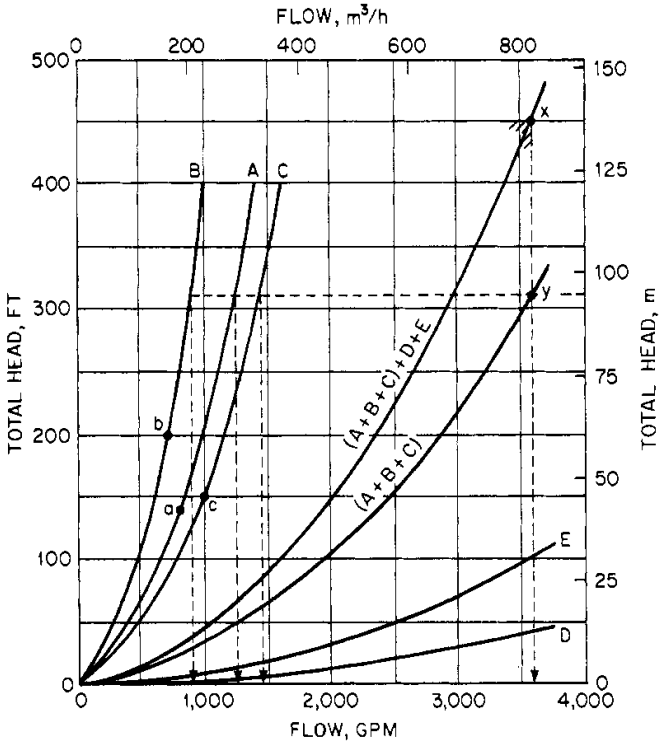


FIGURE 11 System-head curves required for solutions to example problems

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series, resulting in curve $(A + B + C) + D + E$. This latter curve indicates that 450 ft (137 m) total pump head is required at 3600 gpm (817 m³/h), point *X*. Curve $(A + B + C)$ crosses point *Y* at 3600 gpm (817 m³/h) flow through the branches, and this condition requires 310 ft (94.5 m) total head, which is the head across branch points 1 and 2. From each individual component curve, the flow through branches *A*, *B*, and *C* can be read as 1250, 900, and 1450 gpm (284, 204, and 329 m³/h), respectively.

2. Because the total flow through components *A*, *B*, and *C* need be only 2500 gpm (800 + 700 + 1000) [568 m³/h (182 + 159 + 227)], the bypass should be designed to pass 1100 gpm (249 m³/h). Component *B* requires the maximum head, 200 ft (61 m) differential (point *b*) across points 1 and 2. Throttling valves are needed in components *A* and *C* and bypass *F* to increase the head in each branch to 200 ft (61 m) at the required flows. Branch *A* (point *a*) requires only 140 ft (42.7 m) total head to pass 800 gpm (182 m³/h); therefore, the throttling valve must be designed for a 60-ft (42.7-m) head loss. Branch *C* (point *c*) requires only 150 ft (45.7 m) total head to pass 1000 gpm (227 m³/h), requiring a throttling valve for a 50-ft (15.3-m) head loss. The bypass control valve and piping should be designed to produce a 200-ft (61-m) head drop at 1100 gpm (249 m³/h).

At 3600 gpm (817 m³/h), the pump is now required to overcome 200 ft (61 m) total head across points 1 and 2, 40 ft (12.2 m) total head through component *D*, and 100 ft (30.1 m) total head through the system pipe and fittings, component *E*, for a total of 340 ft (103.3 m). The reduction in pumping head from 450 to 340 ft (137.1 to 103.6 m), a saving of 110 ft (33.5 m), or 24.4% water power, is the result of decreasing the branch head from 310 to 200 ft (94.5 to 61 m) by bypassing the excess flow.

PUMP TOTAL HEAD IN BRANCH-LINE SYSTEMS

The total head produced is the difference in total heads measured across the suction and discharge connections of a pump. As explained in Section 11.1, the total head is also the difference between the heads at any two points in the pumping system, one on each side of the pump, plus the sum of the head losses between these two points. Confusion sometimes results when the flow through the pump divides into branches in either closed-loop or open-ended systems. The points of head measurement can be in any branch line, upstream or downstream from the pump, regardless of the flow rate in these lines.

In part 2 of the previous example, the pump or system total head could be measured using points 3 and 4 in Figure 10. If the total head measured at the pump suction, point 3, were 25 ft (7.62 m) gage, the head measured at point 4 would be 205 ft (62.48 m) gage above the same reference datum plane, assuming 10 ft (3.05 m) of friction between the pump discharge and point 1. The difference between the head at point 3 and that at point 4 is 180 ft (54.86 m). The loss of head due to friction and the head drop through component *C* is 10 + 150 = 160 ft (3.05 + 47.5 = 48.75 m). The pump and system total head at 3600 gpm (817 m³/h) is therefore 180 + 160 = 340 ft (54.86 + 48.75 = 103.63 m).

Similarly, the pump and system total head for an open-ended system, such as the one shown in Figure 4, could be found by measuring, for example, the difference between the head at point 2 and that at point 3. Each head measurement would be referred to a common datum plane. The total head loss from 2 to 1 plus the head loss from 1 to 3, at the rate of flow in their respective lines, added to the difference between the head at 2 and that at 3, is the pump and system total head.

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SECTION 11.3
WATERHAMMER

C. SAMUEL MARTIN

INTRODUCTION

By definition waterhammer is a pressure (acoustic) wave phenomenon created by relatively sudden changes in the liquid velocity. In pipelines, sudden changes in the flow (velocity) can occur as a result of (1) pump and valve operation in pipelines, (2) vapor pocket collapse, or (3) even the impact of water following the rapid expulsion of air out of a vent or a partially open valve. Although the name waterhammer may appear to be a misnomer in that it implies only water and the connotation of a "hammering" noise, it has become a generic term for pressure wave effects in liquids. Strictly speaking, waterhammer can be directly related to the compressibility of the liquid, primarily water, in this handbook. For slow changes in pipeline flow for which pressure waves have little to no effect, the unsteady flow phenomenon is called surging.

Potentially, waterhammer can create serious consequences for pipeline designers if not properly recognized and addressed by analysis and design modifications. There have been numerous pipeline failures of varying degrees and resulting repercussions of loss of property and life. Three principal design tactics for mitigation of waterhammer are (1) alteration of pipeline properties such as profile and diameter, (2) implementation of improved valve and pump control procedures, and (3) design and installation of surge control devices.

In this section waterhammer and surging are defined and discussed in detail with reference to the two dominant sources of waterhammer—pump and/or valve operation. Detailed discussion of the hydraulic aspects of both valves and pumps and their effect on hydraulic transients will be presented. The undesirable and unwanted, but often potentially possible, event of liquid column separation and rejoining are a common justification for surge protection devices. Both the beneficial and detrimental effect of free (entrained or entrapped) air in water pipelines will be discussed with reference to waterhammer and surging. Finally, the efficacy of various surge protection devices for mitigation of waterhammer is included.

NOMENCLATURE

a	Sonic (acoustic) velocity of plane wave in a pipe
a_o	Sonic (acoustic) velocity of plane wave in a pipe
A	Cross-sectional area of pipe
A_v	Projected area of partially open valve
A_{vo}	Projected area of fully open valve

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C_f	Valve discharge coefficient based on A_v and ΔH (partially open valve)
C_{fo}	Valve discharge coefficient based on A_{vo} and ΔH (fully open valve)
$C_H=gH/\omega^2D^2$	Dimensionless head coefficient
C_Q	Valve discharge coefficient based on A_{vo} and H
$C_Q=Q/\omega D^3$	Dimensionless flow coefficient
$C_P=P/\rho\omega^3D_I^5$	Dimensionless power coefficient
$C_T=T/\rho\omega^2D_I^5$	Dimensionless torque coefficient
D	Internal diameter of pipe
D_I	Diameter of pump impeller
e	Thickness of pipe wall
E	Elastic (Young's) modulus of pipe wall material
g	Gravitational acceleration
$h=H/H_R$	Dimensionless head ratio
H	Total dynamic head (TDH) across pump
H_R	Rated head, preferably at best efficiency point (BEP)
ΔH	Energy (head) loss attributed to valve
K	Bulk modulus of fluid
K_L	Head loss coefficient based upon ΔH and V
K_{Lo}	Head loss coefficient when valve is fully open
L	Length of pipe
N	Rotational speed of pump impeller (rpm)
N_R	Rated rotational speed of pump impeller
P	Power input on pump shaft
P_R	Rated power of pump
Q	Discharge through pump. Positive during normal pumping
Q_R	Rated discharge through pump
t_c	Valve closure time
t_e	Elastic time constant $(2L/a)$
t_f	Inertia time constant $(LV_o)/(gH_o)$
$T=P/\omega$	Fluid torque on pump shaft
T_R	Rated fluid torque on pump shaft
$v=Q/Q_R$	Dimensionless discharge ratio
V	Fluid velocity
$\alpha=N/N_R$	Dimensionless rotational speed of pump impeller
$\beta=T/T_R$	Dimensionless torque ratio
ρ	Fluid mass density
ω	Rotational speed of pump impeller (rad/sec)
ω_R	Rated rotational speed of pump impeller (rad/sec)

FUNDAMENTALS OF WATERHAMMER AND SURGE

The fundamentals of waterhammer, an elastic process; and surging, an incompressible phenomenon; are both developed on the basis of the basic conservational relationships of physics or fluid mechanics. The acoustic velocity stems from mass balance (continuity),

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while the fundamental waterhammer equation of Joukowsky originates from the application of linear momentum.

Definitions Some of the terms frequently employed in waterhammer are defined as follow:

Waterhammer. A pressure wave phenomenon for which liquid compressibility plays a role.

Surging. An unsteady phenomenon governed solely by inertia. Often termed mass oscillation or referred to as either rigid column or inelastic effect.

Liquid column separation. The formation of vapor cavities and their subsequent collapse and associated waterhammer on rejoining.

Entrapped air. Free air located in a pipeline as a result of incomplete filling, inadequate venting, leaks under vacuum, air entrained from pump intake vortexing, and other sources.[1]

Acoustic velocity. The speed of a waterhammer or pressure wave in a pipeline.

Joukowsky equation. Fundamental relationship relating waterhammer pressure change with velocity change and acoustic velocity. Strictly speaking, valid only for sudden flow changes.

Acoustic Velocity For wave propagation in liquid-filled pipes the acoustic (sonic) velocity is modified by the pipe wall elasticity by varying degrees, depending upon the elastic properties of the wall material and the relative wall thickness. The expression for the wave speed is

$$a = \frac{\sqrt{K/\rho}}{\sqrt{1 + \dfrac{D}{e}\dfrac{K}{E}}} = \frac{a_o}{\sqrt{1 + \dfrac{D}{e}\dfrac{K}{E}}} \tag{1}$$

where E is the elastic modulus of the pipe wall, D is the inside diameter of the pipe, e is the wall thickness, and a_o is the acoustic velocity in the liquid medium. In a very rigid pipe, or in a tank, or in large water bodies, the acoustic velocity a reduces to the well-known relationship $a = a_o = /(K/\rho)$. For water $K = 2.19$ GPa (318,000 psi) and $\rho = 998$ kg/m^3 (1.936 slug/ft^3 or 62.3 lbm/ft^3), yielding a value of $a_o = 1483$ m/s (4865 ft/s), a value many times that of any liquid velocity V.

Joukowsky (Waterhammer) Equation There is always a pressure change Δp associated with the rapid velocity change ΔV across a waterhammer (pressure) wave. The relationship between Δp and ΔV from the basic physics of linear momentum yields the well-known Joukowsky equation.

$$\Delta p = -\rho a \Delta V \tag{2}$$

where ρ is the liquid mass density, and a is the sonic velocity of the pressure wave in the fluid medium in the conduit. Conveniently using the concept of head, the Joukowsky head rise for instantaneous valve closure is

$$\Delta H = \frac{\Delta p}{\rho g} = -\frac{\rho a \Delta V}{\rho g} = \frac{a V_o}{g} \tag{3}$$

The compliance of a conduit or pipe wall can have a significant effect on modification of (1) the acoustic velocity and (2) any resultant waterhammer, as can be shown from Eq. 1 and Eq. 2, respectively. For simple waterhammer waves for which only radial pipe motion (hoop stress) effects are considered, the germane physical pipe properties are Young's elastic modulus (E) and Poisson ratio (μ). Table 1 summarizes appropriate values of these two physical properties for some common pipe materials.

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TABLE 1 Physical properties of common pipe materials

Material	Young's modulus, E GPa	Young's modulus, E 10^6 psi	Poisson's ratio μ
Asbestos cement	23 to 24	3.3 to 3.5	–
Cast iron	80 to 170	11.6 to 24.7	0.25 to 0.27
Concrete	14 to 30	2.0 to 4.4	0.10 to 0.15
Concrete (reinforced)	30 to 60	4.4 to 8.7	–
Ductile iron	172	25.0	0.30
Polyethylene	0.7 to 0.8	0.10 to 0.12	0.46
PVC (polyvinyl chloride)	2.4 to 3.5	0.35 to 0.51	0.46
Steel	200 to 207	29.0 to 30.0	0.30

The effect of the elastic modulus (E) on the acoustic velocity in water-filled circular pipes for a range of the ratio of internal pipe diameter to wall thickness (D/e) is shown in Figure 1 for various pipe materials.

HYDRAULIC CHARACTERISTICS OF VALVES

Valves are integral elements of any piping system used for the handling and transport of liquids. Their primary purposes are flow control, energy dissipation, and isolation of portions of the piping system for maintenance. It is important for the purposes of design and final operation to understand the hydraulic characteristics of valves under both steady and unsteady flow conditions. Examples of dynamic conditions are direct opening or closing of valves by a motor, the response of a swing check valve under unsteady conditions, and the action of hydraulic servo valves. The hydraulic characteristics of valves under either noncavitating or cavitating conditions vary considerably from one type of valve design to another. Moreover, valve characteristics also depend upon particular valve design for a special function, upon absolute size, on manufacturer as well as the type of pipefitting employed. In this section the fundamentals of valve hydraulics are presented in terms of pressure drop (head loss) characteristics. Typical flow characteristics of selected valve types of control—gate, ball, butterfly, and cone—are presented. The hydraulic characteristics of swing check valves and their dynamic response are also described.

Descriptions of Various Types of Valves Valves used for the control of liquid flow vary widely in size, shape, and overall design due to vast differences in application. They can vary in size from a few millimeters in small tubing to many meters in hydroelectric installations, for which spherical and butterfly valves of very special design are built. The hydraulic characteristics of all types of valves, albeit different in design and size, can always be reduced to the same basic coefficients, notwithstanding fluid effects such as viscosity and cavitation. Figure 2 shows cross-sections of some valve types to be discussed with relation to hydraulic performance.

Definition of Geometric Characteristics of Valves The valve geometry, expressed in terms of cross-sectional area at any opening, sharpness of edges, type of passage, and valve shape, has a considerable influence on the eventual hydraulic characteristics. In order to understand the hydraulic characteristics of valves it is useful, however, to express the projected area of the valve in terms of geometric quantities. With reference to Figure 2 the ratio of the projected open area of the valve A_v to the full open valve A_{vo} can be related to the valve opening, either a linear measure for a gate valve, or an angular one for rotary valves such as ball, cone, plug, and butterfly types. It should be noted that this geometric feature of the valve clearly has a bearing on the valve hydraulic performance, but should not be used directly for prediction of hydraulic performance—either steady state or transient.

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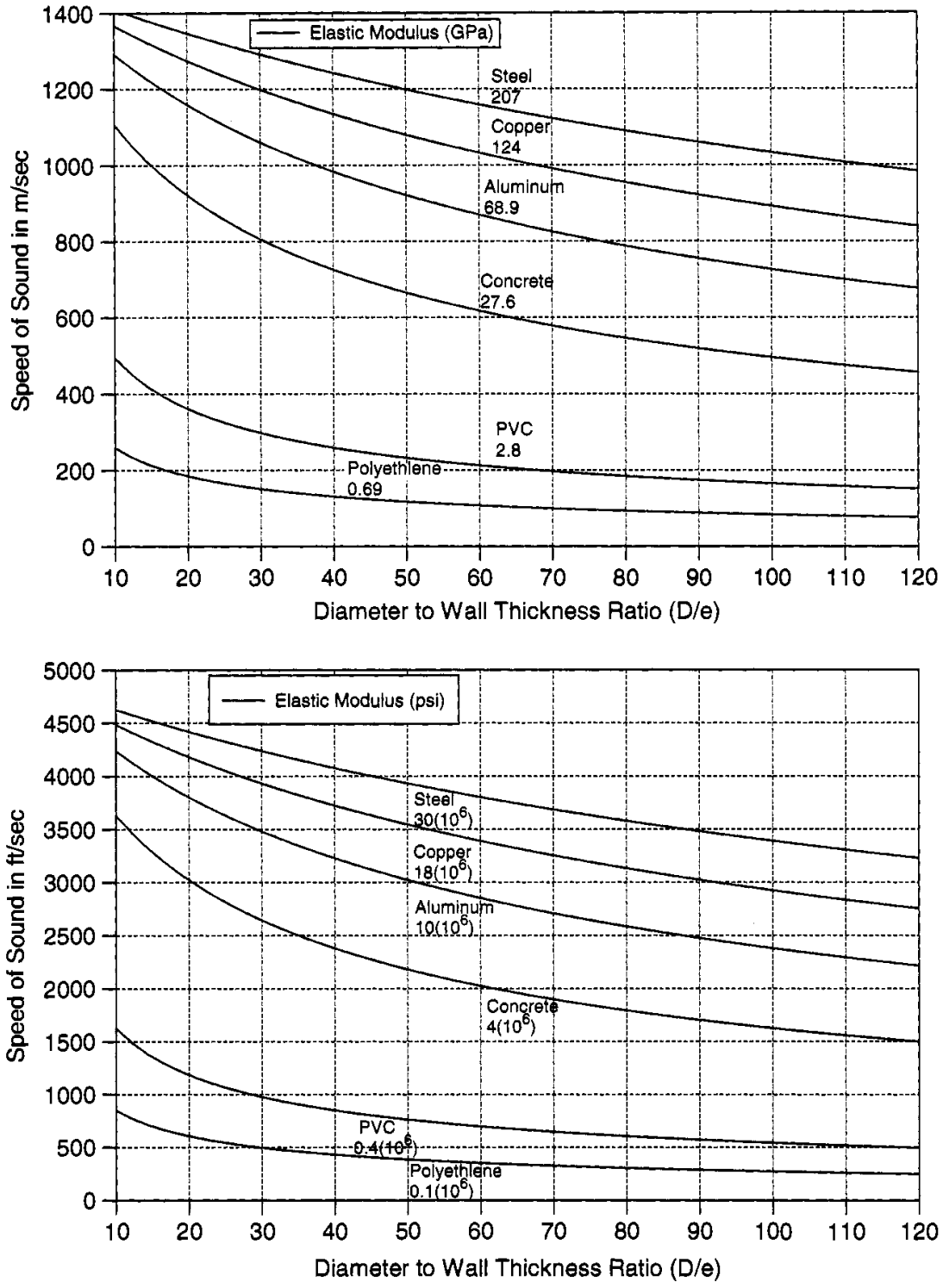


FIGURE 1 Effect of wall thickness of various pipe materials on acoustic velocity in water-filled pipes

The actual hydraulic performance to be used in transient calculations should originate from experiment.

Definition of Hydraulic Performance of Valves The hydraulic performance of a valve depends upon the flow passage through the valve opening and the subsequent recovery of pressure. The hydraulic characteristics of a valve under partial to fully opened conditions

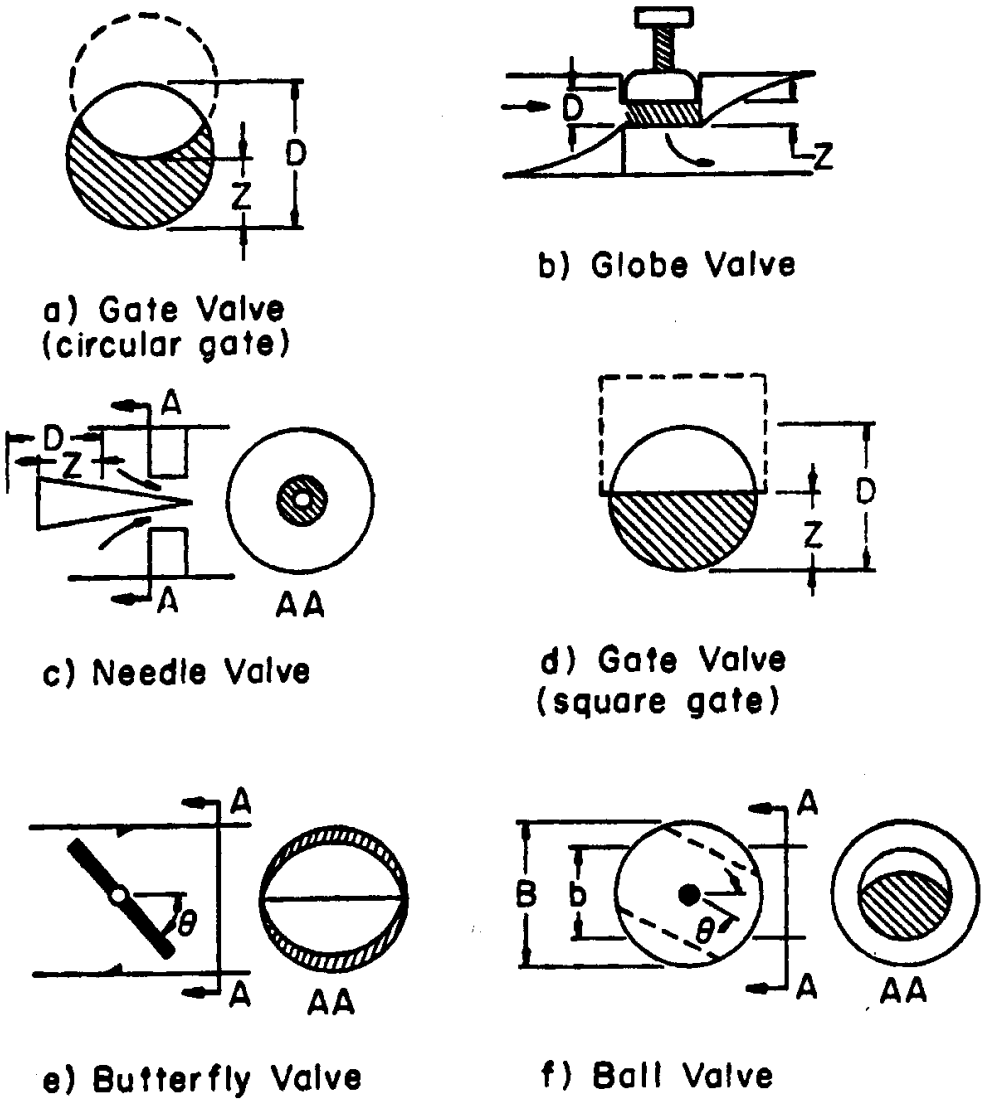


FIGURE 2 Cross sections of selected control valves (after Wood and Jones, Reference 2)

typically relate the volumetric flow rate to a characteristic valve area and the head loss ΔH across the valve. The principal fluid properties that can affect the flow characteristics are fluid density ρ, fluid viscosity μ, and liquid vapor pressure p_v if cavitation occurs. Except for very small valves and/or quite viscous liquids, Reynolds number effects are usually not important, and will be neglected with reference to water. A valve in a pipeline acts as an obstruction, disturbs the flow, and in general causes a loss in energy as well as affecting the pressure distribution both upstream and downstream. The characteristics are expressed either in terms of (a) flow capacity as a function of a defined pressure drop or (b) energy dissipation (head loss) as a function of pipe velocity. In both instances the pressure or head drop is usually the difference in total head caused by the presence of the valve itself, minus any loss due to regular pipe friction between measuring stations.

The proper manner in determining ΔH experimentally is to measure the hydraulic grade line (HGL) far enough both upstream and downstream of the valve so that uniform flow sections to the left of and to the right of the valve can be established, allowing for the extrapolation of the energy grade lines (EGL) to the plane of the valve. Otherwise, the valve head loss is not properly defined. It is common to express the hydraulic characteristics either in terms of a head loss coefficient K_L or as a discharge coefficient where A_v is the area of the valve at any opening, and ΔH is the head loss defined for the valve. Frequently a discharge coefficient is defined in terms of the fully-open valve area. The hydraulic coefficients embody not only the geometric features of the valve through A_v but also the flow characteristics.

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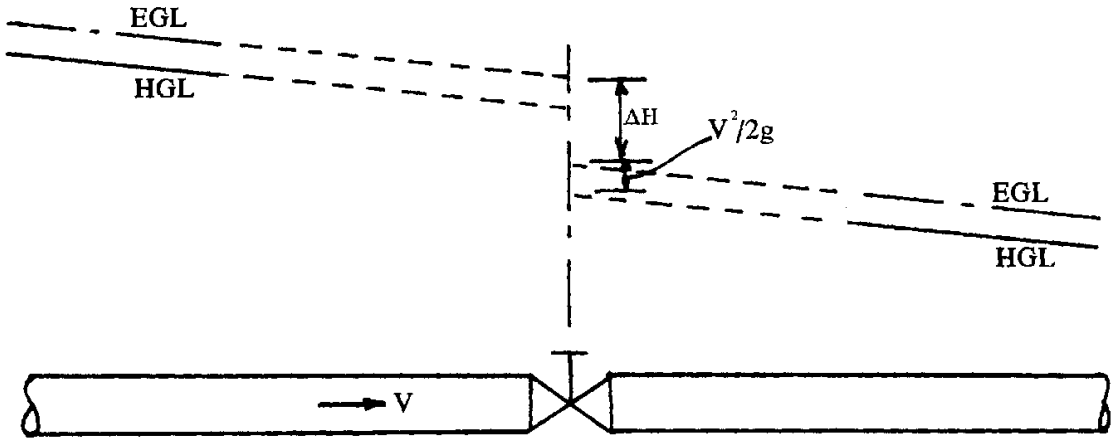


FIGURE 3 Definition of head loss characteristics of a valve

Unless uniform flow is established far upstream and downstream of a valve in a pipeline the value of any of the coefficients can be affected by effects of nonuniform flow. It is not unusual for investigators to use only two pressure taps—one upstream and one downstream, frequently one and ten diameters, respectively. The flow characteristics of valves in terms of pressure drop or head loss have been determined for numerous valves by many investigators and countless manufacturers. Only a few sets of data and typical curves will be presented here for ball, butterfly, gate, and check valves. For a valve located in the interior of a long continuous pipe, as shown in Figure 3, the presence of the valve disturbs the flow both upstream and downstream of the obstruction as reflected by the velocity distribution, and the pressure variation, which will be nonhydrostatic in the regions of nonuniform flow. Accounting for the pipe friction between upstream and downstream uniform flow sections, the head loss across the valve is expressed in terms of the pipe velocity and a head-loss coefficient K_L.

$$\Delta H = K_L \frac{V^2}{2g} \tag{4}$$

Often manufacturers represent the hydraulic characteristics in terms of discharge coefficients.

$$Q = C_f A_{vo} \sqrt{2g\,\Delta H} = C_Q A_{vo} \sqrt{2gH} \tag{5}$$

where

$$H = \Delta H + \frac{V^2}{2g} \tag{6}$$

Both discharge coefficients are defined in terms of the nominal full-open valve area A_{vo} and a representative head, ΔH for C_f and H for C_Q, the latter definition generally reserved for large valves employed in the hydroelectric industry. The interrelationship between C_f, C_Q, and K_L is

$$K_L = \frac{1}{C_f^2} = \frac{1 - C_Q^2}{C_Q^2} \tag{7}$$

Frequently valve characteristics are expressed in terms of a dimensional flow coefficient C_v from the valve industry.

$$Q = C_v \sqrt{\Delta p} \tag{8}$$

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where Q is in US flow units of gpm and Δp is the pressure loss in psi. In transient analysis it is convenient to relate either the loss coefficient or the discharge coefficient to the corresponding value at the fully open valve position, for which the $C_f = C_{fo}$. Hence,

$$\frac{Q}{Q_o} = \frac{C_f}{C_{fo}}\sqrt{\frac{\Delta H}{\Delta H_o}} = \tau\sqrt{\frac{\Delta H}{\Delta H_o}} \tag{9}$$

Traditionally the dimensionless valve discharge coefficient is termed τ and defined by

$$\tau = \frac{C_f}{C_{fo}} = \frac{C_v}{C_{vo}} = \frac{C_Q}{C_{Qo}} = \sqrt{\frac{K_{Lo}}{K_L}} \tag{10}$$

Typical Geometric and Hydraulic Valve Characteristics The geometric projected area of valves shown in Figure 2 can be calculated for ball, butterfly, and gate valves using simple expressions. The dimensionless hydraulic flow coefficient τ is plotted in Figure 4 for various valve openings for the three selected valves along with the area ratio for comparison. The lower diagram, which is based on hydraulic measurements, should be used for transients calculations rather than the upper one, which is strictly geometric.

Valve Operation The instantaneous closure of a valve at the end of a pipe will yield a pressure rise satisfying Joukowsky's equation—Eq. 2 or Eq. 3. In this case the velocity difference $\Delta V = 0 - V_o$, where V_o is the initial velocity of liquid in the pipe. Although Eq. 2 applies across every wavelet, the effect of complete valve closure over a period of time greater than $2L/a$, where L is the distance along the pipe from the point of wave creation to the location of the first pipe area change, can be quite beneficial. Actually, for a simple pipeline the maximum head rise remains that from Eq. 3 for times of valve closure $t_c = 2L/a$, where L is the length of pipe. If the value of $t_c > 2L/a$, then there can be a considerable reduction of the peak pressure resulting from beneficial effects of negative wave reflections from the open end or reservoir considered in the analysis. The phenomenon can still be classified as waterhammer until the time of closure $t_c \gg 2L/a$, beyond which time there are only inertial or incompressible deceleration effects, referred to as surging, also known as rigid column analysis. Table 2 is furnished to classify four types of valve closure, independent of type of valve.

Using standard waterhammer programs, parametric analyses can be conducted for the preparation of charts to demonstrate the effect of time of closure, type of valve, and an indication of the physical process—waterhammer or simply inertia effects of deceleration. The charts are based on analysis of valve closure for a simple reservoir-pipe-valve arrangement. For simplicity fluid friction is often neglected, a reasonable assumption for pipes on the order of hundreds of feet in length.

HYDRAULIC CHARACTERISTICS OF PUMPS

Transient analyses of piping systems involving centrifugal, mixed-flow, and axial-flow pumps require detailed information describing the characteristics of the respective turbomachine, which may pass through unusual, indeed abnormal, flow regimes. Since little if any information is available regarding the dynamic behavior of the pump in question, invariably the decision must be made to use the steady-flow characteristics of the machine gathered from laboratory tests. Moreover, complete steady-flow characteristics of the machine may not be available for all possible modes of operation that may be encountered in practice.

In this section steady-flow characteristics of pumps in all possible zones of operation are defined. The importance of geometric and dynamic similitude is first discussed with respect to both (1) homologous relationships for steady flow and (2) the importance of the assumption of similarity for transient analysis. The significance of the eight zones of operation

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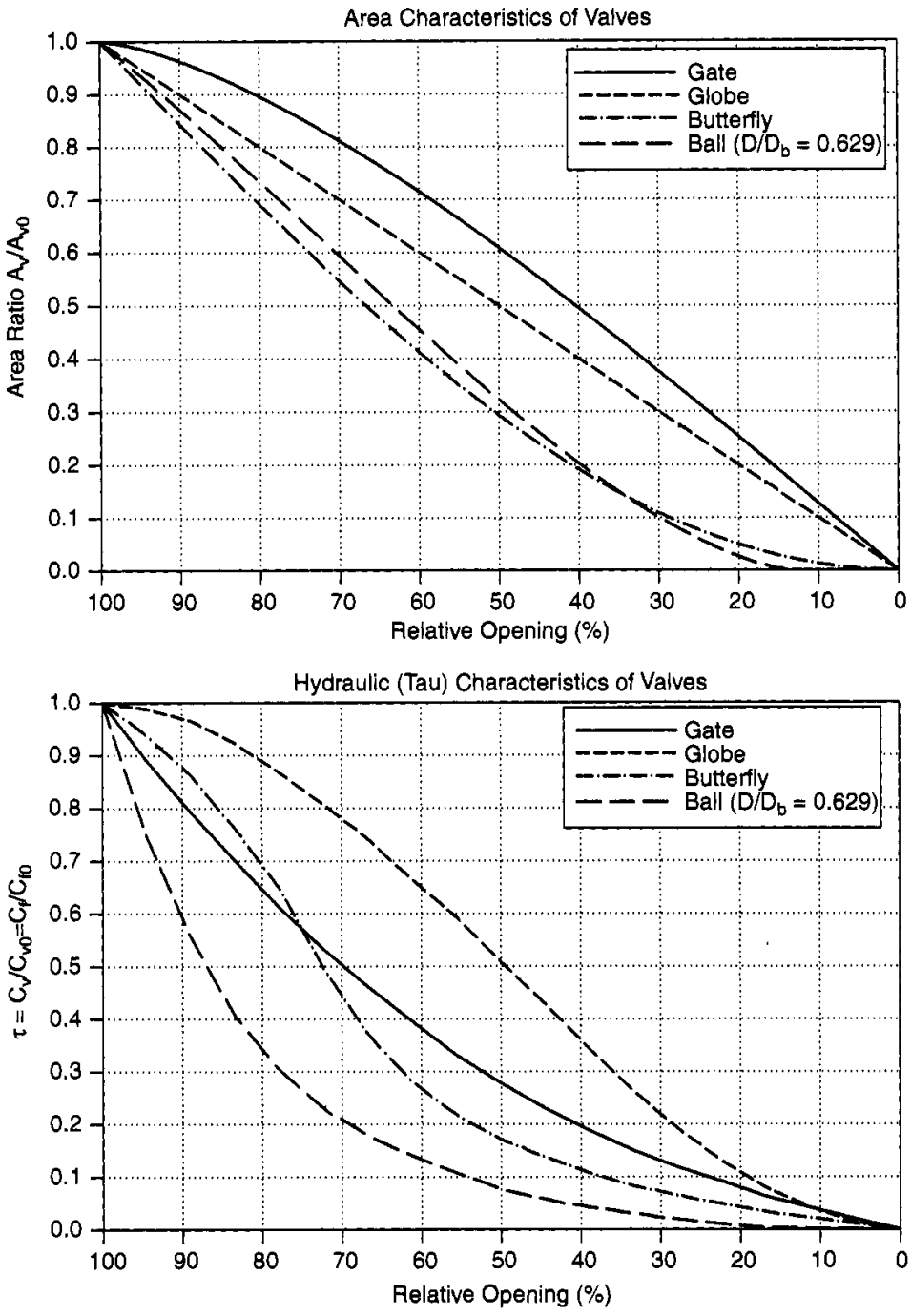


FIGURE 4 Geometric and hydraulic characteristics of typical control valves

TABLE 2 Classification of valve closure

Time of closure t_c	Type of closure	Maximum head ΔH_{max}	Phenomenon
0	Instantaneous	aV_o/g	Waterhammer
$= 2L/a$	Rapid	aV_o/g	Waterhammer
$> 2L/a$	Gradual	$< aV_o/g$	Waterhammer
$\gg 2L/a$	Slow	$\ll aV_o/g$	Surging

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within each of the four quadrants is presented in detail with reference to three possible modes of data representation. The steady-flow characteristics of pumps are discussed in detail with regard to the complete range of possible operation. The loss of driving power to a pump is usually the most critical transient case to consider for pumps, because of the possibility of low pipeline pressures which may lead to (1) pipe collapse due to buckling, or (2) the formation of a vapor cavity and its subsequent collapse. Other waterhammer problems may occur due to slam of a swing check valve, or from a discharge valve closing either too quickly (column separation), or too slowly (surging from reverse flow). For radial-flow pumps for which the reverse flow reaches a maximum just subsequent to passing through zero speed (locked rotor point), and then is decelerated as the shaft runs faster in the turbine zone, the head will usually rise above the nominal operating value. As reported by Donsky, mixed-flow and axial-flow pumps may not even experience an upsurge in the turbine zone because the maximum flow tends to occur closer to runaway conditions.[8]

Definition of Pump Characteristics The essential parameters for definition of hydraulic performance of pumps are defined as

Impeller diameter. Exit diameter of pump rotor D_I

Rotational speed. The angular velocity (rad/sec) is ω, while $N = 2\pi\omega/60$ is in rpm

Flow rate. Capacity Q at operating point in chosen units

Total dynamic head (TDH). The total energy gain (or loss) H across pump, defined as

$$H = \left(\frac{p_d}{\gamma} + z_d\right) - \left(\frac{p_s}{\gamma} + z_s\right) + \frac{V_d^2}{2g} - \frac{V_s^2}{2g} \tag{11}$$

where subscripts s and d refer to suction and discharge sides of the pump, respectively.

Homologous (Affinity) Laws Dynamic similitude, or dimensionless representation of test results, has been applied with perhaps more success in the area of hydraulic machinery than in any other field involving fluid mechanics. Due to the sheer magnitude of the problem of data handling it is imperative that dimensionless parameters be employed for transient analysis of hydraulic machines that are continually experiencing changes in speed as well as passing through several zones of normal and abnormal operation. For liquids for which thermal effects may be neglected, the remaining fluid-related forces are pressure (head), fluid inertia, resistance, phase change (cavitation), surface tension, compressibility, and gravity. If the discussion is limited to single-phase liquid flow, three of the above fluid effects—cavitation, surface tension, and gravity (no interfaces within machine)—can be eliminated, leaving the forces of pressure, inertia, viscous resistance, and compressibility. For the steady or even transient behavior of hydraulic machinery conducting liquids the effect of compressibility may be neglected.

In terms of dimensionless ratios the three forces yield an Euler number (ratio of inertia force to pressure force), which is dependent upon geometry, and a Reynolds number. For all flowing situations the viscous force, as represented by the Reynolds number, is definitely present. If water is the fluid medium, the effect of the Reynolds number on the characteristics of hydraulic machinery can usually be neglected, with the major exception being the prediction of the performance of a large hydraulic turbine on the basis of model data. For the transient behavior of a given machine the actual change in the value of the Reynolds number is usually inconsequential anyway. The elimination of the viscous force from the original list reduces the number of fluid-type forces from seven to two—pressure (head) and inertia, as exemplified by the Euler number. The appellation geometry in the functional relationship in the above equation embodies primarily the shape of the rotating impeller; the entrance and exit flow passages, including effects of vanes, diffusers, and so on; secondly the effect of surface roughness; and lastly the geometry of the streamline pattern, better known as kinematic similitude in contrast to the first two, which are related to geometric similarity. Kinematic similarity is invoked on the assumption that similar flow patterns can be specified by congruent velocity triangles composed of peripheral speed U and absolute fluid velocity V at inlet or exit to the vanes.

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This allows for the definition of a flow coefficient, expressed in terms of impeller diameter D_I and angular velocity ω

$$C_Q = \frac{Q}{\omega D_I^3} \tag{12}$$

The reciprocal of the Euler number (ratio of pressure force to inertia force) is the head coefficient, defined as

$$C_H = \frac{gH}{\omega^2 D_I^2} \tag{13}$$

For power requirements a power coefficient can be defined

$$C_P = \frac{P}{\rho \omega^3 D_I^5} \tag{14}$$

For transient analysis, the desired parameter for the continuous prediction of pump speed ω is the unbalance torque T. Since $T = P/\omega$, the torque coefficient becomes

$$C_T = \frac{T}{\rho \omega^2 D_I^5} \tag{15}$$

It is traditional in hydraulic transient analysis to refer pump characteristics to so-called *rated conditions,* which preferably should be the optimum or best efficiency point (BEP), but sometimes defined as the duty, nameplate, or design point. Nevertheless, in terms of *rated conditions*, for which the subscript R is employed, the following ratios are defined

$$\textit{Flow: } v = \frac{Q}{Q_R} \qquad \textit{Speed: } \alpha = \frac{\omega}{\omega_R} = \frac{N}{N_R} \qquad \textit{Head: } h = \frac{H}{H_R} \qquad \textit{Torque: } \beta = \frac{T}{T_R}$$

Next, for a given pump undergoing a transient, for which D_I is a constant, Eqs. 12 to 15 can be written in terms of the above ratios:

$$\frac{v}{\alpha} = \frac{C_Q}{C_{QR}} = \frac{Q_R}{Q}\frac{\omega}{\omega_R} \qquad \frac{h}{\alpha^2} = \frac{C_H}{C_{HR}} = \frac{H}{H_R}\frac{\omega_R^2}{\omega^2} \qquad \frac{\beta}{\alpha^2} = \frac{C_T}{C_{TR}} = \frac{T}{T_R}\frac{\omega_R^2}{\omega^2}$$

Abnormal Pump (Four Quadrant) Characteristics The performance characteristics discussed up to this point correspond to pumps operating normally. During a transient, however, the machine may experience either a reversal in flow, or rotational speed, or both, depending upon the situation. It is also possible that the torque and head may reverse in sign during passage of the machine through abnormal zones of performance. The need for characteristics of a pump in abnormal zones of operation can best be described with reference to Figure 5, which is a simulated pump power-failure transient. A centrifugal pump is delivering water at a constant rate when there is a sudden loss of power from the prime mover—in this case an electric motor. For the postulated case of no discharge valves, or other means of controlling the flow, the loss of driving torque leads to an immediate deceleration of the shaft speed, and in turn the flow. The three curves are dimensionless head (h), flow (v), and speed (α), shown in Figure 5 respectively as percentages H, Q, and N. With no additional means of controlling the flow, the higher head at the final delivery point (another reservoir) will eventually cause the flow to reverse ($v < 0$) while the inertia of the rotating parts has maintained positive rotation ($\alpha > 0$). Up until the time of flow reversal the pump has been operating in the normal zone, albeit at a number of off-peak flows.

In order to predict system performance in regions of negative rotation and/or negative flow the analyst requires characteristics in these regions for the machine in question. Indeed, any peculiar characteristic of the pump in these regions could be expected to have

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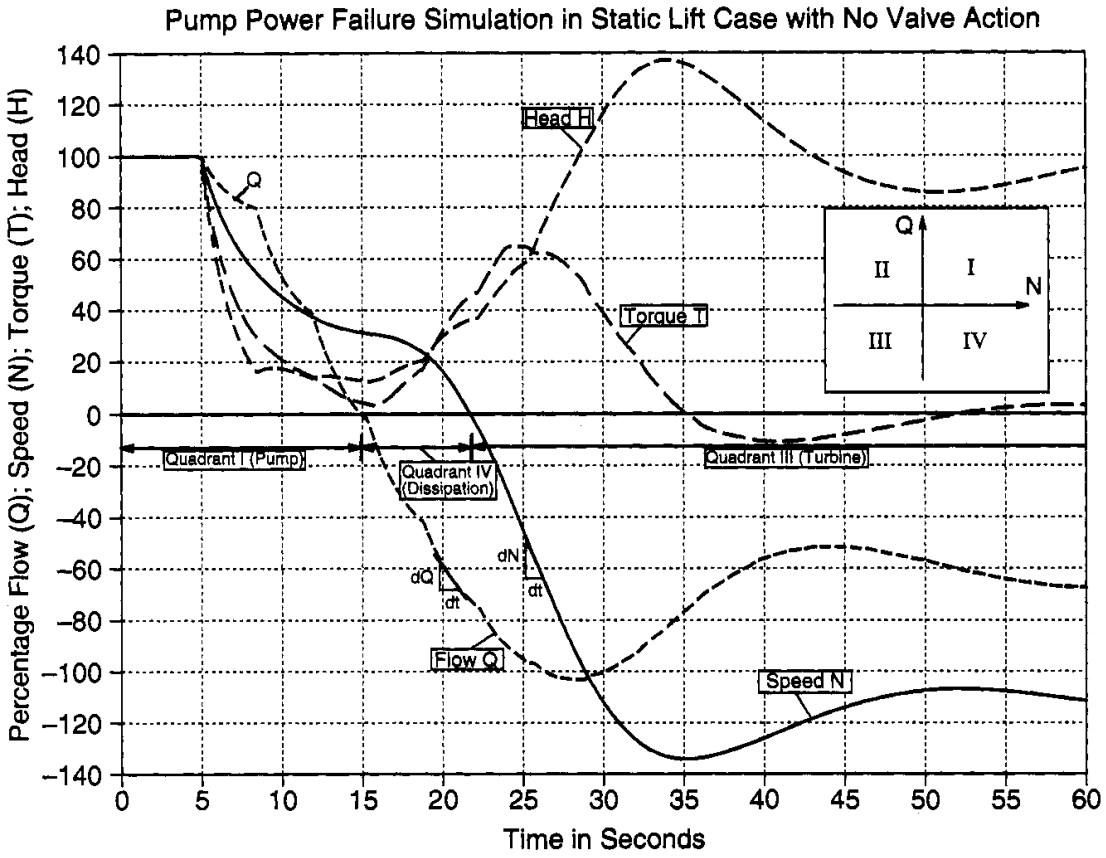


FIGURE 5 Simulated pump trip without valves in a single pipeline system

an influence on the hydraulic transients. It is important to stress that the results of such analyses are critically governed by the following three factors: (1) availability of complete pump characteristics in zones the pump will operate, (2) complete reliance upon dynamic similitude (homologous) laws during transients, and (3) assumption that steady-flow derived pump characteristics are valid for transient analysis.

Investigations by Knapp[4] and Kittredge[5] facilitated the understanding of abnormal operation, as well as served to reinforce the need for test data. (Kittredge's summary of that work appears in Section 2.1.3 of this book.) Following the work by Knapp[6] and Swanson,[7] and a summary of their results by Donsky,[3] eight possible zones of operation, four normal and four abnormal, will be discussed here with reference to Figure 6, developed by Martin.[8] In Figure 6 the head H is shown as the difference in the two reservoir elevations to simplify the illustration. The effect of pipe friction may be ignored for this discussion by assuming that the pipe is short and of relatively large diameter. The regions referred to on Figure 6 are termed Zones and Quadrants, the latter definition originating from plots of lines of constant head and constant torque on a flow-speed plane ($v - \alpha$ axes). Quadrants I ($v > 0$, $\alpha > 0$) and III ($v < 0$, $\alpha < 0$) are defined in general as regions of pump or turbine operation, respectively. It will be seen, however, that abnormal operation (neither pump nor turbine mode) may occur in either of these two quadrants. A very detailed description of each of the eight zones of operation is in order. It should be noted that all of the conditions shown schematically in Figure 6 can be contrived in a laboratory test loop using an additional pump (or two) as the master and the test pump as a slave. Most, if not all, of the zones shown can also be experienced by a pump during a transient under the appropriate set of circumstances.

QUADRANT I Zone A (normal pumping) in Figure 6 depicts a pump under normal operation for which all four quantities, Q, N, H, and T, are regarded as positive. In this case $Q > 0$, indicating useful application of energy. Zone B (energy dissipation) is a condition of positive flow, positive rotation, and positive torque, but negative head—quite an abnormal

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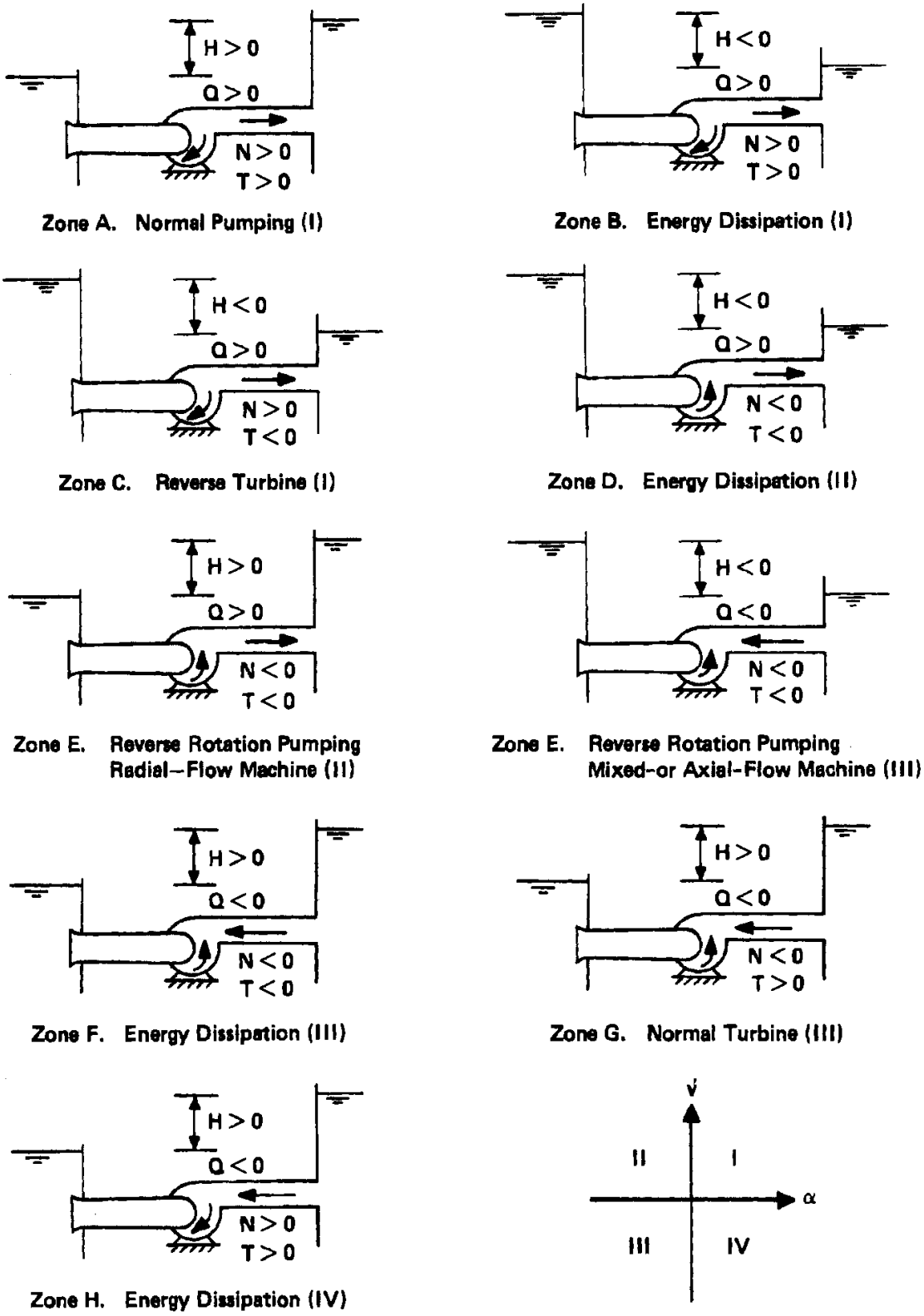


FIGURE 6 Four quadrants and eight zones of possible pump operation (after Martin, Reference 8)

condition. A machine could operate in Zone B by (1) being overpowered by another pump or by a reservoir during steady operation, or (2) by a sudden drop in head during a transient caused by power failure. It is possible, but not desirable, for a pump to generate power with both the flow and rotation in the normal positive direction for a pump, Zone C (reverse turbine), caused by a negative head, and resulting in a positive efficiency because

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of the negative torque. The maximum efficiency would be quite low due to the bad entrance flow condition and unusual exit velocity triangle.

QUADRANT IV Zone H, labeled energy dissipation, is often encountered shortly after a trip out or power failure of a pump, as illustrated in Figure 5. In this instance the combined inertia of all the rotating elements—motor, pump and its entrained liquid, and shaft—has maintained pump rotation positive but at a reduced value at the time of flow reversal caused by the positive head on the machine. This purely dissipative mode results in a negative or zero efficiency. It is important to note that both the head and fluid torque are positive in Zone H, the only zone in Quadrant IV.

QUADRANT III A machine that passes through Zone H during a pump power failure will then enter Zone G (normal turbining) provided that reverse shaft rotation is not precluded by a mechanical ratchet. Although a runaway machine rotating freely is not generating power, Zone G is the precise mode of operation for a hydraulic turbine. Note that the head and torque are positive, as for a pump but that the flow and speed are negative, opposite to that for a pump under normal operation (Zone A).

Subsequent to the trip out or load rejection of a hydraulic turbine or the continual operation of a machine that failed earlier as a pump, Zone F (energy dissipation) can be encountered. The difference between Zones F and G is that the torque has changed sign for Zone F, resulting in a braking effect, which tends to slow the freewheeling machine down. In fact, the real runaway condition is attained at the boundary of the two zones, for which torque $T = 0$.

QUADRANT II The two remaining Zones, D and E, are very unusual and infrequently encountered in operation, with the exception of pump/turbines entering Zone E during transient operation. Again it should be emphasized that both zones can be experienced by a pump in a test loop, or in practice in the event a machine is inadvertently rotated in the wrong direction by improper wiring of an electric motor. Zone D is a purely dissipative mode that normally would not occur in practice unless a pump, which was designed to increase the flow from a higher to lower reservoir, was rotated in reverse, but did not have the capacity to reverse the flow (Zone E, mixed or axial flow), resulting in $Q > 0, N < 0$, $T < 0$, for $H < 0$. Zone E, for which the pump efficiency > 0, could occur in practice under steady flow if the preferred rotation as a pump was reversed. There is always the question regarding the eventual direction of the flow. A radial-flow machine will produce positive flow at a much reduced capacity and efficiency compared to $N > 0$ (normal pumping), yielding of course $H > 0$. On the other hand, mixed and axial-flow machines create flow in the opposite direction (Quadrant III), and $H < 0$, which corresponds still to an increase in head across the machine in the direction of flow.

Representation of Pump Data for Numerical Analysis It is conventional in transient analyses to represent h/α^2 and β/α^2 as functions of v/α, as shown in Figures 7 and 8 for a radial-flow pump. The curves on Figure 7 are only for positive rotation ($\alpha > 0$), and constitute pump Zones A, B, and C for $v > 0$ and the region of energy dissipation subsequent to pump power failure (Zone H), for which $v < 0$. The remainder of the pump characteristics are plotted in Figure 8 for $\alpha < 0$. The complete characteristics of the pump plotted in Figures 7 and 8 can also be correlated on what is known as a Karman-Knapp circle diagram, a plot of lines of constant head (h) and torque (β) on the coordinates of dimensionless flow (v) and speed (α). Figure 9 is such a correlation for the same pump. The complete characteristics of the pump require six curves, three each for head and torque. For example, the h/α^2 curves from Figures 7 and 8 can be represented by continuous lines for $h = 1$ and $h = -1$, and two straight lines through the origin for $h = 0$. A similar pattern exists for the torque (β) lines. In addition to the eight Zones A-H illustrated in Figure 6, the four Karman-Knapp quadrants in terms of v and α, are well defined. Radial lines on Figure 9 correspond to constant values for v/α in Figures 7 and 8, allowing for relatively easy transformation from one form of presentation to the other.

In computer analysis of pump transients, Figures 7 and 8, while meaningful from the standpoint of physical understanding, are fraught with the difficulty of $|v/\alpha|$ becoming

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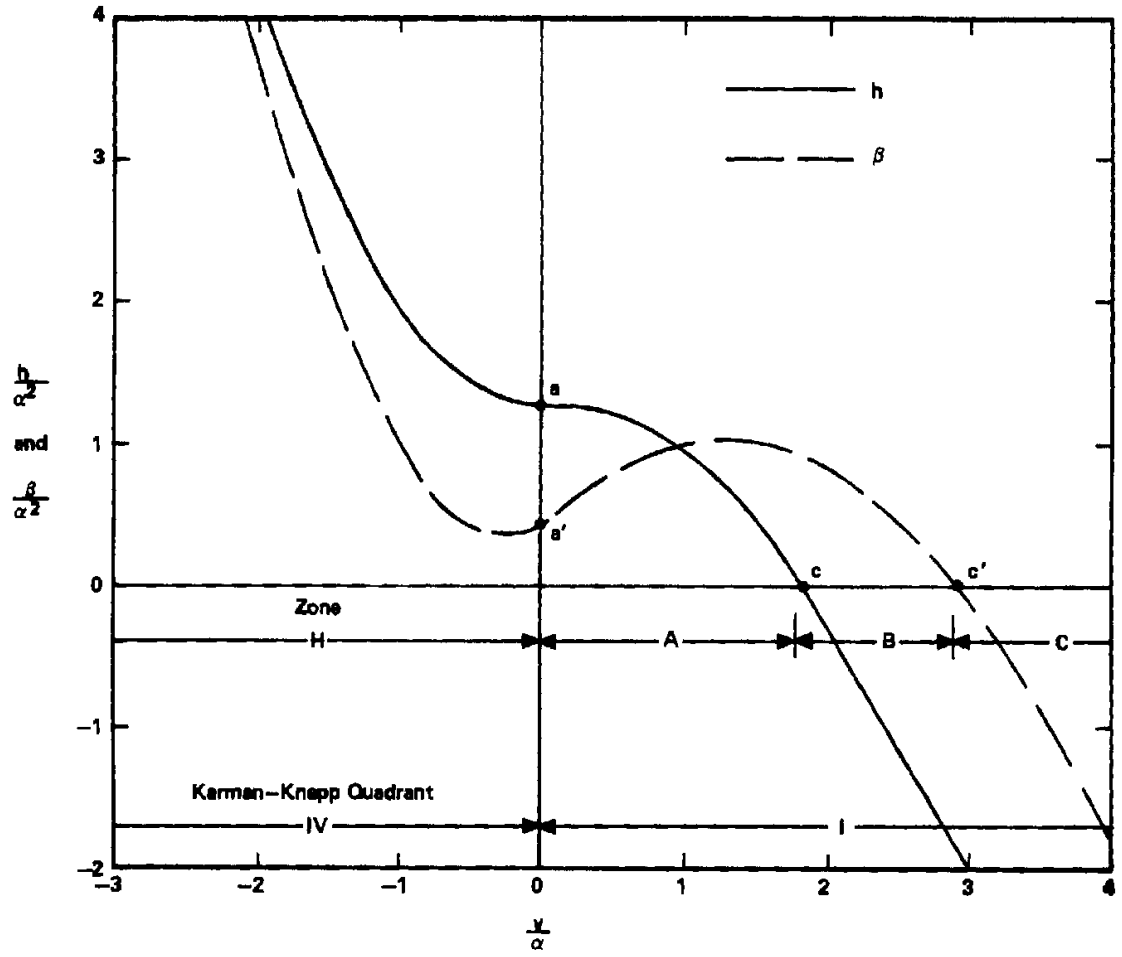
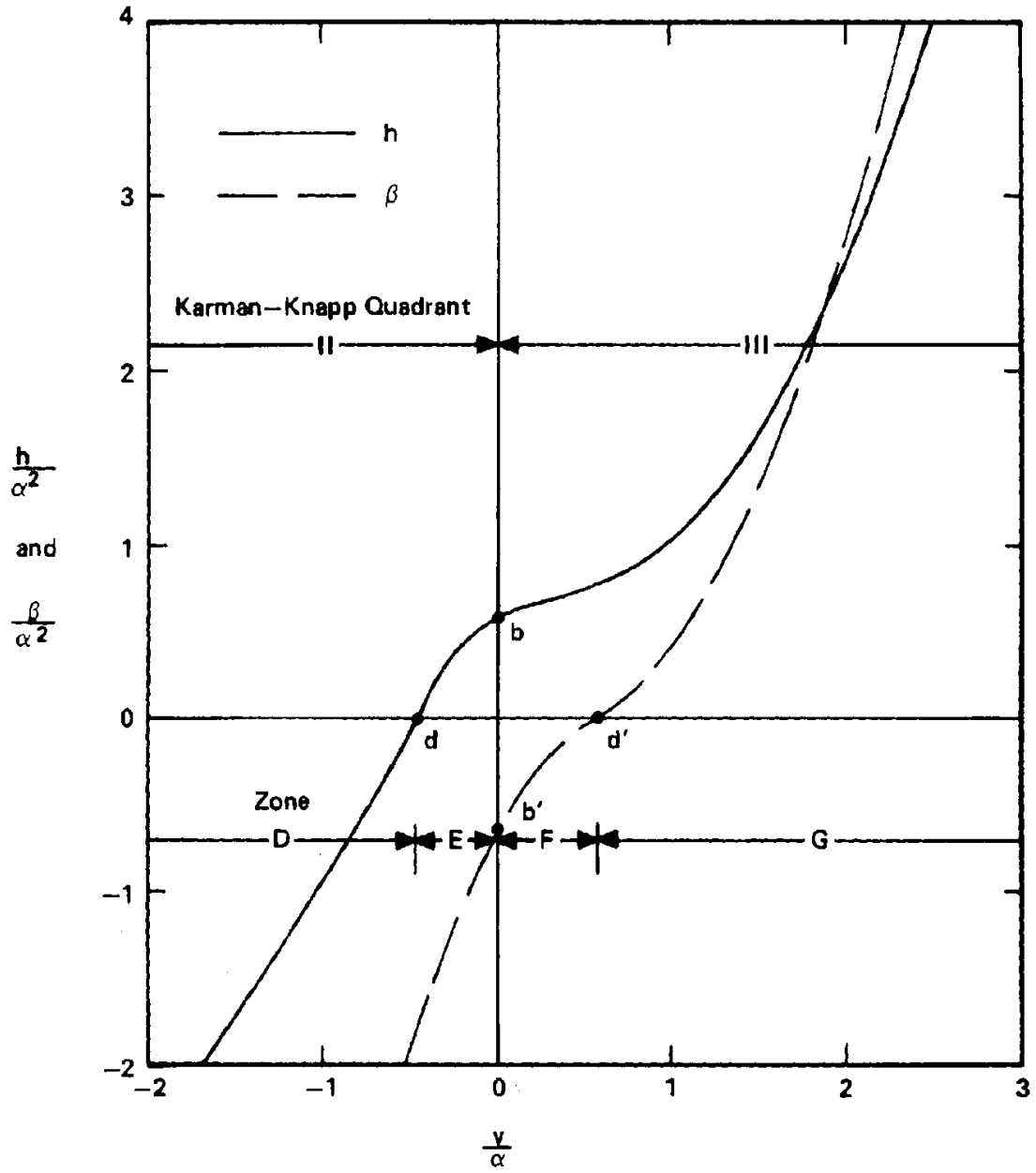


FIGURE 7 Complete head and torque characteristics of radial-flow pump for positive rotation (after Martin, Reference 8)

infinite as the unit passes through, or remains at, zero speed ($\alpha = 0$). Some have solved that problem by switching from h/α^2 versus v/α to h/v^2 versus α/v, and likewise for β, for $|v/\alpha| > 1$. This technique doubles the number of curves on Figures 7 and 8, and thereby creates discontinuities in the slopes of the lines at $|v/\alpha| = 1$, in addition to complicating the storing and interpolation of data. Marchal, Flesh, and Suter devised a useful transformation that allowed the complete pump characteristics to be represented by two single curves, as shown for the same pump in Figure 10.[9] The difficulty of v/α becoming infinite was eliminated by utilizing the function $\tan^{-1}(v/\alpha)$ as the abscissa. The eight zones, or four quadrants can then be connected by the continuous functions. Although some of the physical interpretation of pump data has been lost in the transformation, Figure 10 is now a preferred correlation for transient analysis using a digital computer because of function continuity and ease of numerical interpolation. The singularities in Figures 7 and 8 and the asymptotes in Figure 9 have now been avoided.

Critical Data Required for Hydraulic Analysis of Systems with Pumps Regarding data from manufacturers such as pump curves (normal and abnormal), pump and motor inertia, motor torque-speed curves, and valve curves, probably the most critical for pumping stations are pump-motor inertia and valve closure time. Normal pump curves are usually available and adequate. A few examples of abnormal pump curves have been given in this section, in Section 2.1.3 and in Stepanoff[10] for radial- and axial-flow pumps; however, such information describes only how those particular pumps behave in such

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Homologous Head and Torque Characteristics for Radial–Flow Pump (Ω_s = 0.465 in Universal Units) for Negative Rotation ($\alpha < o$)

FIGURE 8 Complete head and torque characteristics of radial-flow pump for negative rotation (after Martin, Reference 8)

circumstances. For any other pump, actual data of this kind for the machine in question is essential for reliable hydraulic analysis of a pump system. Motor torque-speed curves are only needed when evaluating pump start-up. For pump trip the inertia of the combined pump and motor is quite important.

SURGE PROTECTION AND SURGE CONTROL DEVICES

There are numerous techniques for controlling transients and waterhammer, some involving design considerations and others the consideration of surge protection devices. There

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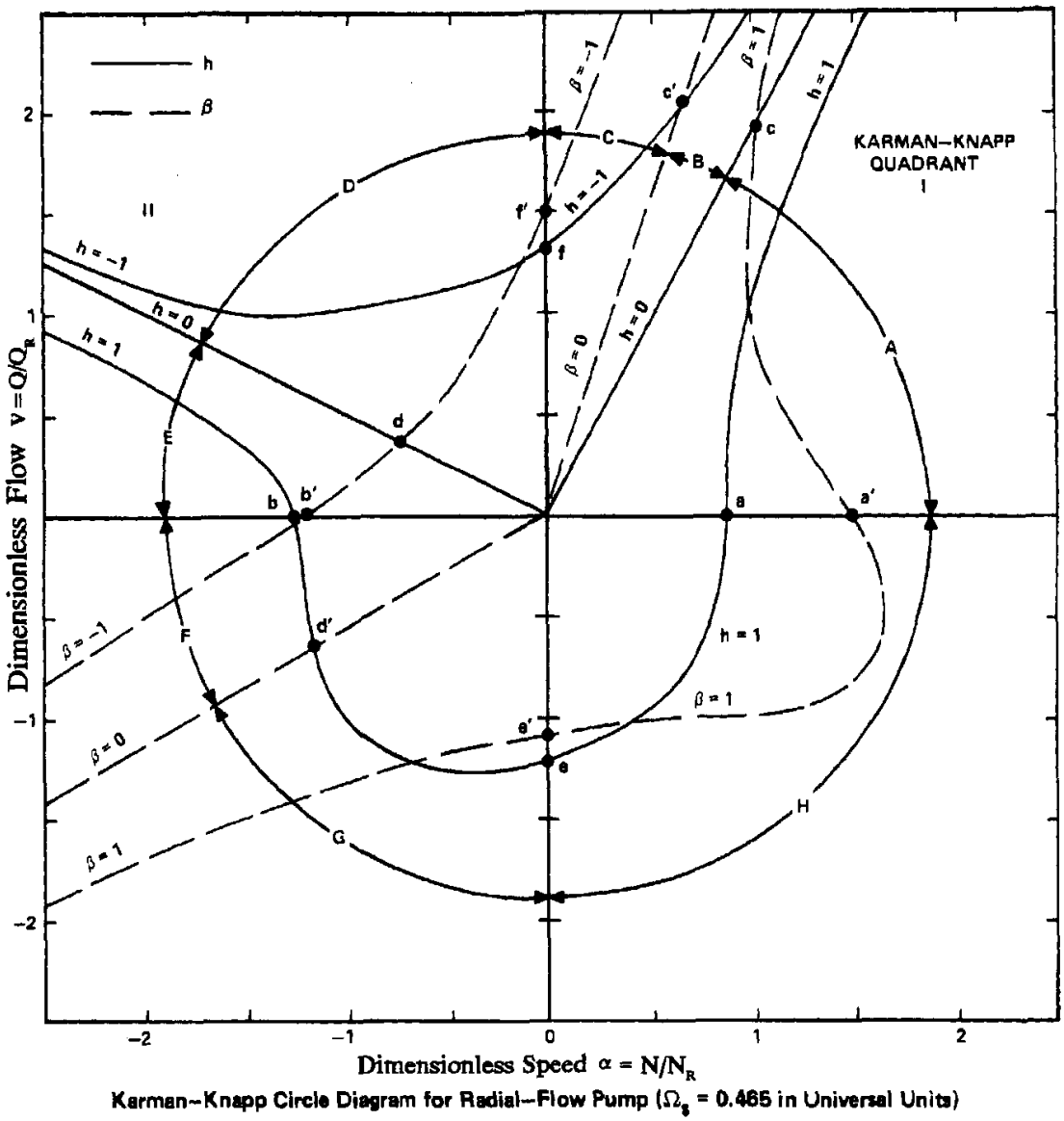


FIGURE 9 Complete four-quadrant head and torque characteristics of radial-flow pump (after Martin, Reference 8)

must be a complete design and operational strategy devised to combat potential waterhammer in a system. The transient event may either initiate a low-pressure event (downsurge) as in the case of a pump power failure, or a high-pressure event (upsurge) caused by the closure of a downstream valve. It is well known that a downsurge can lead to the undesirable occurrence of water column separation, which itself can result in severe pressure rises following the collapse of a vapor cavity. In some systems negative pressures are not even allowed because of (1) possible pipe collapse or (2) ingress of outside water or air.

The means of controlling the transient will in general vary, depending upon whether the initiating event results in an upsurge or downsurge. For pumping plants the major cause of unwanted transients is typically the complete outage of pumps due to loss of electricity to the motor. For full pipelines pump start-up, usually against a closed pump discharge valve for centrifugal pumps, does not normally result in significant pressure transients. Indeed, the majority of transient problems in pumping installations are associated with the potential (or realized) occurrence of water column separation and vapor-pocket collapse, resulting from the trip out of one or more pumps, with or without valve action. The pump-discharge valve, if actuated too suddenly, can even aggravate the downsurge problem. To combat the downsurge problem there are a number of options, mostly

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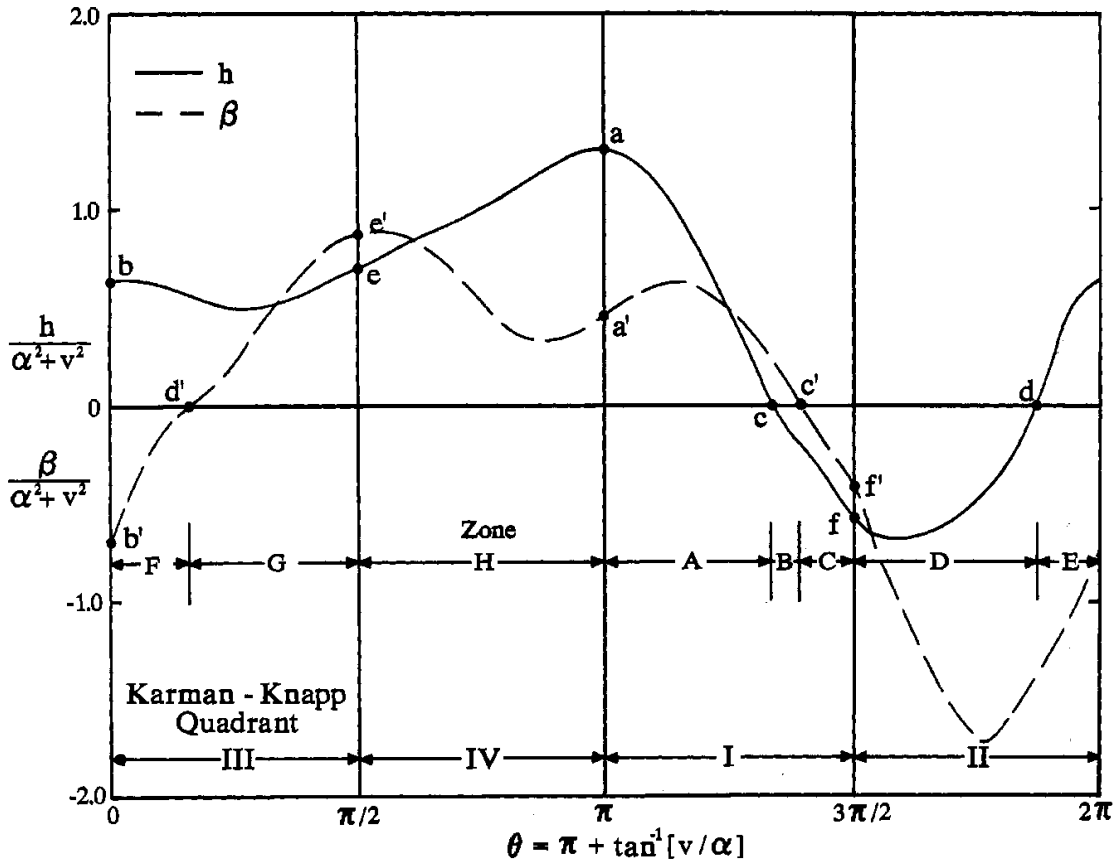


FIGURE 10 Complete head and torque characteristics of radial-flow pump in Suter diagram (after Martin, Reference 8)

involving the design and installation of one or more surge protection devices. In this section various surge protection techniques will be discussed, followed by an assessment of the virtue of each with respect to pumping systems in general.

Critical Parameters for Transients Before discussing surge protection devices, some comments will be made regarding the various pipeline, pump and motor, control valve, flow rate and other parameters that affect the magnitude of the transient. For a pumping system the four main parameters are (1) pump flow rate, (2) pump and motor WR^2, (3) any valve motion, and (4) pipeline characteristics. The pipeline characteristics include piping layout—both plan and profile—pipe size and material, and the acoustic velocity. So-called short systems respond differently than long systems. Likewise, valve motion and its effect, whether controlled valves or check valves, will have different effects on the two types of systems. In this regard both the Main Zone and High Zone are long systems.

The pipeline characteristics—item number (4)—relate to the response of the system to a transient such as pump power failure. Clearly, the response will be altered by the addition of one or more surge protection device or the change of (1) the flow rate, or (2) the WR^2, or (3) the valve motion. Obviously, for a given pipe network and flow distribution there are limited means of controlling transients by (2) WR^2 and (3) valve actuation. If these two parameters cannot alleviate the problem then the pipeline response needs to be altered by means of surge protection devices.

Critique of Surge Protection For pumping systems, downsurge problems have been solved by various combinations of the procedures and devices mentioned above. In many instances local conditions and preferences of engineers have dictated the choice of methods and/or devices. On-line devices such as accumulators and simple surge tanks are quite effective, albeit expensive, solutions. Examples of the various types and deployment of these devices are shown in Figures 11 and 12. One-way surge tanks can also be effective

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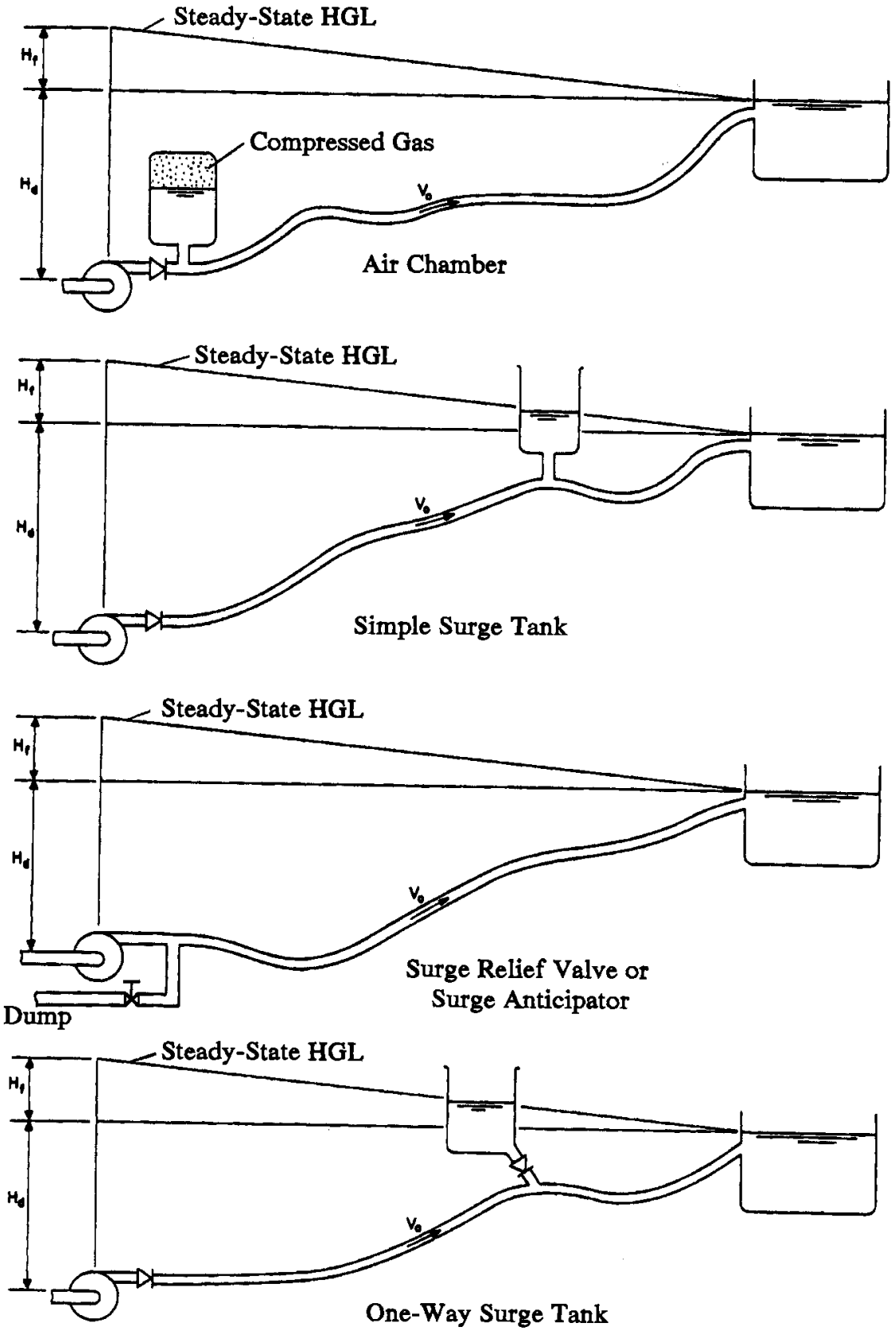


FIGURE 11 Schematic of various surge protection devices for pumping installations

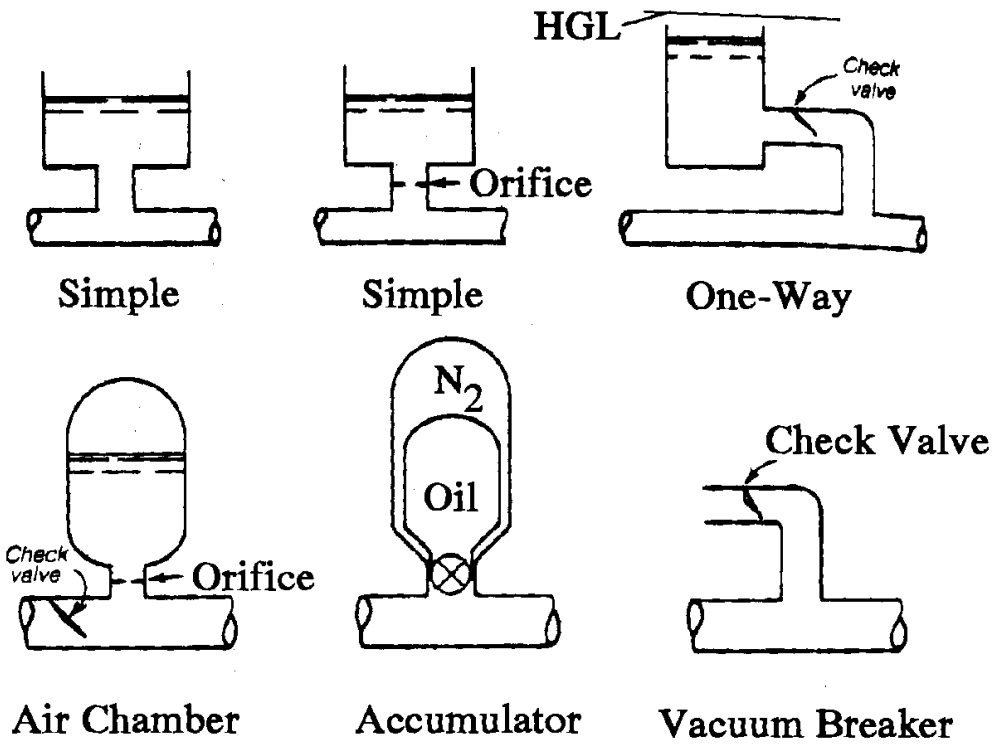


FIGURE 12 Cross-sectional view of surge tanks and gas-related surge protection devices

when judiciously sized and sited. Surge anticipation valves should not be used when there is already a negative pressure problem. Indeed, there are installations where surge anticipation functions of such valves have been deactivated, leaving only the surge relief feature. Moreover, there have been occasions for which the surge anticipation feature aggravated the low-pressure situation by an additional downsurge caused by premature opening of the valve.

Regarding the consideration and ultimate choice of surge protection devices, subsequent to calibration of analysis with test results, evaluation should be given to simple surge tanks or standpipes, one-way surge tanks, and hydropneumatic tanks or air chambers. A combination of devices may prove to be the most desirable and most economical.

The admittance of air into a piping system can be effective, but the design of air-vacuum-valve location and size is critical. If air may be permitted into pipelines careful analysis would have to be done to ensure effective results. The consideration of air-vacuum breakers is a moot point if specifications such as the Ten State Standards limit the pressures to 20 psig.

Surge Protection Control and Devices Pump discharge valve operation. In gravity systems the upsurge transient can be controlled by an optimum valve closure—perhaps two stage, as mentioned by Wylie and Streeter.[11] As shown by Fleming, an optimized closing can solve a waterhammer problem caused by pump power failure if coupled with the selection of a surge protection device.[12] For pump power failure a control valve on the pump discharge can often be of only limited value in controlling the downsurge, as mentioned by Sanks.[13] Indeed, the valve closure can be too sudden, aggravating the downsurge and potentially causing column separation, or too slow, allowing a substantial reverse flow through the pump. It should also be emphasized that an optimum controlled motion for single-pump power failure is most likely not optimum for multiple-pump failure. The use of microprocessors and servomechanisms with feedback systems can be a general solution to optimum control of valves in conjunction with the pump and pipe system. For pump discharge valves the closure should not be too quick to exacerbate downsurge, nor too slow to create a substantial flow back through the valve and pump before closure.

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CHECK VALVES Swing check valves or other designs are frequently employed in pump discharge lines, often in conjunction with slow acting control valves. As indicated by Tullis, a check valve should open easily, have a low head loss for normal positive flow, and create no undesirable transients by its own action.[14] For short systems, a slow responding check valve can lead to waterhammer because of the high reverse flow generated before closure. A spring- or counterweight-loaded valve with a dashpot can (1) give the initial fast response followed by (2) slow closure to alleviate the unwanted transient. The proper selection of the load and the degree of damping is important, however, for proper performance.

Check valve slam is also a possibility from stoppage or failure of one pump of several in a parallel system, or resulting from the action of an air chamber close to a pump undergoing power failure. Check valve slam can be reduced by the proper selection of a dashpot.

SURGE ANTICIPATOR VALVES AND SURGE RELIEF VALVES A surge anticipation valve (Figure 11), frequently installed at the manifold of the pump station, is designed to open initially under (1) pump power failure, or (2) the sensing of underpressure, or (3) the sensing of overpressure, as described by Lescovitch.[15] On the other hand, the usual type of surge relief valve opens quickly upon sensing an overpressure, then closes slowly, as controlled by pilot valves. The surge anticipation valve is more complicated than a surge relief valve in that it not only embodies the relief function at the end of the cycle, but also has the element of anticipation. For systems for which water column separation will not occur, the surge anticipation valve can solve the problem of upsurge at the pump due to reverse flow or wave reflection, as reported in an example by White.[16] An example of a surge relief valve only is provided by Weaver.[17] For systems for which water column separation will not occur, Lundgren provides charts for simple pipeline systems.[18]

As reported by Parmakian surge anticipation valves can exacerbate the downsurge problem inasmuch as the opening of the relief valve aggravates the negative pressure problem.[19,20,21] Incidents have occurred involving the malfunctioning of a surge anticipation valve, leading to extreme pressures because the relief valve did not open.

PUMP BYPASS In shorter low-head systems a pump bypass line can be installed in order to allow water to be drawn into the pump discharge line following power failure and a downsurge. As explained by Wylie and Streeter, there are two possible bypass configurations.[11] The first involves a control valve on the discharge line and a check valve on the bypass line between the pump suction or wet well and the main line. The check valve is designed to open subsequent to the downsurge, possibly alleviating column separation down the main line. The second geometry would reverse the valve locations, having a control valve in the bypass and a check valve in the main line downstream of the pump. The control would open on power failure, again allowing water to bypass the pump into the main line.

OPEN (SIMPLE) SURGE TANK A simple on-line surge tank or standpipe (Figures 11 and 12) can be an excellent solution to both upsurge and downsurge problems. These devices are quite common in hydroelectric systems where suitable topography usually exists. They are practically maintenance-free, available for immediate response as they are on line. For pumping installations open simple surge tanks are rare because of height considerations and the absence of high points near most pumping stations. As mentioned by Parmakian, simple surge tanks are the most dependable of all surge protection devices.[19] One disadvantage is the additional height to allow for pump shutoff head. Overflowing and spilling must be considered, as well as the inclusion of some damping to reduce oscillations. As stated by Kroon *et al.* (1984) the major drawback to simple surge tanks is their capital expense.[22]

ONE-WAY SURGE TANK The purpose of a one-way surge tank (Figures 11 and 12) is to prevent initial low pressures and potential water column separation by admitting water into the pipeline subsequent to a downsurge. The tank is normally isolated from the pipeline by one or more lateral pipes in which there is one or more check valves to allow flow into the pipe if the HGL is lower in the pipe than the elevation of the water in the open tank. Under normal operating conditions the higher pressure in the pipeline keeps the check

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valve closed. The major advantage of a one-way surge tank over a simple surge tank is that it does not have to be at the HGL elevation as required by the latter. It has the disadvantage, however, on only combating initial downsurges, and not initial upsurges. One-way surge tanks have been employed extensively by the U. S. Bureau of Reclamation in pump discharge lines, principally by the instigation of Parmakian, the originator of the concept.[19] Another example of the effective application of one-way surge tanks in a pumping system is reported by Martin.[23]

Considerations for design are (1) location of high points or knees of the piping, (2) check valve and lateral piping redundancy, (3) float control refilling valves and water supply, and other appurtenances. Maintenance is critical in order to insure the operation of the check valve(s) and tank when needed.

AIR CHAMBER (HYDROPNEUMATIC SURGE TANK) If properly designed and maintained, an air chamber (Figures 11 and 12) can alleviate both negative and positive pressure problems in pumping systems. They are normally located within or near the pumping station where they would have the greatest effect. As stated by Fox[24] and others, an air chamber solution may be quite effective in solving the transient problem, but quite expensive. Air chambers have the advantage that the tank, sometimes multiple, can be mounted either vertically or horizontally. The principal criteria are available water volume and air volume for the task at hand.

For design, consideration must be given to compressed air supply, water level sensing, sight glass, drains, pressure regulators, and possible freezing. Frequently, a check valve is installed between the pump and the air chamber. Since the line length between the pump and air chamber is usually quite short, check valve slamming may occur, necessitating the consideration of a dashpot on the check valve to cushion closure.

The assurance of the maintenance of air in the tank is essential, usually 50% of tank volume, otherwise the air chamber can be quite ineffective. An incident occurred at a raw water pumping plant where an air chamber became waterlogged due to the malfunctioning of the compressed air system. Unfortunately, pump power failure occurred at the same time, causing water column separation and waterhammer, leading to pipe rupture.

AIR-VACUUM AND AIR-RELEASE VALVES Another method for preventing subatmospheric pressures and vapor cavity formation is the admittance of air from air-vacuum valves (vacuum breakers, Figure 12) at selected points along the piping system. Proper location and size of air-vacuum valves can prevent water column separation and reduce waterhammer effects, as calculated and measured by Martin.[25] The sizing and location of the valves are critical, as stated by Kroon et al.[22] In fact, as reported by Parmakian the inclusion of air-vacuum valves in a pipeline did not eliminate failures.[20,21] Unless the air-vacuum system is properly chosen substantial pressures can still occur due to the compression of the air during resurge, especially if the air is at very low pressures within the pipeline when admitted. Moreover, the air must be admitted quickly enough to be effective.

As shown by Fleming, vacuum breakers can be a viable solution.[12] The advantage of an air-vacuum breaker system, which is typically less expensive than other measures such as air chambers, must be weighed against the disadvantages of air accumulation along the pipeline and its subsequent removal. Maintenance and operation of valves is critical in order for assurance of valve opening when needed. Air removal is often accomplished with a combined air release air-vacuum valve. For finished water systems the admittance of air is not a normal solution and must be evaluated carefully. Moreover, air must be carefully released so that no additional transient is created.

FLYWHEEL Theoretically, a substantial increase in the rotating inertia (WR^2) of a pump-motor unit can greatly reduce the downsurge inasmuch as the machine will not decelerate as rapidly. Typically, the motor may constitute from 75 to 90% of the total WR^2. Additional WR^2 by the attachment of a flywheel will reduce the downsurge. As stated by Parmakian, a 100% increase in WR^2 by the addition of a flywheel may add up to 20% to the motor cost.[19] He further states that a flywheel solution is only economical in some marginal cases. Flywheels are usually an expensive solution, mainly useful only for short

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systems. A flywheel has the advantage of practically no maintenance, but the increased torque requirements for starting must be considered.

UNINTERRUPTED POWER SUPPLY (UPS) The availability of large uninterrupted power supply systems are of potential value in preventing the primary source of waterhammer in pumping; that is, the generation of low pressures due to pump power failure. For pumping stations with multiple parallel pumps, a UPS system could be devised to maintain one or more motors while allowing the rest to fail, inasmuch as there is a possibility of maintaining sufficient pressure with the remaining operating pump(s). The solution usually is expensive, however, with few systems installed.

DESIGN CONSIDERATIONS

Any surge or hydraulic transient analysis is subject to inaccuracies due to incomplete information regarding the systems and its components. This is particularly true for a water distribution system with its complexity; presence of pumps, valves, tanks, and so on; and some uncertainty with respect to initial flow distribution. The ultimate question is how all of the uncertainties combine in the analysis to yield the final solution. There will be offsetting effects and a variation in accuracy in terms of percentage error throughout the system. Some of the uncertainties are as follows.

The simplification of a pipe system, in particular a complex network, by the exclusion of pipes below a certain size and the generation of equivalent pipes surely introduces some error, as well as the accuracy of the steady-state solution. However, if the major flow rates are reasonably well known, then deviations for the smaller pipes is probably not too critical. As mentioned earlier incomplete pump characteristics, especially during reverse flow and reverse rotation, introduce calculation errors. Valve characteristics that must be assumed rather than actual are sources of errors, in particular the response of swing check valves and pressure-reducing valves. The analysis is enhanced if the response of valves and pumps from recordings can be put in the computer model.

For complex pipe network systems it is difficult to assess uncertainties until much of the available information is known. Under more ideal conditions that occur with simpler systems and laboratory experiments, one can expect accuracies when compared to measurement on the order of 5 to 10%, sometimes even better. The element of judgment does enter into accuracy. Indeed, two analyses could even differ by this range because of different assumptions with respect to wave speeds, pump characteristics, valve motions, system schematization, and so on. It is possible to have good analysis and poorer analysis, depending upon experience and expertise of the user of the computer code. This element is quite critical in hydraulic transients. Indeed, there can be quite different results using the same code.

Computer codes, which normally are based upon the method of characteristics (MOC), are invaluable tools for assessing the response of systems to changes in surge protection devices and their characteristics. Obviously, the efficacy of such an approach is enhanced if the input data and network schematization is improved via calibration. Computer codes have the advantage of investigating a number of options as well as optimizing the sizing of surge protection devices. The ability to calibrate a numerical or analysis code to a system certainly improves the determination of the proper surge protection. Otherwise, if the code does not reasonably well represent a system surge protection devices can either be inappropriate or under- or oversized.

Computer codes that do not properly model the formation of vapor pockets and subsequent collapse can cause considerable errors. Moreover, there is also uncertainty regarding any free or evolved gas coming out of solution. The effect on wave speed is known, but this influence cannot be easily addressed in an analysis of the system. It is simply another possible uncertainty.

Even for complicated systems such as water distribution networks hydraulic transient calculations can yield reasonable results when compared to actual measurements provided that the entire system can be properly characterized. In addition to the pump,

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motor, and valve characteristics there has to be sufficient knowledge regarding the piping and flow demands. An especially critical factor for a network is the schematization of the network; that is, how is a network of thousands of pipes simplified to one suitable for computer analysis, say hundreds of pipes, some actual and some equivalent. According to Thorley, a network with loops tends to be more forgiving regarding waterhammer because of the dispersive effect of many pipes and the associated reflections.[26] On the other hand, Karney and McInnis show by a simple example that wave superposition can cause amplification of transients.[27] Since water distribution networks themselves have not been known to be prone to waterhammer as a rule, there is meager information as to simplification and means of establishing equivalent pipes for analysis purposes. Large municipal pipe networks are good examples wherein the schematization and the selection of pipes characterizing the networks need to be improved in order to represent the system better.

NEGATIVE PRESSURES AND WATER COLUMN SEPARATION

For finished water transmission and distribution systems the application of 20 psig (1.38 barg) as a minimum pressure to be maintained under all conditions should prevent column separation from occurring provided analytical models have sufficient accuracy. Although water column separation and collapse is not common in large networks, it does not mean that the event is not possible. The modeling of water column separation is clearly difficult for a complicated network system. Water column separation has been analytically modeled with moderate success for numerous operating pipelines. Clearly, not only negative pressures, but also water column separation, are unwanted in pipeline systems, and should be eliminated by installation of properly designed surge protection devices.

If the criterion of a minimum pressure of 20 psig (1.38 barg) is imposed then the issue of column separation and air-vacuum breakers are irrelevant, except for prediction by computer codes. Aside from research considerations, column separation is simulated for engineering situations mainly to assess the potential consequences. If the consequences are serious, as they often are in general, either operational changes or more likely surge protection devices are designed to alleviate column separation. For marginal cases of column separation the accuracy of pressure prediction becomes difficult. If column separation is not to be allowed and the occurrence of vapor pressure can be adequately predicted, then the simulation of column separation itself is not necessary.

Some codes do not simulate water column separation, but instead only maintain the pressure at cavity location at vapor pressure. The results of such an analysis are invalid, if indeed an actual cavity occurred, at some time subsequent to cavity formation. This technique is only useful to know if a cavity could have occurred, as there can be no assessment of the consequences of column separation. The inability of any code to model water column separation has the following implications (1) the seriousness of any column separation event, if any, can not be determined, and (2) once vapor pressure is attained, the computation model loses its ability to predict adequately system transients. If negative pressures below 20 psig (1.38 barg) are not to be allowed the inability of a code to assess the consequences of column separation and its attendant collapse is admittedly not so serious. The code need only flag pressures below 20 psig (1.38 barg) and negative pressures, indicating if there is a need for surge protection devices.

The ability of any model to properly simulate water column separation depends upon a number of factors. The principal ones are:

- Accurate knowledge of initial flow rates.
- Proper representation of pumps, valves, and piping system.
- Allowing a vapor pocket to form, grow, and collapse.
- Maintenance of vapor pressure within cavity while it exists.
- Determination of volume of cavity at each time step.
- Collapse of cavity at the instant the cavity volume is reduced to zero.

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TIME CONSTANTS FOR HYDRAULIC SYSTEMS

- Elastic Time Constant : $$t_e = \frac{2L}{a} \quad (16)$$

- Flow Time Constant : $$t_f = \frac{LV_o}{gH_o} \quad (17)$$

- Pump and Motor Inertia Time Constant: $$t_m = \frac{I\omega_R}{T_R} = \frac{I\omega_R^2}{\rho g Q_R H_R \eta_{RT}} \quad (18)$$

- Surge Tank Oscillation Inelastic Time Constant: $$t_s = 2\pi\sqrt{\frac{L_T}{g}\frac{A_t}{A_T}} \quad (19)$$

CASE STUDIES—EXPERIENCES WITH SURGE PROTECTION DEVICES

For three large water pumping systems with various surge protection devices waterhammer analyses and site measurements have been conducted. The surge protection systems in question are (1) one-way and simple surge tanks, (2) an air chamber, and (3) air-vacuum breakers.

Case Study with One-Way and Simple Surge Tanks A very large pumping station has been installed and commissioned to deliver water over a distance of over 30 km. Three three-stage centrifugal pumps run at a synchronous speed of 720 rpm, with individual rated capacities of 1.14 m^3/s, rated heads of 165 m, and rated power of 2090 kW. Initial surge analysis indicated potential water column separation. The surge protection system was then designed with one-way and simple surge tanks as well as air-vacuum valves strategically located.

The efficiency of these various surge protection devices was assessed from site measurements. Measurements of pump speed, discharge valve position, pump flow rate, and pressure at seven locations were conducted under various transient test conditions. The site measurements under three-pump operation allowed for improvement of hydraulic transient calculations for future expansion to four and five pumps. Figure 13 illustrates the profile of the ground and the location of the three pairs of surge tanks. The first and second pairs of surge tanks are of the one-way (feed tank) variety, while the third pair are simple open on-line tanks.

Pump trip tests were conducted for three-pump operation with cone valves actuated by the loss of motor power. For numerical analysis a standard computer program applying the method of characteristics was employed to simulate the transient events. Figure 14 shows the transient pressures for three-pump power failure. The transient pressures agree reasonably well for the first 80s. The minimum HGLs in Figure 13 also show good agreement, as well as the comparison of measured and calculated pump speeds in Figure 14. The test program and transient analysis clearly indicated that the piping system was adequately protected by the array of surge tanks inasmuch as there were no negative pressures.

Case Study with Air Chamber Hydraulic transients caused by simultaneous tripping of pumps at the pumping station depicted on Figure 15a were evaluated to assess the necessity of surge protection. Without the presence of any protective devices such as accumulators, vacuum breakers, or surge suppressors, waterhammer with serious consequences was shown to occur due to depressurization caused by the loss of pumping pressure following sudden electrical outage. In the case of no protection a large vapor cavity would occur at the first high point above the pumping station, subsequently collapsing after the water column between it and the reservoir stops and reverses. This phenomenon, called water column separation, can be mitigated by maintaining the pressures above vapor pressure.

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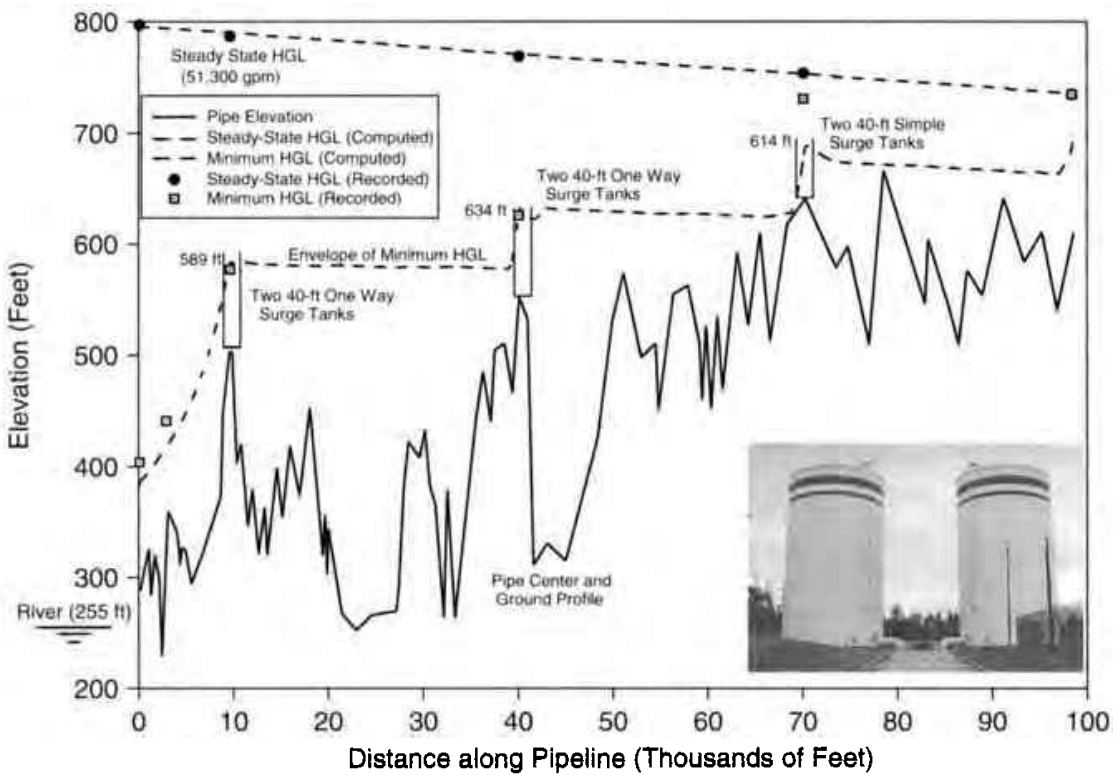


FIGURE 13 Case study of pump power failure at pumping station with three pairs of surge tanks—two pairs one way and one pair simple surge tanks (after Martin, Reference 23)

The efficacy of the 11.6 m (38 ft) diameter air chamber shown in Figure 15a was investigated analytically and validated by site measurements for three-pump operation. The envelope of the minimum HGL drawn on Figure 15a shows that all pressures remained positive. Figure 15b compares the site measurement with the calculated pressures obtained by a standard waterhammer program utilizing MOC.

Case Study with Air-Vacuum Breaker Air-inlet valves or air-vacuum breakers are frequently installed on liquid piping systems and cooling water circuits for the purpose of (1) eliminating the potential of water column separation and any associated waterhammer subsequent to vapor pocket collapse, (2) protecting the piping from an external pressure of nearly a complete vacuum, and (3) providing an elastic cushion to absorb the transient pressures.

A schematic of the pumping and piping system subject to the field test program is shown in Figure 16a. This system provides the cooling water to a power plant by pumping water from the lower level to the upper reservoir level. There are five identical vertical pumps in parallel connected to a steel discharge pipe 1524-mm (60 in) in diameter. On the discharge piping of each pump there are 460-mm (18 in) diameter swing check valves. Mounted on top of the 1524-mm (60 in) diameter discharge manifold is a 200-mm (8 in) diameter pipe, in which is installed a swing check valve with a counter weight. Air enters the vacuum breaker through the tall riser, which extends to the outside of the pump house.

Transient pressures were measured in the discharge header for simultaneous trip out of three, four, and five pumps. The initial prediction of the downsurge caused by pump power failure was based on the method of characteristics with a left end boundary condition at the pumps, junction boundary condition at the change in diameter of the piping, and a constant pressure boundary condition at the right end of the system.

The predicted pressure head variation in the pump discharge line is shown in Figure 16b for a simulated five-pump trip out. The predicted peak pressure for the five-pump trip out

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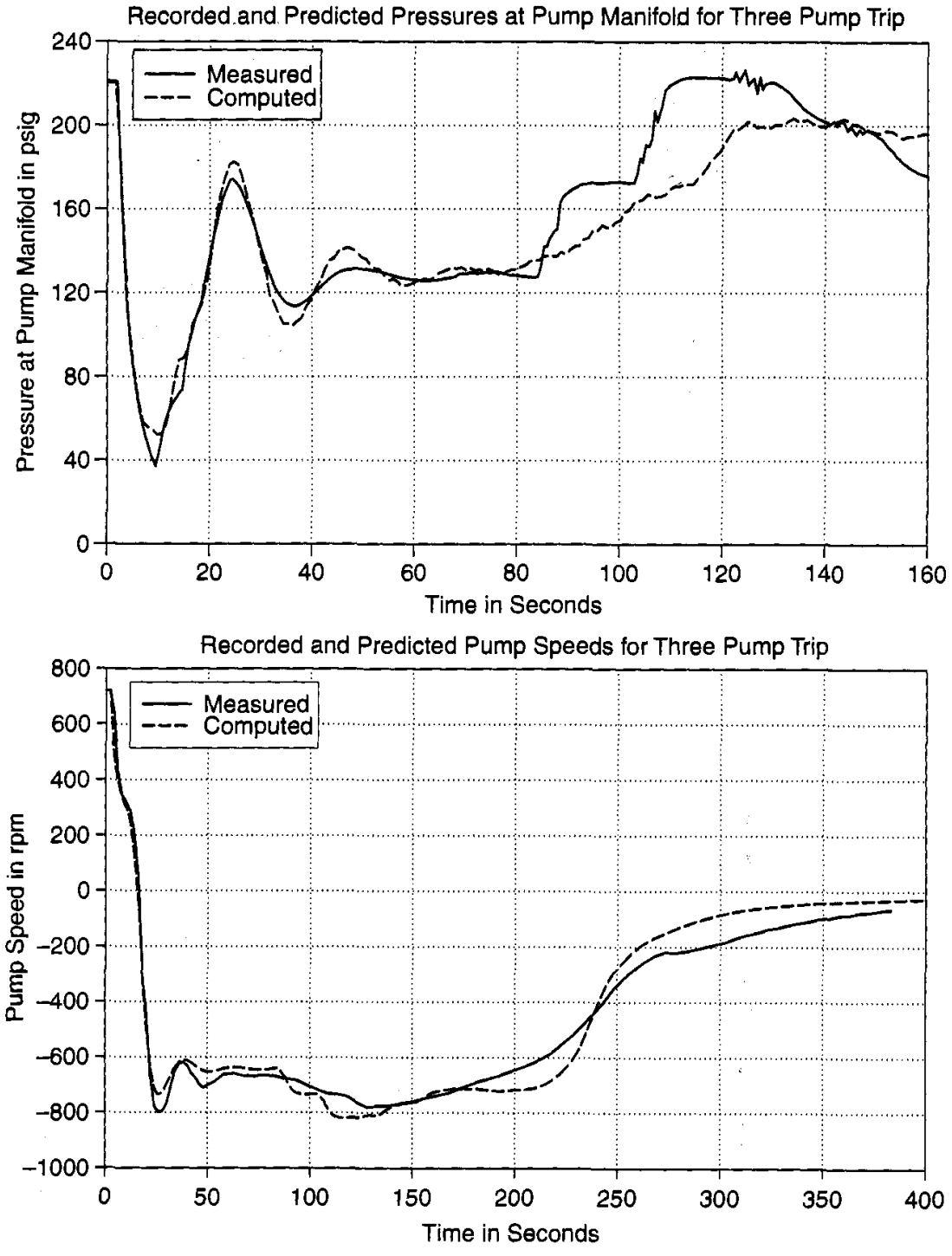


FIGURE 14 Case study of pump power failure at pumping station with three pairs of surge tanks—two pairs one way and one pair simple surge tanks (after Martin, Reference 23)

compares favorably with the corresponding measured peak, but the time of occurrence of the peaks and the subsequent phasing vary considerably. Analysis without a vacuum breaker or other protective device in the system predicted waterhammer pressure caused by collapse of a vapor pocket to exceed 2450 kPa (355 psi). The vacuum breaker effectively reduced the peak pressure by 60%. Peak pressures can be adequately predicted by a simplified liquid column, orifice, and air spring system. Water column separation can be eliminated by air-vacuum breakers of adequate size.

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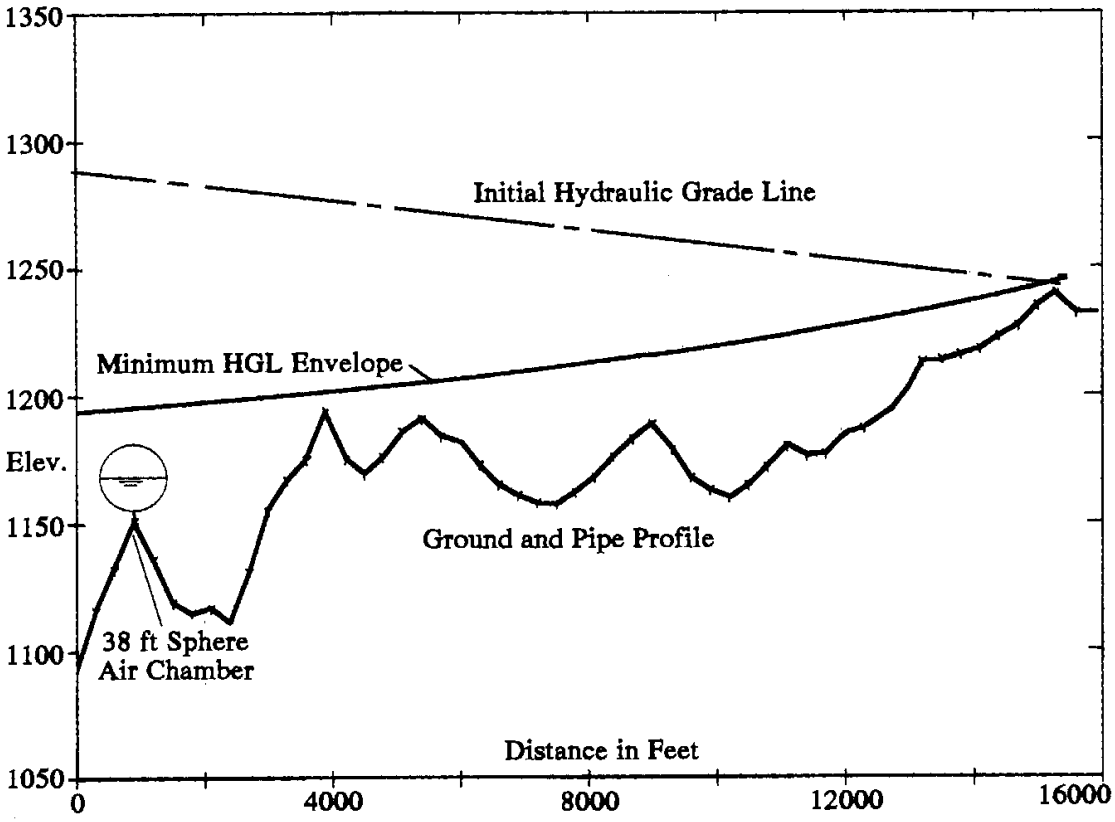


FIGURE 15A Pipe profile and HGL for case study of air chamber performance for raw water supply.

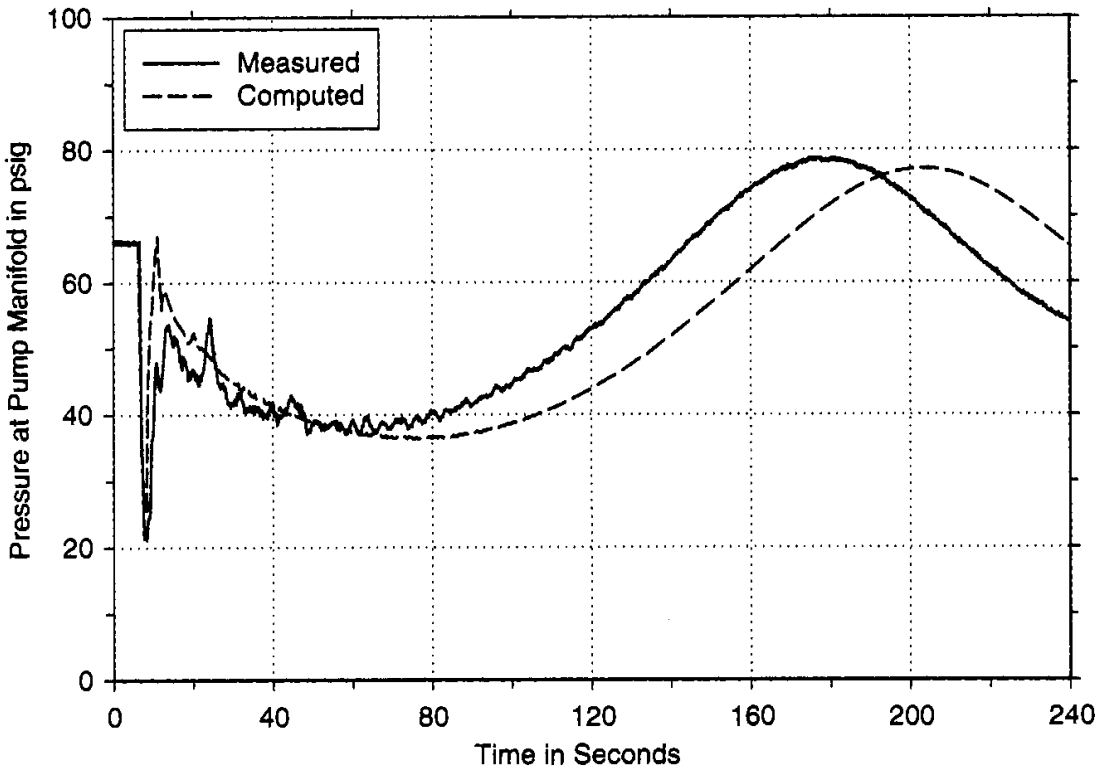


FIGURE 15B Comparison of computed and recorded pressures for case study of air chamber performance for raw water supply

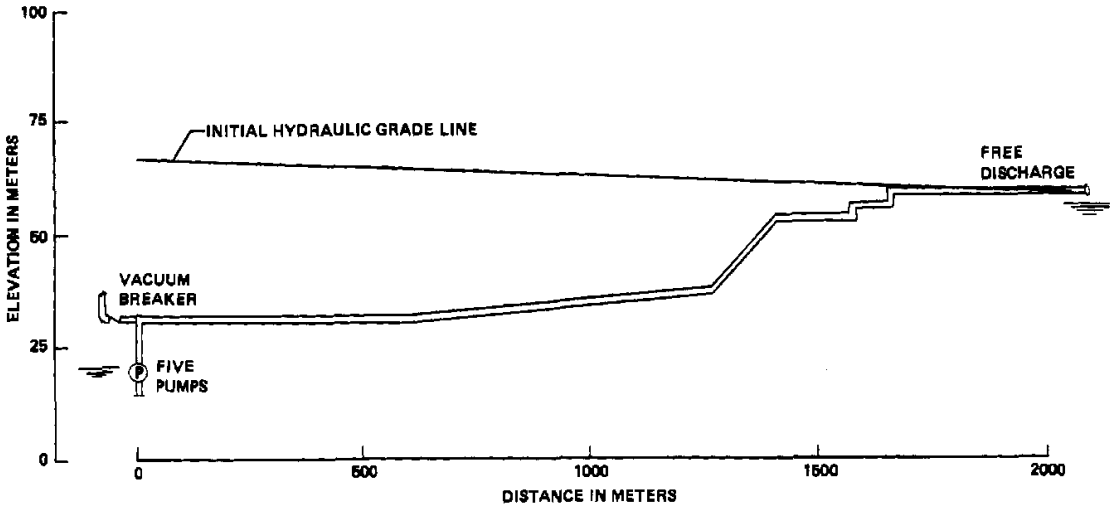


FIGURE 16A Pipe profile for case study of vacuum breaker air chamber performance for raw water supply (after Martin, Reference 25)

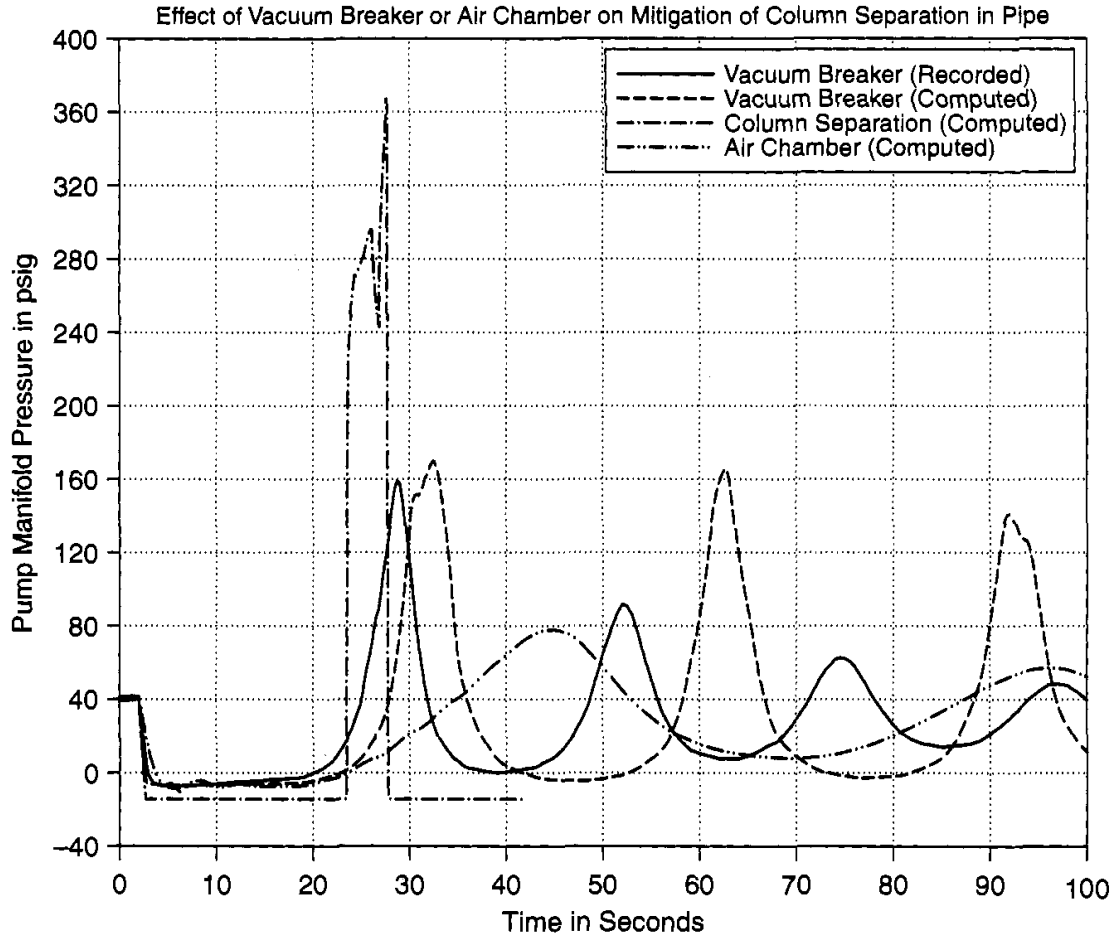


FIGURE 16B Comparison of computed and recorded pressures for case study of vacuum breaker air chamber performance for raw water supply (after Martin, Reference 25)

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SECTION 11.4
MINIMUM FLOW CONTROL SYSTEMS

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DAVID A. KALIX

In many cases, pumps are required to operate over a range of flow rates from a maximum well above the design value down to the minimum that is possible for a given pump size, type, and speed. Guidelines for establishing minimum flow are discussed in Sections 2.1.1 and 2.1.3 for centrifugal pumps, which can be throttled at the discharge to reduce the flow rate. Positive displacement pumps (both reciprocating and rotary), on the other hand, cannot be throttled; so, for the safety of the system and the user, these machines are usually equipped with pressure-actuated relief valves. However, both centrifugal and positive displacement pumps can be equipped with flow-bypass systems (and sometimes variable-speed capability) in order to accommodate varying flow requirements. For centrifugal pumps, which are the focus of this section, such systems act to protect against flow rates falling below the minimum allowable.

In designing centrifugal pumps, engineers strive to develop specific internal geometry that will produce head and flow with low energy loss. Each pump is designed for a specific head versus flowrate for the given impeller speed. The head and flowrate on this family of "characteristic" curves where the energy loss is minimum is known as the Best Efficiency Point (BEP).

In application, pumps spend a significant portion, if not all of their life, operating at conditions other than BEP. This is normal and is due to a combination of system design conditions, available pump designs, and actual plant operating requirements. Low flow related problems occur when pumps continuously operate in or repetitively cycle into flow regions that are significantly below the BEP (for example, 50% of BEP). In these low-flow regions, pump and system component performance and longevity can be adversely affected.

Low flow problems are known to be worst for large high-energy pumps (for example, boiler feedwater), for pumps handling hot liquids, for pumps that handle liquids that have solid particles, and for pumps for low net positive suction head (NPSH) service. A well-designed minimum flow control system can establish an environment that will substantially improve pump and system performance.

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FACTORS AFFECTING LOW FLOW RATE PUMP OPERATION

Collectively, the following broadly classed factors have been recognized as the major contributors to low flow pump problems.

Thermal Factors The increase in temperature of the liquid within the pump is directly related to the pump's efficiency. The energy that is available to heat the flowing liquid and the pump casing is basically the difference between the power input to the pump (brake horsepower) and the useful work done by the pump (liquid horsepower). At low flow conditions, centrifugal pumps are very inefficient and a significant amount of input energy is lost and heats the liquid and the pump assembly. Refer to Section 2.1.3 and Chap. 15 for more discussion on thermal effects.

Hydraulic Instabilities When the pump is operating significantly below the BEP, flow streamlines (i.e., patterns) within the pump change considerably from the rated design streamlines. Fluid eddies are most likely to develop at the inlet and discharge of the impeller resulting in flashing, cavitation, and shock waves that often produce vibration and serious component erosion. This phenomenon is classically known as internal recirculation. It can occur at the pump inlet (suction) and discharge. Refer to Section 2.1.1 and 2.1.3 for a more in-depth discussion on this topic.

Mechanical Loads As the flowrate through the pump decreases, steady state loads increase and superimposed dynamic cyclic loads appear radially and axially on the impeller and shaft. The dynamic cyclic component increases significantly when recirculation within the pump occurs. Bearing damage, shaft and impeller breakage, and rubbing wear on casing, impeller and wear rings can occur. See Sections 2.1.3 and 2.1.4 for discussion of this subject.

Axial-flow and mixed-flow pumps with high specific speed produce comparatively higher head and take comparatively more power at low flow. A bypass system may be necessary not only to reduce component loading and stress but also to prevent motor overload. See Section 2.1.3 and Section 11.1 for discussion.

Abrasive Fluids Liquids containing a large amount of abrasive particles, such as sand or ash, must flow continuously through the pump. If flow decreases, the particles can circulate inside the pump passages and quickly erode the impeller casing, wear rings, and shaft.

ESTABLISHING MINIMUM PUMP FLOW REQUIREMENTS

The bypass system designer must know the minimum pump flow specified by the pump manufacturer in order to properly design a bypass system. The four previously discussed topics should be evaluated in detail by the pump manufacturer to establish the minimum flowrate specification. Minimum flowrate specifications are generally established through a combination of analytical and experimental techniques coupled with field performance data.

Thermal Considerations The maximum allowable temperature rise of the pump is primarily based on two points: the permissible pump casing and shaft thermal growth, and the flash point temperature of the pumpage. Pump manufacturers use analytical, laboratory and field data to validate their thermal analysis to ensure that pumps do not seize within the allowable temperature operating ranges. Refer to Section 2.1.3 for temperature rise calculations.

Applications involving extremely high or low temperature fluids may require more in-depth analysis to determine if individual component thermal growth is the limiting factor in determining minimum flowrate. Additionally, certain chemicals, which polymerize or solidify at particular temperatures, may establish the minimum flowrate specification. Chap. 15 and Section 2.1.3 provide more detail on this subject.

Hydraulic Considerations The minimum bypass flow requirement for most pumps is based on minimum continuous stable flowrate, a hydraulic criterion, rather than a

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temperature rise. Pump internal recirculation will occur at both the impeller inlet and impeller outlet as flowrates are reduced. Internal recirculation will occur at flowrates well above those that cause temperature concerns. Refer to Section 2.1.3 for a detailed evaluation of this topic.

Mechanical Considerations It is necessary to know how head, radial thrust, axial thrust, and power vary with capacity before deciding on minimum allowable flow. Bearing capacity, motor rating, and stresses in drive and driven components are important influences.

Abrasive Wear Considerations Relatively high bypass flowrates may be required to protect the pump against abrasives in the liquid. Heavy wear can occur at flows below 85% of the best efficiency point. The designer must establish the minimum pump flow specification using the pump manufacturer's recommendation and his experience with comparable pumps and liquid/solid mixtures.

MINIMUM FLOW CONTROL SYSTEM DESIGN FACTORS

Pump Size Capacity, power, specific speed, and suction specific speed are all factors that must be examined when designing a bypass system. These factors have a direct impact on the cost of building and operating the bypass system. Use of a continuous bypass system will require an even larger pump and driver to supply both the process and bypass flow requirements simultaneously.

Discharge Pressure High discharge pressures result in high head loss in bypass valves, components, and lines. Liquids that can flash and cavitate demand special precautions to minimize damage in valving, orifices, and piping.

Available Heat Sink Bypass flow must be reintroduced into the system far enough upstream to prevent progressive temperature buildup or flow disturbance in the pump suction. This may mean a simple discharge back to an uninsulated inlet line or discharge to a receiving tank or cooler with enough area and enough inflow of cool liquid to handle the thermal load. Bypass flow can discharge into a deaerator storage tank, a condenser, a flash tank, or a cooling pond. Elevation, distance, and pressure inside the receiving tank are also factors, as is the fact that the interior must be available for inspection and for repair of spargers, spray or distribution pipes, orifices, and backpressure regulators.

Pump Design A pump's design and materials of construction often affect the minimum allowable percentage of flow. With thermal effects, pumps vary in the length of time that they will tolerate shut-off or low flow. This is important in designing the bypass system valves, instrumentation, and controls.

Hydraulic effects at low flow are most apparent in high-energy pumps. The pump manufacturer should state the continuous minimum hydraulically acceptable flow for a given pump—and how it was determined. With axial-flow pumps, the shape of the pump head curve may be a factor in selecting the required bypass flow percentage. If feasible, a witness shop test of the pump should be specified to demonstrate and verify minimum flow recommendations.

Liquid Pumped Liquids that flash and cavitate generally required a high bypass flow percentage. Examples are liquids near the boiling point or at high pressure. Abrasives in the liquid may require more bypass than would be needed for thermal reasons alone.

Energy Costs A high energy cost to operate the pump requires careful consideration of bypass system design. The evaluation should compare equipment installation costs, maintenance, and energy costs for various bypass configurations. Figure 1 shows the annual pumping costs for a continuous bypass type system based on bypass flow, pressure, and energy rates. The example shows a pump with a discharge head of 500 ft (152 m)

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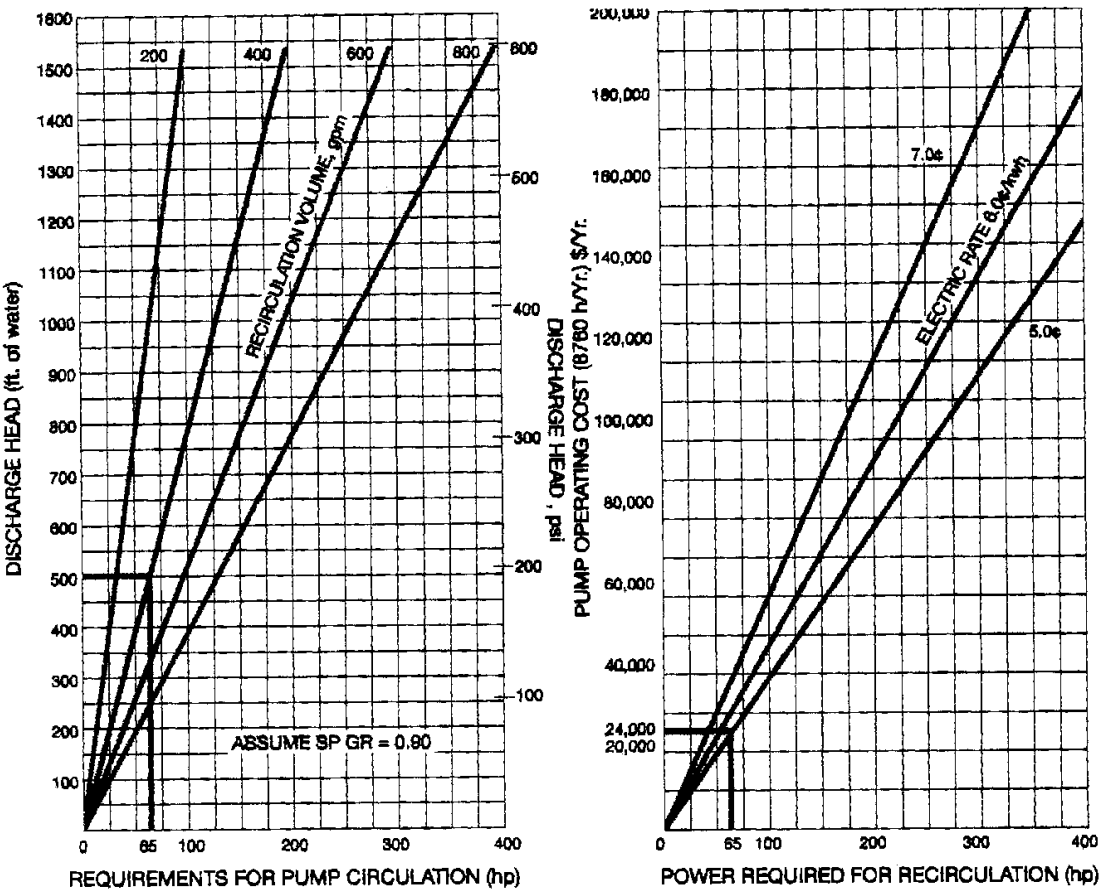


FIGURE 1 Annual pumping cost estimate for continuous bypass systems (metric conversions: ft × 0.3048 = m, gpm × 0.277 = m^3/h, hp × 0.746 = W)

and a bypass flow requirement of 400 gpm (91 m^3/h). Based on an energy cost of 5.0 cents/KWH, the annual cost for continuous bypass is $24,000. An automatic bypass system will only open the bypass when process flow demand is low. The total design life energy costs of a continuous bypass system can easily exceed the hardware costs of an automatic bypass system.

Noise Considerations Bypass systems can easily exceed OSHA requirements for occupied spaces if not designed properly. High-pressure drops and high-fluid velocity increase noise. Multistage pressure reduction, heavy wall pipe, insulation, and silencers will all combine to reduce noise to acceptable levels. See Chap. 10 for a further discussion of this topic.

Process System Design and Operational Expectations Bypass system design will depend on the plant design life and the expected process operational requirements. For example, a swing-loaded electric power plant will have far different pump operating requirements than a low-pressure emergency fire water system. The bypass system designer must evaluate installation, maintenance, and operating costs for the life of the process system with the expected utilization. If the actual operating conditions differ significantly from the design, the configuration of the bypass system should be reevaluated. For example, a process designed for normal operation with one pump at 75% of maximum capacity may put severe demands on the bypass system if the pump is operated continuously at 20% of maximum capacity.

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LOW FLOW PROTECTION SYSTEMS

Bypass Systems—Types/Design Considerations

CONTINUOUS BYPASS SYSTEMS As the title implies, continuous bypass systems provide continuous flow whenever the pump is running, regardless of the process demand. Figure 2 illustrates a simple system, with a bypass line branching off the pump discharge upstream of the main line check valve and containing a fixed orifice dimensioned by analysis to provide minimum required pump flow. The bypass line discharges into a reservoir that is at a lower pressure than the pump discharge. The bypass line can also discharge directly into the pump supply line. However, the piping system design must ensure that the bypass liquid temperature does not increase to an unacceptable level. Additionally, vapor bubbles formed in the bypass by the pressure reduction process may be introduced into the pump. This will affect pump performance and longevity. Locating the bypass branch-off before the discharge check valve as shown keeps backflow from the process or from a parallel operating pump from going back to the receiving tank or back through the pump during a pump shutdown. Consideration should also be given to installing a check valve in the bypass line.

The size of the bypass pipe depends on flow and piping configuration. If the pressure drop through the orifice results in flashing flow, the orifice should be located at the end of the bypass piping and should discharge directly into the larger receiving tank or larger downstream piping. The discharge flow should be directed so it does not impinge on the walls of the receiving tank but rather into the liquid to absorb the force of cavitation implosion. Valves and fittings located immediately downstream of the orifice may be damaged by cavitation. Full ported gate or ball valves are less susceptible to damage. Waterhammer and erosion are not problems in well-designed continuous bypass systems.

The pump and prime mover must be sized to simultaneously supply both the bypass flow and the maximum process flow at the required pump discharge pressure.

AUTOMATIC BYPASS SYSTEMS Automatic recirculation systems control the bypass flow in relation to the process flow. The sum of the process and bypass flowrates will always exceed the minimum flowrate specified to protect the pump. Two essential elements of these systems are a device to measure the process flow and a device to control the bypass flow. Compared with continuous bypass, these systems reduce energy consumption and pump horsepower requirements.

The bypass flow can be regulated by either "Modulated" or "On-Off" control. When bypass flow is "Modulated," the bypass flow is inversely proportional to the process flow.

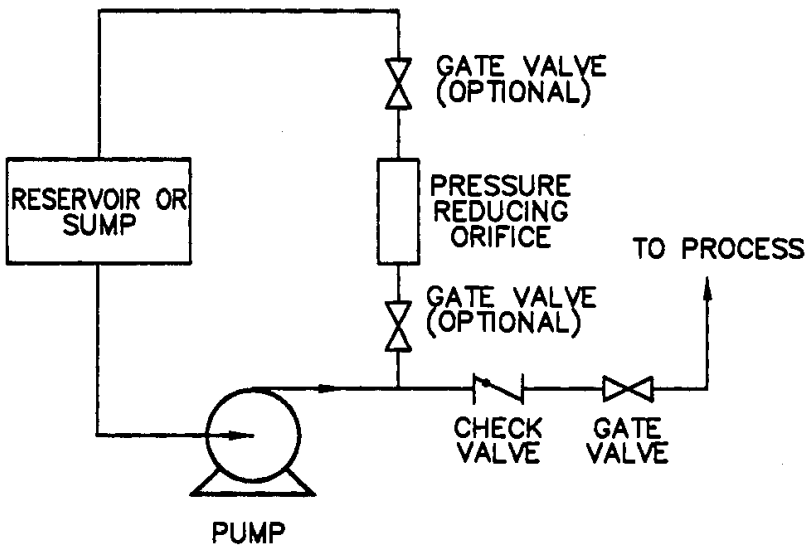


FIGURE 2 Typical continuous bypass system

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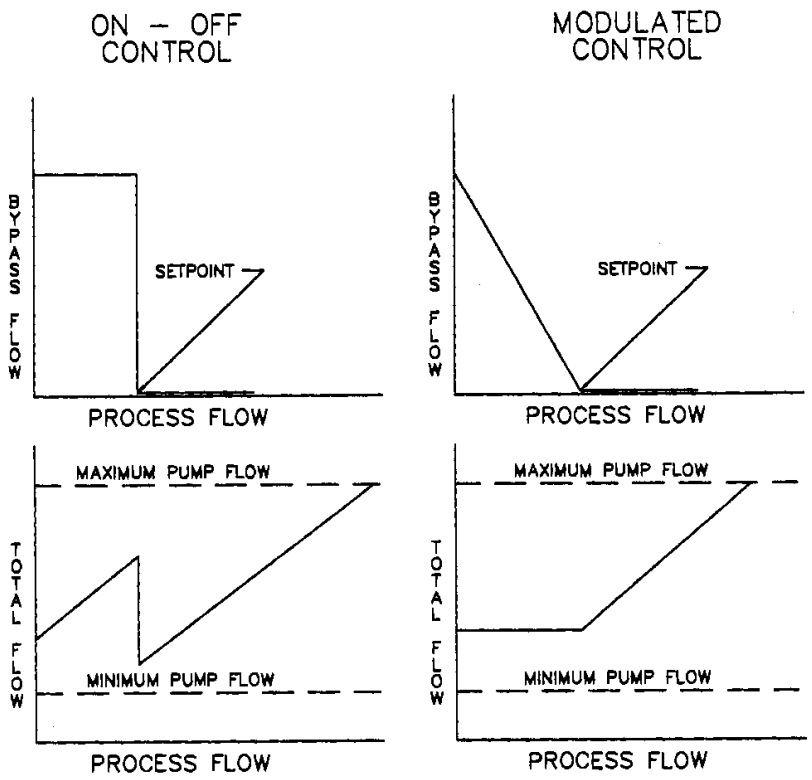


FIGURE 3 Comparison of On-Off versus Modulated automatic bypass flow control

The bypass is shut when process flow is greater than the control setpoint (i.e., minimum flowrate required), opens more as the process flow is reduced, and is full open when the process flow is zero. With "On-Off" bypass control, the bypass is open fully when process flow is below the setpoint and is shut when the process flow is above the setpoint.

Figure 3 illustrates these control methods and the effect on total pump flow. The total pump flow is the sum of the process and bypass flows; it is the flow that enters the pump suction. The "On-Off" control graph shows that there is a step change in the total flow when the bypass opens or closes. The "Modulated" control regulates the total flow smoothly when the process flow is below the setpoint. Either method will protect the pump from damage due to low flow. However, "On-Off" control may produce large pump discharge pressure variations depending on the shape of pump head curve. Rapid pump flow changes may produce damaging hydraulic or mechanical shocks. This will reduce pump life and increase maintenance. Modulated bypass control is required when the minimum flowrate specification exceeds approximately 40% of the rated pump capacity. If "On-Off" bypass control is used above this point, the pump may exceed its capacity rating with resultant pressure pulsation and unsteady flow.

CONTROL LOOP SYSTEMS A typical automatic bypass system, based on an instrumented flow control loop, is shown in Figure 4. This system requires a flow meter, a check valve, a bypass control valve, a valve controller and a pressure-reducing orifice.

The flow meter can be placed at the pump inlet or discharge before the bypass tee (as opposed to it having to be placed after the bypass tee in an "On-Off" system). The controller compares the flow meter signal with the setpoint (minimum pump flow required) and operates the bypass control valve.

Single stage control valves are commonly used in low-pressure systems with an orifice for pressure reduction located downstream. However, these are not suitable for high-pressure differentials or for the controlling of cavitation or flashing. Pressure reducing valves with multistage trim, good seat tightness, and low-pressure recovery characteristics are widely used in this service.

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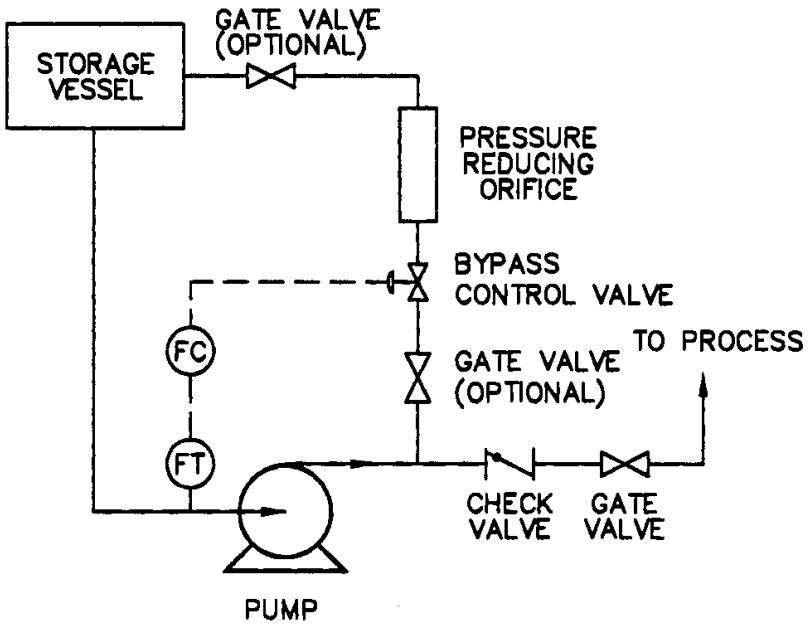


FIGURE 4 Typical automatic bypass system based on flow control

The bypass pressure reducing orifice handles the high velocity and erosive forces of the bypass flow. However, the handling of cavitation and flashing by this fixed restriction is limited. The orifice is sized to operate properly at one bypass flow condition. At low flows, the orifice does not provide the correct pressure reduction as designed.

The check valve is placed in the main line, as shown in Figure 4, to prevent reverse flow through the pump. The pump and motor bearings can be damaged if rotated in reverse.

A typical automatic bypass system based on pump differential head is shown in Figure 5. This system is identical to the flow controlled bypass system except that pump differential head is measured instead of flow. This system can only be used with continuously rising pump head characteristic curves. The system will have to be adjusted for each individual pump characteristic and later adjusted if this characteristic changes over the life of the pump. Refer to Section 11.1, "Throttling Orifices" and Section 11.2, "Centrifugal Pump Bypass."

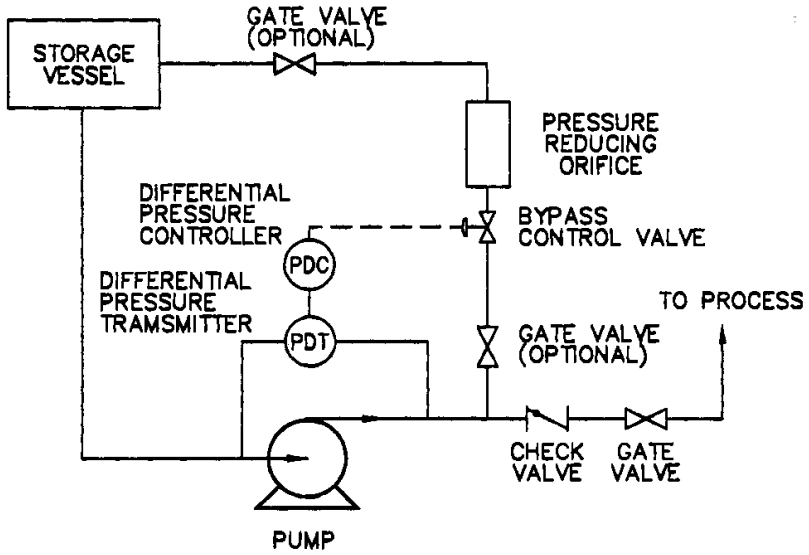


FIGURE 5 Typical automatic bypass system based on pump differential head

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Depending on the bypass valve and control instrumentation selected, these systems can provide either "Modulating" or "On-Off" bypass control. Valves, instrumentation, and controls must be individually sized for the service conditions. All components must be integrated together to provide bypass flow control that meets the pump and process operational design criteria.

The Automatic Recirculation Control (ARC®) Valve This valve provides bypass flow control, pressure reduction, and reverse flow pump protection all within a single unit. This single valve combines the functions of the check valve, pressure reducing orifice, pipe tee, control instrumentation flow meter, and bypass control valve that are all required in the instrumented control loop bypass system. Valves can be designed to provide either "Modulated" or "On-Off" bypass control. The operation of all ARC valves is fundamentally the same. The flow sensing element (check valve disc) responds to the process flow demand and opens or closes the self-contained bypass control valve. Differences in ARC valve design center around bypass valve actuation method, repairability, serviceability, and adjustability. Figure 6 illustrates a typical ARC valve bypass system.

Pilotless Trim Design The simplest of ARC valve designs operate the bypass flow control valve directly by the flow sensing element. The construction of a typical ARC for low-pressure service is shown in Figure 7. The basic operation of this valve is shown in Figure 8. At zero main flow ("A"), the bypass provides full recirculation flow. As main flow increases ("B"), the bypass modulates proportionally closed. At full main flow ("C"), the bypass is shut. The pressure reduction from the pump discharge to the bypass is accomplished by the characterized orifices in the bypass element that is attached to the disc. However, the differential pressure from inlet to bypass imposes static and dynamic forces on the disc assembly. These unbalanced forces interfere with the normal disc motion and limit the maximum differential pressure for pilotless trim. There is an upper differential pressure limit at which this design will not operate satisfactorily and pilot operated bypass control must be used.

Pilot Operated Trim Pilot operated bypass control valves utilize a combined hydraulic-mechanical force to control the bypass flow. Only a relatively small mechanical force is necessary to hydraulically generate the large forces required to operate a bypass flow control valve in a high-pressure system. Figure 9 illustrates a pilot-operated bypass that utilizes a lever connected to the check valve disc to actuate the pilot. Figure 10 illustrates a pilot operated multistage bypass operated by an in-line pilot valve. Pilot-operated valves require more components and seals than unbalanced valves and are therefore more costly.

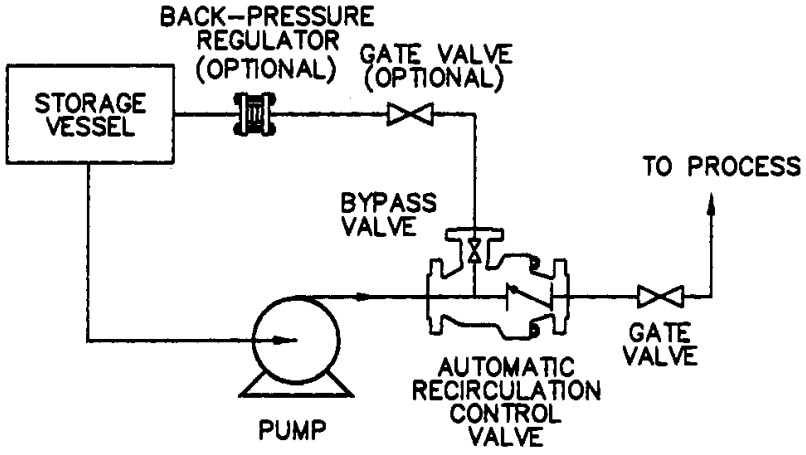


FIGURE 6 Typical ARC valve bypass system

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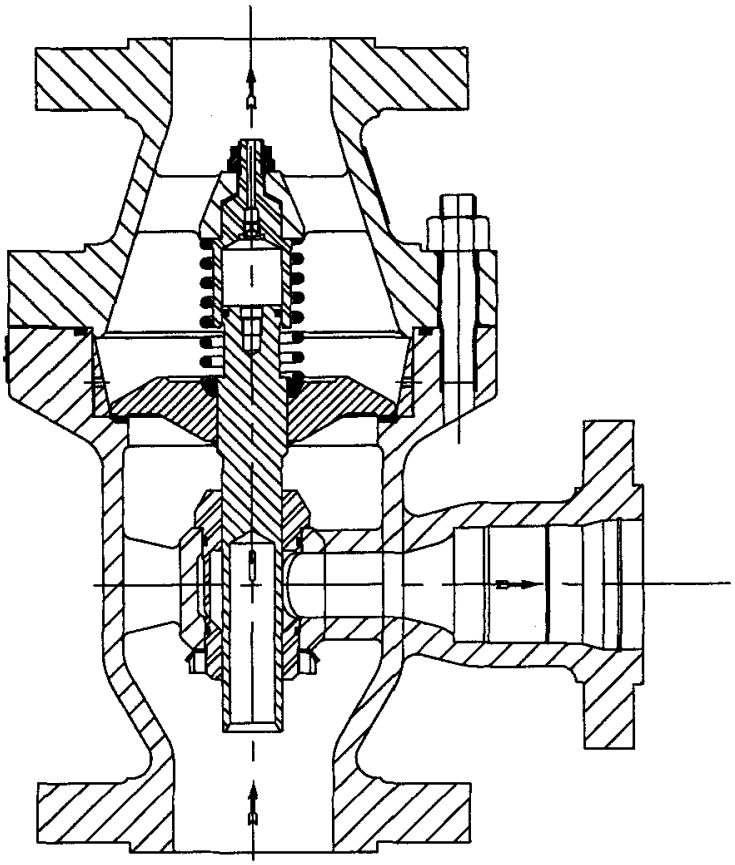


FIGURE 7 Construction of a typical ARC valve for low-pressure service

Serviceability ARC valves can be designed for in-line serviceability. Access is provided to the valves internal components for inspection or repair without disturbing the process or bypass piping. Figure 11 illustrates a valve designed to be serviced in-line.

SAFETY SHUTDOWN SYSTEMS

These controls are normally provided as backup pump protection for the continuous bypass and automatic bypass control systems. They should only operate if the normal system fails due to mechanical or operator causes. In each case, the protection system trips the pump off.

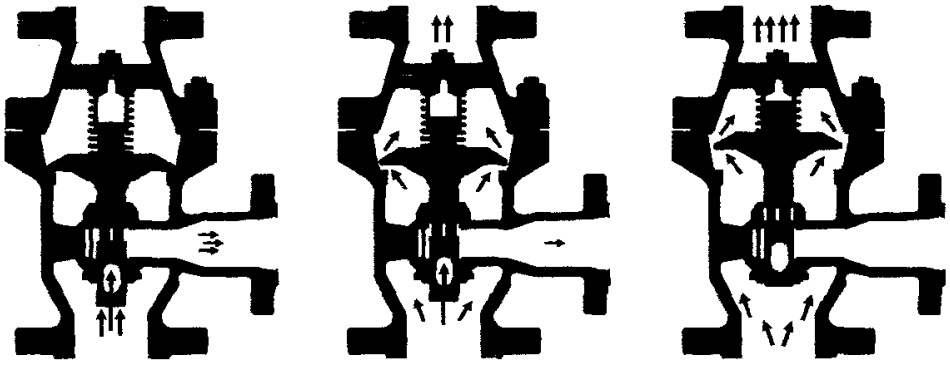


FIGURE 8A through C Basic ARC valve operation at three process flow conditions

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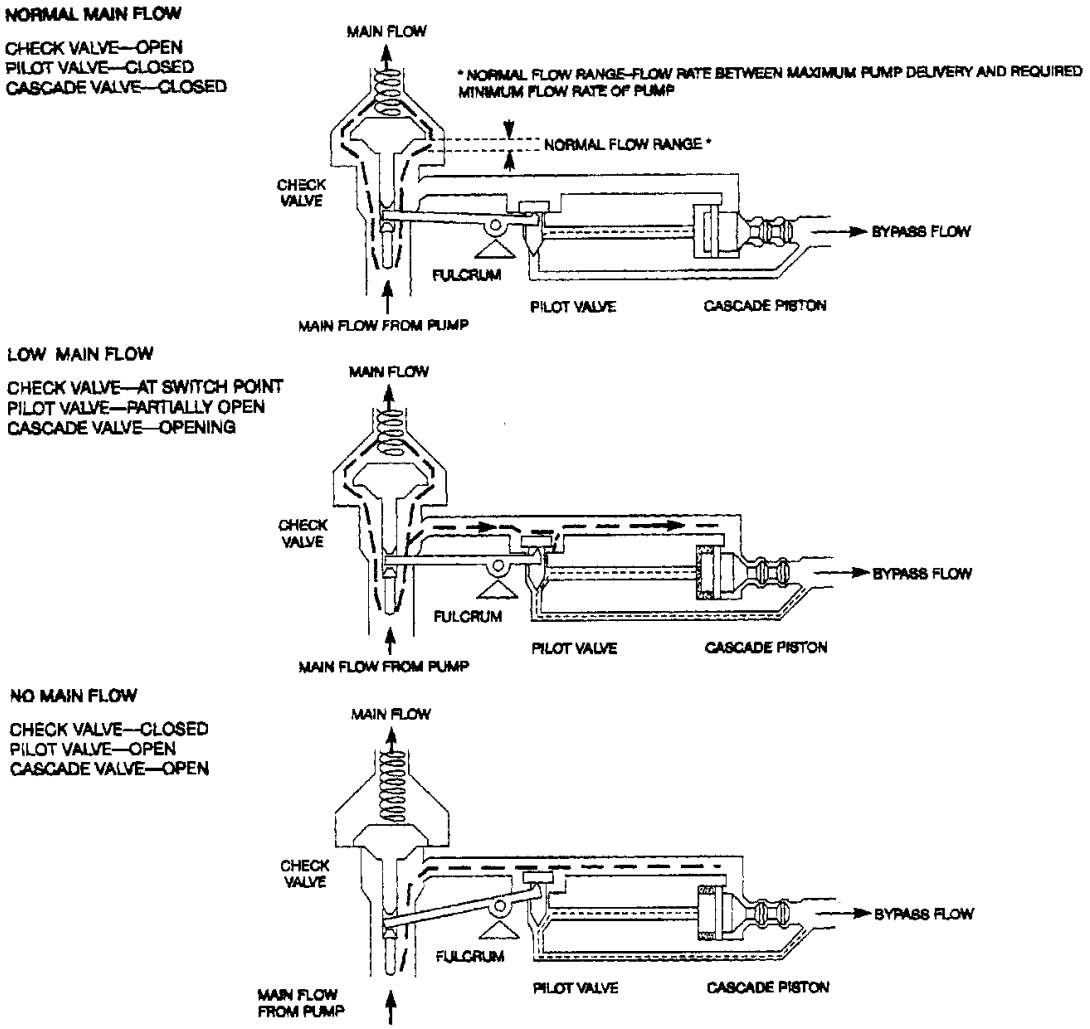


FIGURE 9 Operation of a typical pilot-operated ARC bypass valve actuated through a lever mechanism

Head This requires only an accurate pressure switch, set for a discharge pressure corresponding to the total pump head when the pump is close to the shutoff capacity. For conservative results, calculate based on minimum suction pressure. To use this method, the pump characteristic must rise continuously to shutoff. Control circuitry may require delays to reject spurious pressure surges.

Temperature A temperature sensor mounted on the pump casing or in the discharge and close to the pump can signal for a shutdown of all or some of the operating pumps. The trip can be on either a high-temperature sensing or a high differential temperature sensing between inlet and discharge. This method can protect against large temperature rises (occurring at perhaps 5% or less of capacity) but is less sensitive to smaller rises at higher percentages of capacity where adverse hydraulic effects begin for large and high-power pumps.

Low-Flow Systems These protection systems can be configured in various ways. A low-flow switch can trip the pump off at a predetermined setpoint. A flow meter can signal a controller to shut down the pump. If this protection circuit is installed, the pump start circuit must temporarily bypass this trip in order to start the pump. This is normally accomplished by a spring-loaded switch that bypasses the low-flow trip when held in the "Start" position, but activates the trip protection in the "Run" position.

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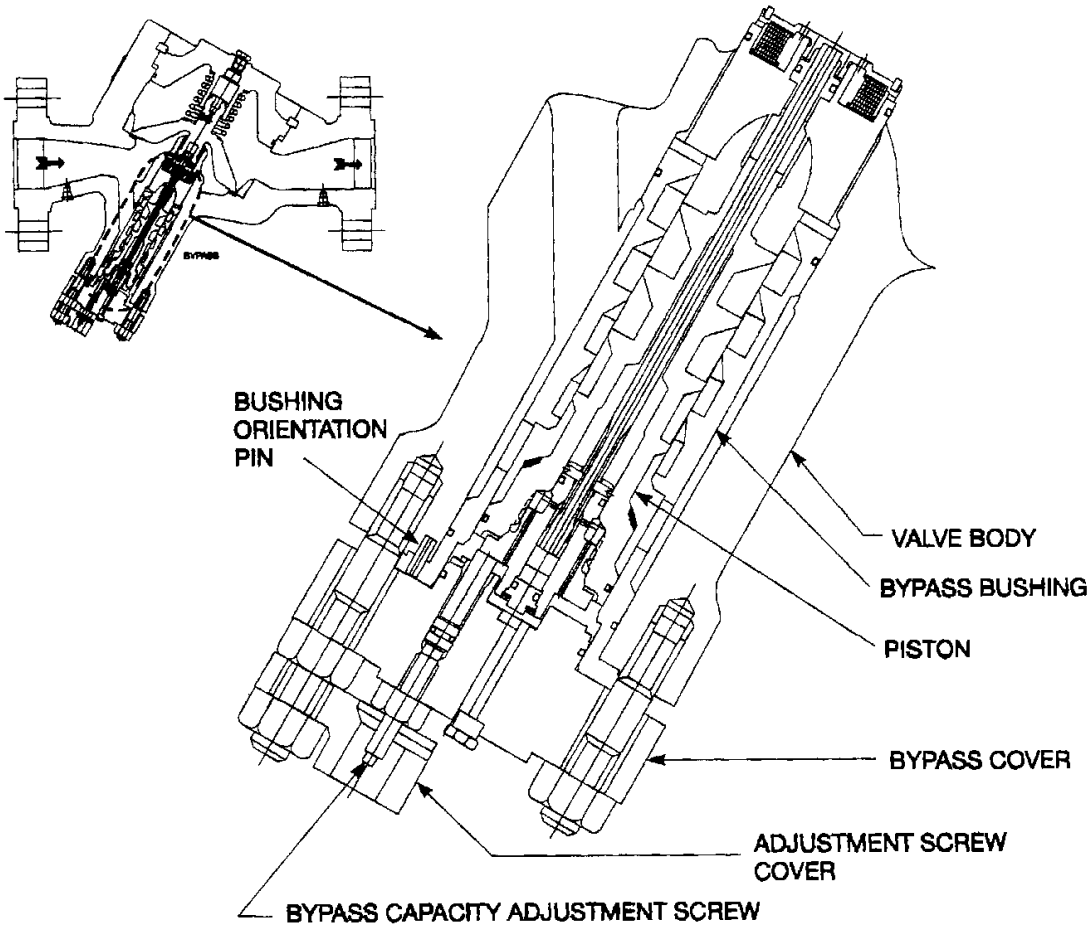


FIGURE 10 Pilot operated multistage bypass trim actuated by an in-line pilot valve

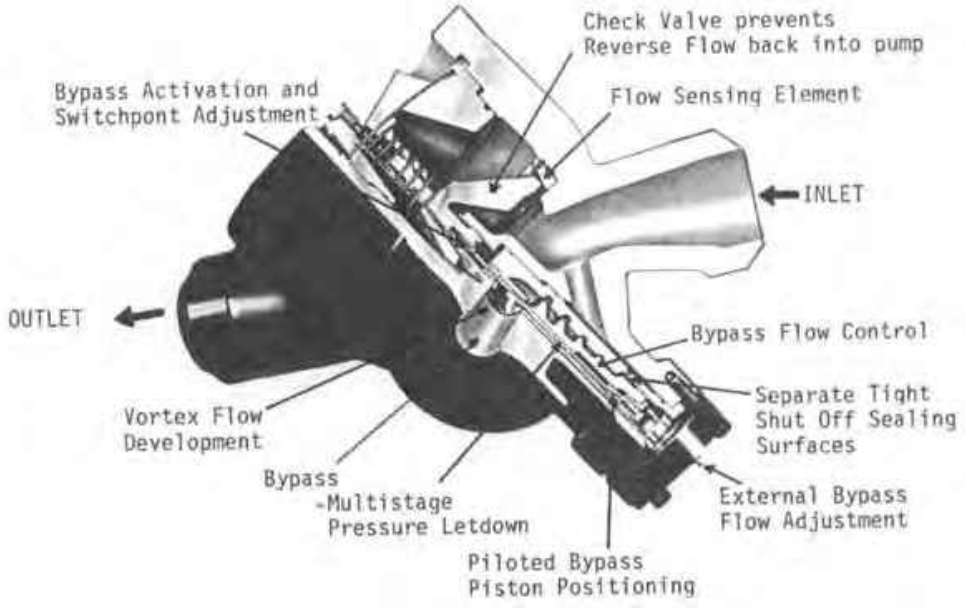


FIGURE 11 Typical in-line serviceable ARC valve

OTHER BYPASS SYSTEM HARDWARE

Orifices In high-pressure applications, the system often does not provide adequate pressure in the bypass line to prevent cavitation or flashing. Either of these conditions is undesirable because it can cause damage to both valves and the pipe system or cause a reduction in flow below the minimum desired, jeopardizing the pump protection system. All pressure-reducing valves will experience a velocity-induced recovery effect that will limit the amount of pressure drop a valve can take and cause a reduction in flow capacity.

The requirement of backpressure is generic to all pressure-reducing applications. Pressure reduction even by multiple stage cascading can minimize the requirement; however, no valve design will redefine a fluid's physical properties. This becomes especially important in modulating systems. A fixed orifice will not provide the proper backpressure at all flow levels.

As the flow in the bypass line is reduced, the orifice becomes less effective. Proper system design should be used to optimize valve pressure reduction and consider all fluid dynamic effects downstream of any pressure-reducing device. When adequate backpressure is not available downstream of a pressure reducing valve, vapor bubbles will form in the zone just downstream of the valve last stage control surface. This zone is defined as the "Vena Contracta" and represents the point of highest fluid velocity and lowest pressure.

The potential for 1) damage to downstream piping components, and 2) flow reduction exists from this point. When line pressure remains below the fluid vapor pressure, any existing bubbles will remain and expand as piping friction further reduces line pressure. This can be defined as *"flashing condition"* and is characterized by a polished appearance on affected surfaces. When the line pressure drops below the fluid vapor pressure and then recovers, any entrapped vapor bubbles will collapse (implode). This is defined as a *"cavitating condition"* and is characterized by a cinder-like appearance on affected surfaces. The resolution of either condition is best addressed by eliminating vapor formation. This can be assured by the provision of adequate back pressure through the use of a fixed or variable orifice.

Fixed Orifice Simple, easily replaced orifices that reduce the pressure are an effective way to reduce bypass head and provide adequate backpressure in bypass systems. Several stages may be necessary, however, to break down high-pressure drops without flashing. For calculations of flow through standard-shaped orifices, see Sections 11.1 and 11.2.

Coefficients of discharge for oddly shaped multistage orifices are difficult to calculate. However, manufacturers of these specialties can furnish curves of delivery as a function of pressure.

Variable Orifice In modulating systems, a fixed orifice will not provide the proper backpressure over a wide flow range. A backpressure regulator (BPR) has a variable orifice with a spring-loaded plunger that is designed to open at a specified differential pressure. If flow and differential pressure increase, it opens further to maintain the differential pressure and backpressure constant. Figure 12 illustrates a typical BPR construction. Figure 13 shows a standard BPR installation. The BPR is normally located as close to the receiver vessel as possible so that the correct backpressure is maintained in the entire bypass line.

Valves, Piping, and Fittings For cold water at low pressure, a simple power-actuated, globe-type bypass valve is often adequate. In modulating bypass systems, the bypass valve must resist throttling damage, particularly if the water is hot. Staging the pressure drop in the valve is the most common way to reduce or eliminate flashing and cavitation damage to the valve trim or body. Figure 14 illustrates a typical multistage pressure reducing valve (PRV).

Pressure is reduced in stages to ensure that the pressure never decreases below the fluid vapor pressure. This prevents cavitation and the resultant valve damage and noise. Figure 15 illustrates typical calculations for reducing pressure in sequential stages. Refer to Chap. 7 for detailed information regarding valve sizing and selection.

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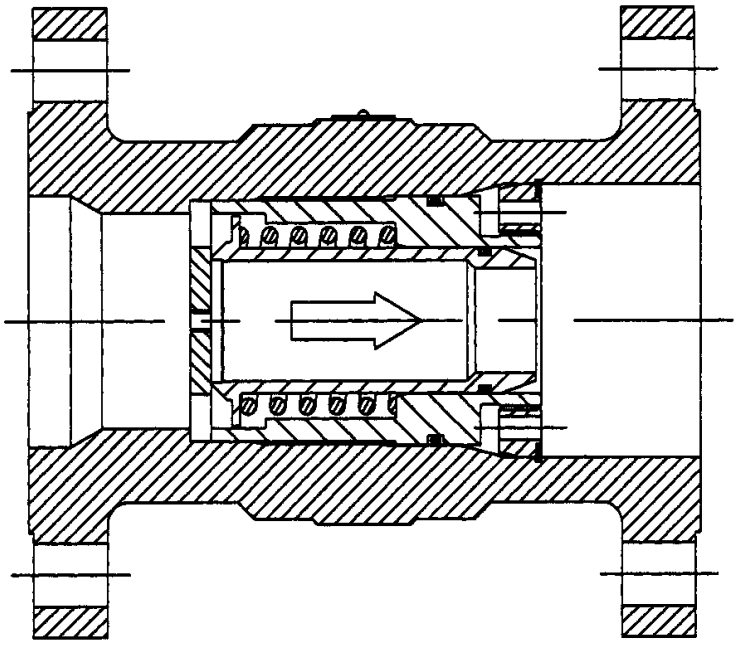


FIGURE 12 Construction of a typical backpressure regulator

Pipe material like that used for the main discharge is adequate for nearly all of the bypass line. Near the orifice and control valve outlets, however, heavier wall or higher chrome content will lengthen life. Welded piping is common for high pressures. To prevent erosion, pipe fittings (especially elbows) should not immediately follow an orifice.

Flow Meters Flow rate is the variable that must be measured for most automatic bypass multicomponent control systems. The meter may have any type of primary element that will produce an accurate signal at the process flow for which the bypass must be controlled.

A simple orifice meter or venturi tube is commonly used. The user must have the required straight upstream and downstream pipe lengths or use flow straighteners to obtain an accurate reading. The device must be properly sized to provide both accurate indication at relatively low process flows and satisfactory pressure drop at maximum process flow conditions.

Flow meters can be located either upstream or downstream of the pump. Meters located upstream are at lower internal pressure but are larger in diameter with larger flanges. In addition, pressure drop at rated flow is important to avoid low pump NPSH. Meters located downstream are normally smaller but must be rated for the maximum pump discharge pressure.

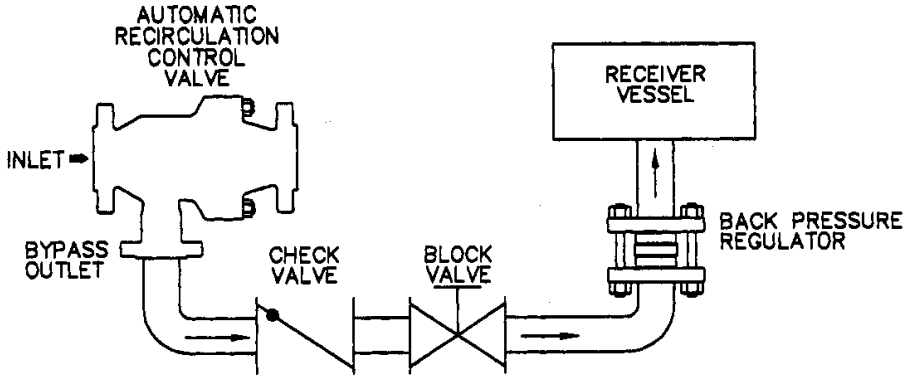


FIGURE 13 Backpressure regulator installation

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FIGURE 14 Construction of a typical multistage pressure reducing valve (Flowserve Corporation)

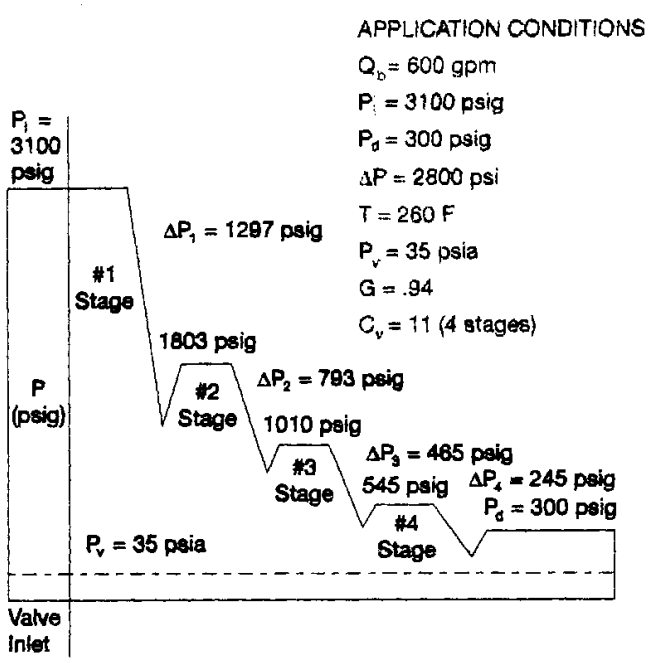


FIGURE 15 Calculations for pressure reduction through a multistage valve

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Controls Controls range in complexity from a simple pressure switch that stops a pump drive motor to modulating systems that meter flow and maintain a selected minimum flow through the pump. The control system may have to provide a deadband near the opening point and be sufficiently stable to hold bypass flow nearly constant in spite of erratic flow during upsets and startup.

For example, when the primary flow-measuring element is upstream of the bypass branch off, sufficient decease in flow will cause the bypass control to open in an "On-Off" bypass system. The main control valve, farther downstream, will then also open to maintain flow to the boiler or process. The primary element will sense the added flow and, in a simple bypass control system, close the bypass valve, initiating hunting in flow and valve action. To prevent this, the setpoint for bypass valve closing must be greater than twice the minimum bypass flow. If minimum flow is in the 30–50% range, this wastes energy during bypass. This control problem is avoided if a modulating system is installed.

SPECIAL BYPASS SYSTEMS

Pumps in Parallel It is possible to provide a single bypass valve for two or more pumps, as shown in Figure 16. The bypass line and valve must be able to handle simultaneously the total bypass flow of all pumps. The pumps must operate together and they must have identical head-capacity curves. If they do not, the bypass flow will differ between pumps. The pump with the lower head may have insufficient bypass flow or could be completely shut-off.

In the system of Figure 16 (left), the main line check valve prevents backflow through an idle pump. If the branch off were installed as shown in Figure 16 (right), an additional check valve in each bypass connection would be necessary to stop the backflow. Therefore, this latter configuration is not recommended.

An alternative arrangement that provides better operational flexibility and pump protection is illustrated in Figure 17. Each pump is individually protected by its own automatic recirculation control (ARC) valve. Pumps can be operated individually or in any combination and the pump head characteristic curves do not have to be identical.

Pumps In Series A single bypass line and valve can protect two pumps in series (see Figure 18). The designer must provide for the larger of the two separately evaluated bypass flows and must take into account the heating effect of the upstream pump. The pumps cannot operate singly unless additional piping and controls are installed.

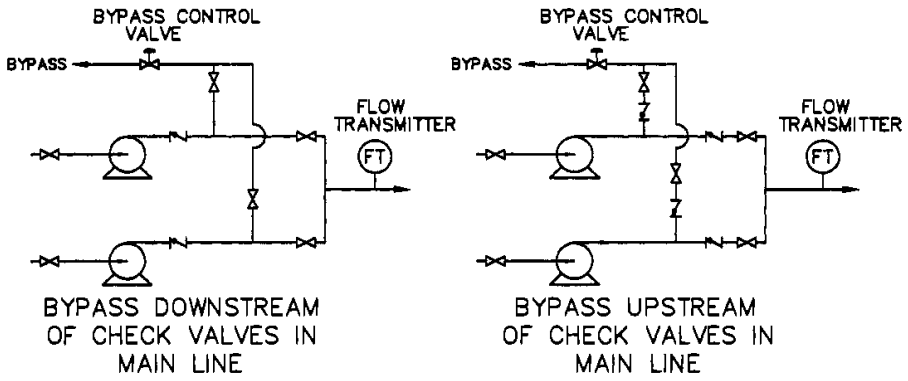


FIGURE 16 Examples of a single bypass valve that provides minimum flow control for two parallel pumps

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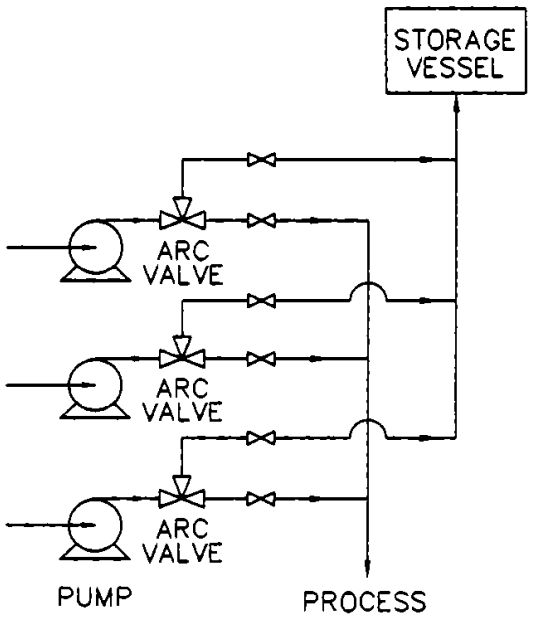


FIGURE 17 Multiple parallel pumps individually protected by ARC valves

VARIABLE SPEED PUMP BYPASS CONSIDERATIONS

Large horsepower pumps are often powered by steam turbines or with motors connected through a variable speed coupling. The effect of decreased speed and discharge head is to reduce the minimum flow requirement for the pump. Therefore, if the minimum required bypass flow is calculated and sized with the pump at normal operating speed and pressure, the bypass flow will be adequate when the pump is operated at lower speeds.

Warning If a constant backpressure device such as a backpressure regulator is used to control bypass pressure, precautions must be taken to ensure that the pump is never operated at pressures below the BPR setting. If the pump is operated below the BPR setting, the pump will have no bypass flow and may be damaged in a short time.

In a typical boiler system, a constant speed booster pump, requiring less minimum flow than the boiler-feed pump, could be protected by the bypass flow for the variable speed boiler feed pump. The designer must determine if the booster pump bypass flow is sufficient when the high-pressure feed pump is running at minimum discharge pressure.

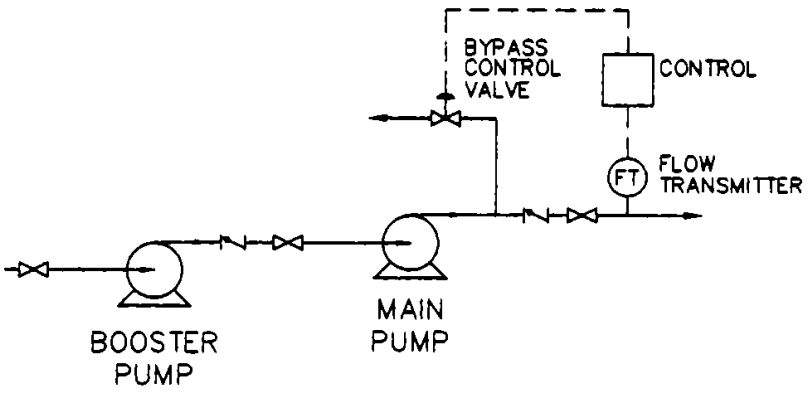


FIGURE 18 Example of a single bypass line that protects two pumps in series

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C • H • A • P • T • E • R • 12

PUMP SERVICES

SECTION 12.1
WATER SUPPLY

JIM BIRD

"Water is the driving force of all nature." (Leonardo da Vinci)

INTRODUCTION

The supply of water to households and industry in sufficient quantities and of a high enough quality to make it something that is taken for granted is one of the hallmarks of the United States' infrastructure. In the years just following the turn of the twenty-first century many retrospective surveys were conducted and one prominent engineering organization determined that the development of reliable and safe water supply and distribution systems ranked fourth on a list of the greatest engineering achievements of the twentieth century.[1] Arguably the key element in all these systems is the ability to draw water from the source, transport it to treatment facilities, process it through the treatment and finally send it out to consumers where it is used with little consideration of the effort required to supply such a basic need. Standing as the most important piece of machinery in each of these steps is the pump, in a wide variety of the classifications described in Chap. 1 of this book.

WATER RESOURCES

"In one drop of water are found all the secrets of all the oceans." (Kahlil Gibran)

The Water Cycle The first step in the full utilization of any resource is an understanding of its nature. When the earth and its atmosphere are viewed as a whole it defines a closed system not gaining or losing matter, including water. What this means is that the water present at the beginning is still the water that is here today. When taking a closer look into the earth's system it is evident that water is always in the process of change. Readily recognized are the many forms of water that exist at any one time and how freely water moves between them. This movement from phase to phase is a combination of the filtration, distillation, and purification processes that are combined into the steps of the *water cycle* (see Figure 1).

The water cycle has no starting point. But, one can begin in the oceans, since that is where most of earth's water exists. The sun, which drives the water cycle, heats water in the oceans. Some of it evaporates as vapor into the air. Ice and snow can sublimate directly

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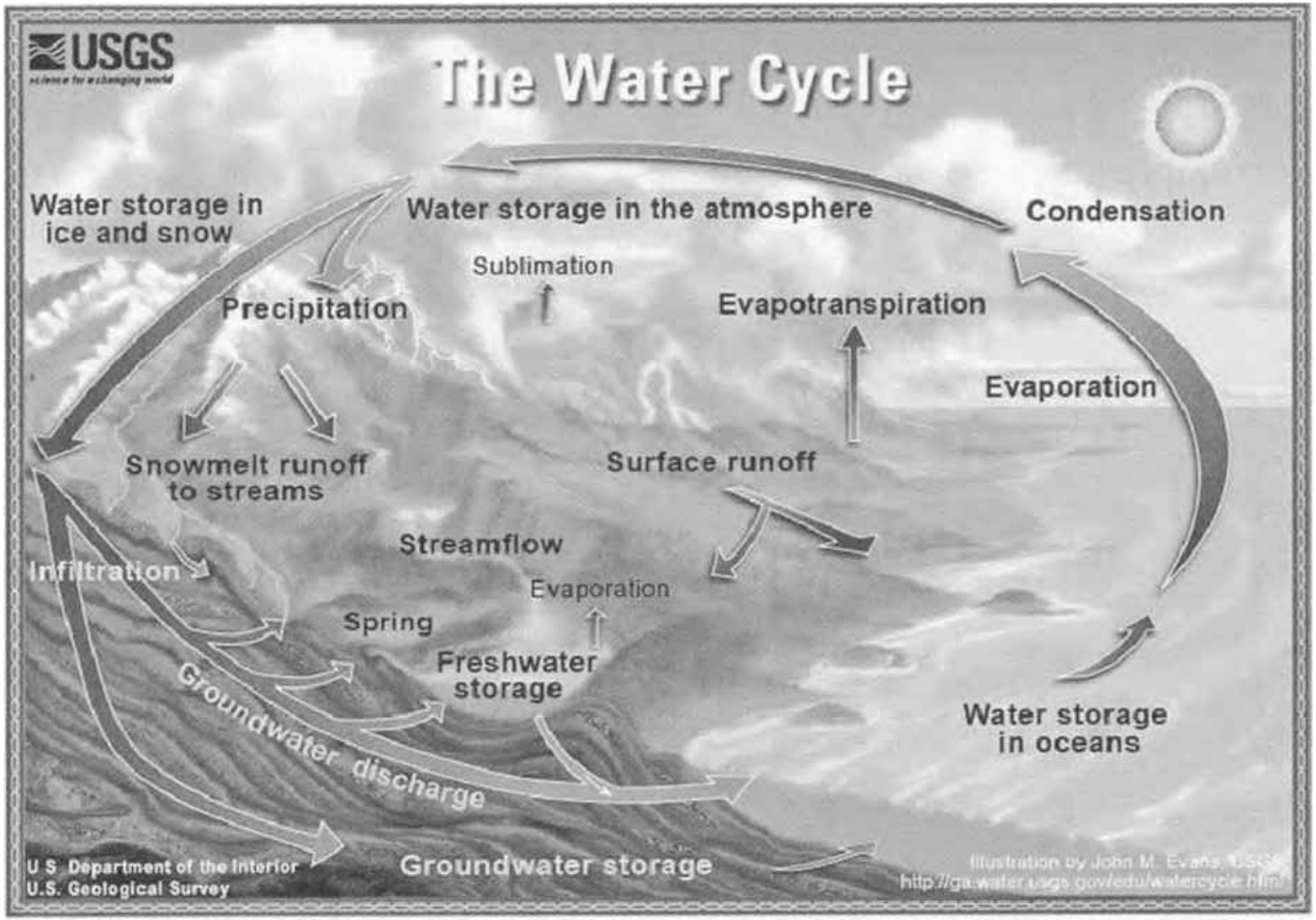


FIGURE 1 The hydrologic cycle (U.S. Geological Survey, U.S. Department of the Interior)

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into water vapor. Rising air currents take the vapor up into the atmosphere, along with water from transpiration, which is water transpired from plants and evaporated from the soil. The vapor rises into the air where cooler temperatures cause it to condense into clouds. Air currents move clouds around the globe, cloud particles collide, grow, and fall out of the sky as precipitation. Some precipitation falls as snow and can accumulate as ice caps and glaciers, which can store frozen water for thousands of years. Snowpacks in warmer climates often thaw and melt when spring arrives, and the melted water flows overland as snowmelt. Most precipitation falls back into the oceans or onto land, where, due to gravity, the precipitation flows over the ground as surface runoff. A portion of runoff enters rivers in valleys in the landscape, with streamflow moving water toward the oceans. Runoff and groundwater seepage accumulate and are stored as freshwater in lakes. Not all runoff flows into rivers, though. Much of it soaks into the ground as infiltration. Some water infiltrates deep into the ground and replenishes aquifers (saturated subsurface rock), which store huge amounts of freshwater for long periods of time. Some infiltration stays close to the land surface and can seep back into surface-water bodies (and the ocean) as groundwater discharge, and some groundwater finds openings in the land surface and emerges as freshwater springs. Over time, though, all of this water keeps moving, some to reenter the ocean, where the water cycle is continued.[2]

Water Supply Although the earth's water, in various forms, is constantly progressing through the hydrologic cycle the quantity present in each step along the way remains relatively constant. Table 1 lists the storage locations and the amount that they hold.

The majority of these locations present a usable source of water that can be retrieved and used to meet the daily demands of societies. The sources exhibit great variations in the form, quality, and accessibility of the water. A cursory analysis of the percentages in the table produces the following observations:

- Freshwater resources comprise less than 3% of the total available water supply.
- Water in its solid form represents about two-thirds of the available freshwater.
- Groundwater storage contains the vast majority of the liquid freshwater.
- Lakes store only 1% of the water.
- Streams and rivers hold only 0.013% of the available freshwater.

In general, the analysis supports one general conclusion that the ease of access to available water supplies is inversely proportional to the amount of water contained in the specific resource. The challenge to water-supply professionals is how to manage the resources available in their location, supply a reliable product to their consumers and make certain that there remains a supply for those downstream or next door.

Water Usage Total withdrawals of available water in the United States for the year 2000 were estimated to be 408,000 million gallons per day (MGD) (= 17,875,500 L/s). This usage was apportioned as shown in Figure 2. Withdrawals of the available water supply for

TABLE 1 Earth's water inventory

Location	Volume (million cubic km)	Percent of Total
Oceans	1370	97.25
Ice caps and glaciers	29	2.05
Groundwater	9.5	0.68
Lakes	0.125	0.01
Soil moisture	0.065	0.005
Atmosphere	0.013	0.001
Streams and rivers	0.0017	0.0001
Biosphere	0.0006	0.00004

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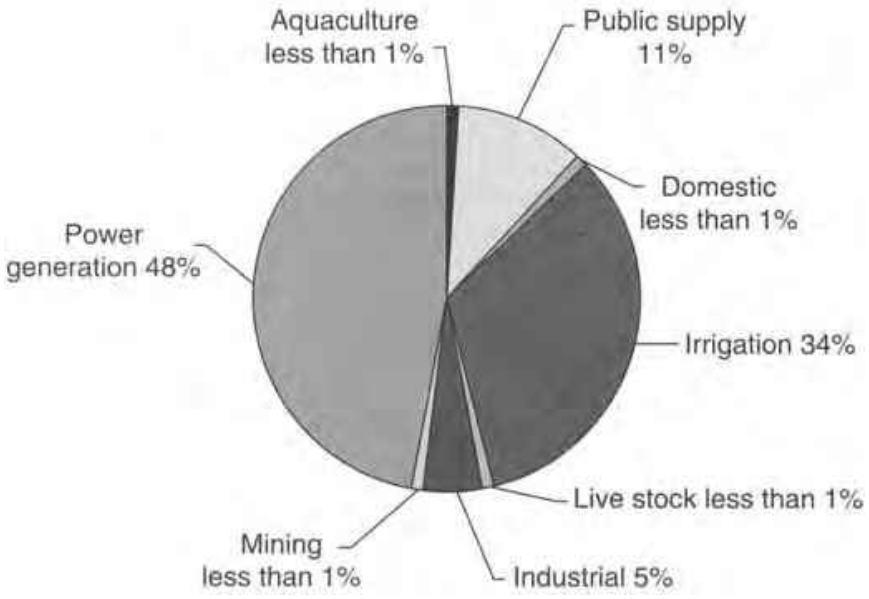


FIGURE 2 Water usage for the United States—2000 (U.S. Geological Survey, U.S. Department of the Interior)

public supply accounted for about 11% of this total or about 44,000 MGD (1,927,800 L/s). The public-supply category includes water withdrawn by public and private water suppliers that furnish water to at least 25 people or have a minimum of 15 connections. Public-supply water may be delivered to users for domestic, commercial, industrial, or thermoelectric-power purposes. Some public-supply water may be delivered to other public suppliers or used in the processes of water and wastewater treatment. Public-supply water is used for such public services (public uses) as pools, parks, and public buildings; or be unaccounted for (losses) because of system leaks or such nonmetered services as firefighting or the flushing of water lines. Some public suppliers treat saline water before it is distributed. However, all public-supply withdrawals in this report are considered freshwater.[3]

Figure 3 shows the steady increase of withdrawals with the growth of the population. Growth in demand without a growth in supply challenges the water-supply professionals

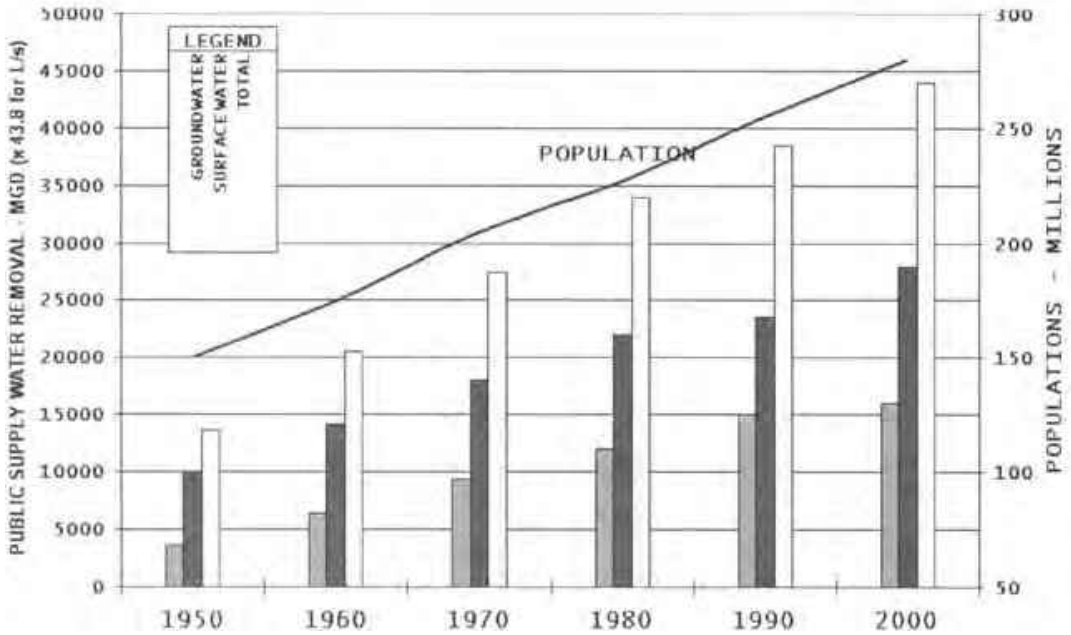


FIGURE 3 Growth of public water usage for the United States—1950 to 2000 (U.S. Geological Survey, U.S. Department of the Interior)

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to both better manage traditional sources and look to nontraditional sources to satisfy users.

The withdrawal of water for public supply still relies heavily on the most convenient source, surface water as shown in Figure 3. Since 1950 the percentage of the total usage for public supply being taken from surface water sources has dropped from 75 to 64%, showing an increased reliance on groundwater. However, the actual withdrawal amount from surface water still has grown from 10,000 to 28,000 MGD (438,100 to 1,226,800 L/s).

Traditional Water Sources The primary source of freshwater for public water supply is the watershed. This is all the water in the hydrologic cycle from the precipitation deposited on the ground until the water flows out into the sea. A watershed is defined as an area of land that drains all the streams and rainfall to a common outlet such as the outflow of a reservoir, mouth of a bay, or any point along a stream channel. The watershed consists of surface water—lakes, streams, reservoirs, and wetlands—and all the underlying groundwater.[3] The Chesapeake Bay Watershed, shown in Figure 4, drains a large portion of the northeast United States to the Atlantic Ocean. But this watershed is made up of many smaller ones each feeding the streams and rivers that eventually feed the bay. Within the watershed the precipitation is carried along in various ways made up of two principle divisions, surface water and groundwater.

SURFACE WATER The most available water in the watershed is that on the surface. Surface water is made up of all the streams, rivers, ponds, and lakes within the watershed.



FIGURE 4 Chesapeake Bay Watershed (Chesapeake Bay Program)

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Reservoirs may be built along the flow path to retain supplies close to points of use. Due to the relative ease of removal surface water is the most relied upon source for freshwater.

GROUNDWATER A significant amount of the precipitation falling into a watershed will infiltrate the ground, where it will enter the local aquifer, a geological formation or structure that stores and/or transmits water. Some water that infiltrates will remain in the shallow soil layer forming an unconfined aquifer, where it will gradually move downhill, through the soil, and eventually enters the stream by seepage into the stream bank. Some of the water may infiltrate much deeper, recharging confined groundwater aquifers. Water may travel long distances or remain in storage for long periods before returning to the surface (see Figure 5).

At any point in the watershed there can be interchange between the surface and groundwater. Surface water is absorbed into the ground as it flows in streams or sits in water in an unconfined aquifer can come to the surface where the land drops below the local water table. Water in a confined aquifer can come to the surface through springs or artesian wells.

EFFECTS OF SOURCE ON WATER QUALITY Although the quality of water obtained from both groundwater and surface supplies varies greatly according to local climatological, hydrological, and geological conditions, some general comparisons can be made between groundwater and surface supplies. Surface supplies generally contain more bacteria, algae, and suspended solids than do groundwater sources and thus require specific treatment to eliminate the resulting turbidity, colors, and tastes. Groundwater supplies generally contain few bacteria and in some instances are pure enough for domestic use without chemical treatment. Frequently, however, groundwater supplies contain significant amounts of dissolved minerals. Depending upon the mineral type, the resulting supply may exhibit such characteristics as exterme hardness, toxicity or undesirable color, odor or taste. Special treatment such as aeration, lime softening, and disinfection may be required to remove such objectionable characteristics.

Developing Sources of Water A popular pump journal placed on its cover the question, "Water: The Next Oil?" The lead article began with the statement, "Take this to the bank: water will eventually replace oil as our planet's core commodity."[4] Also, Ismail

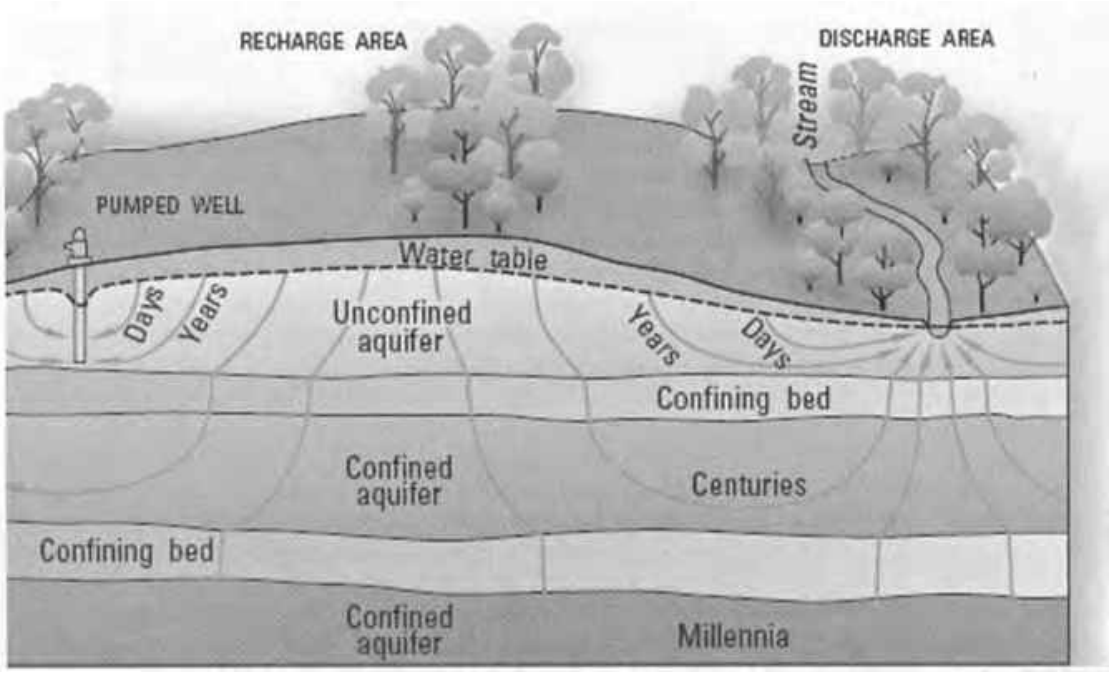


FIGURE 5 Groundwater storage and flow (U.S. Geological Survey, U.S. Department of the Interior)

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Serageldin, World Bank Vice President for Environmental Affairs in 2000, was quoted as saying, "The wars of the twenty-first century will be fought over water." With this outlook on the growth of demand for water and the relatively constant supply in each of the steps of earth's water inventory, the need to develop resources other than the watershed has caused a look to the larger but lesser used sources.

OCEANS At over 97% of the available water, the oceans represent the promise of an inexhaustible water supply. The problem is to make the water potable at a cost and quality acceptable to the market. Currently, two methods exist for the separation of the dissolved salts from the feedwater, thermal, and membrane technology.

In the thermal process, essentially distillation, the feedwater is evaporated and then condensed. The energy cost of distillation is one of its major drawbacks. The most common methods of distillation include vapor compression (VC), multistage flash (MSF), and multieffect distillation (MED). The vapor compression process involves evaporating the input water and then compressing the vapor. The vapor is then condensed, and the heat released is used to evaporate the input water. In the multistage flash process, saline feedwater is heated and the pressure is lowered so that the water "flashes" into steam. This process constitutes one stage; then comes a number of stages in series, each of which is at a lower pressure. In multieffect distillation, there is a number of evaporators in series, and vapor from one series is used to evaporate water in the next effect.

In membrane technologies pressure is applied to force a solution through a membrane. The membrane is semipermeable, meaning it is designed to allow the passage of water while trapping the impurities. The most common membrane technology is reverse osmosis. The name comes from the fact that it reverses the natural process of osmosis where the solution passes through the membrane from low to high concentration. The reversal of the natural process is accomplished by applying pressure to the high concentrate side of the membrane above the natural osmotic pressure. Pretreatment of the feed solution is necessary to eliminate solids and microorganisms that would otherwise impede the process.

SOLID WATER Where accessible, ice in glaciers and ice packs, representing two-thirds of the earth's supply is an attractive source of freshwater. In the United States, Alaska represents the most promising resource. There are an estimated 100,000 glaciers in Alaska covering approximately 28,800 mi^2, or 3% of Alaska. This is 128 times the area covered by glaciers in the rest of the United States, storing three-fourths of all freshwater in Alaska. Many of the state's 100,000 glaciers are inaccessible, practically speaking, but where they can be accessed the State of Alaska regulates the removal of ice. People or companies wanting to harvest Alaska's glaciers must apply for a permit if they plan to take away more than 20 tons a year. Glacier ice is now harvested from glaciers that calve into tidewater. The state permit grants a right to the water that makes up the ice. Not every glacier is fair game, and a permit isn't to be granted if the harvest would interfere with the rights of other users, such as wildlife, fish, and humans.[5] Glacier ice harvesting is a successful business in Alaska, a 1-ton block of clean glacial ice sells for $500 wholesale.[6]

WATER DEMAND

"We never know the worth of water 'til the well is dry." (Thomas Fuller)

Water Consumption The primary design parameter in the development of a public water supply system is the capacity of the system needed to meet the demand of the service area. This is dependent on many variables including the population to be served, usage habits and growth estimates for the utility. The consumption rate is normally expressed as gpcd (gallons per capita per day) in U.S. customary units or lpcd (liters per capita per day) in SI units. The world water consumption per inhabitant varies greatly; in the United States the average is 132 gpcd (500 lpcd), in western Europe 40 gpcd (150 lpcd) and in Africa 13 gpcd (50 lpcd).[7] Even within a country the rates can vary greatly. Using the data from Figure 3 the per capita consumption curve shown in Figure 6 can be drawn.

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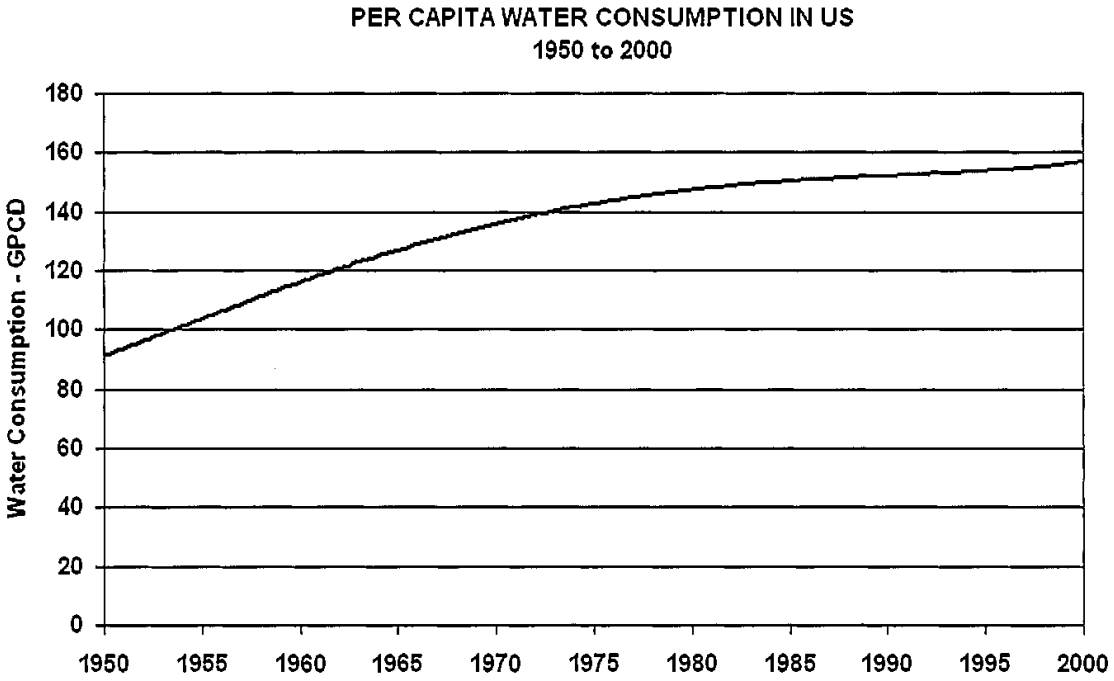


FIGURE 6 Per capita water consumption in the United States (U.S. Geological Survey, U.S. Department of the Interior) (liters = gallons × 3.78)

The consumption rate typically used for design purposes is 175 to 200 gpcd (670 to 760 lpcd). This range is supported by the chart above; however, this should be compared to the historical data for the City of New York shown in Figure 7. This demonstrates the effects of an effective water conservation education effort resulting in not only a reduction in per capita usage but in real total usage even with a population increase.

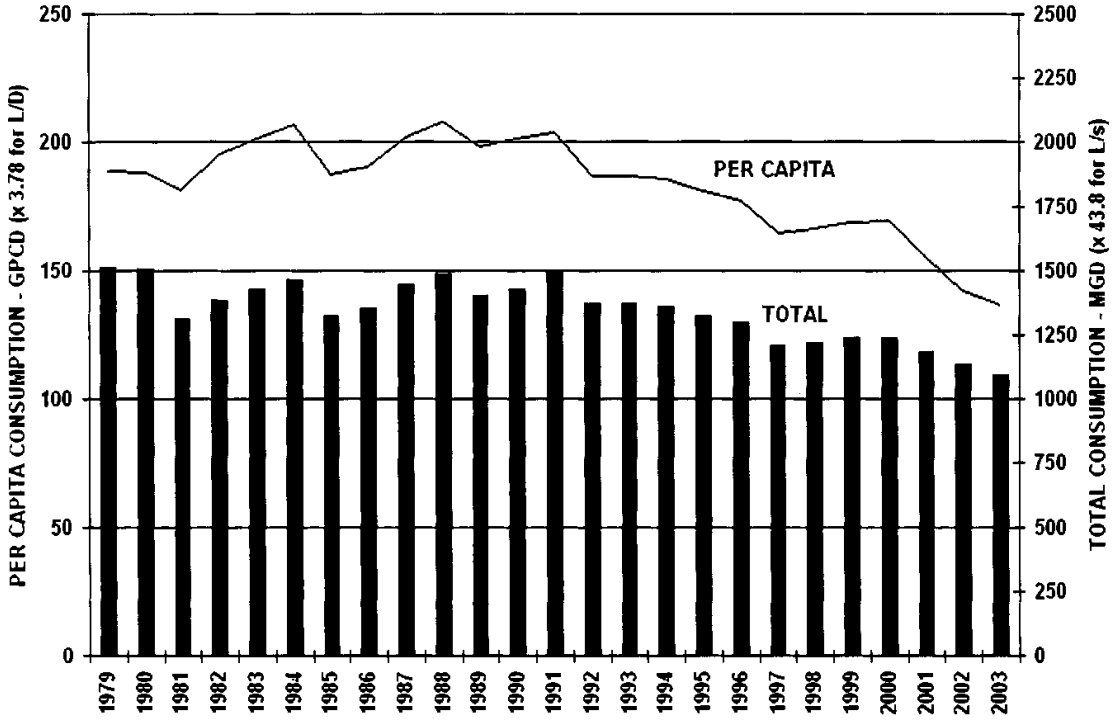


FIGURE 7 Total and per capita water consumption in NYC (The City of New York, Department of Environmental Protection)

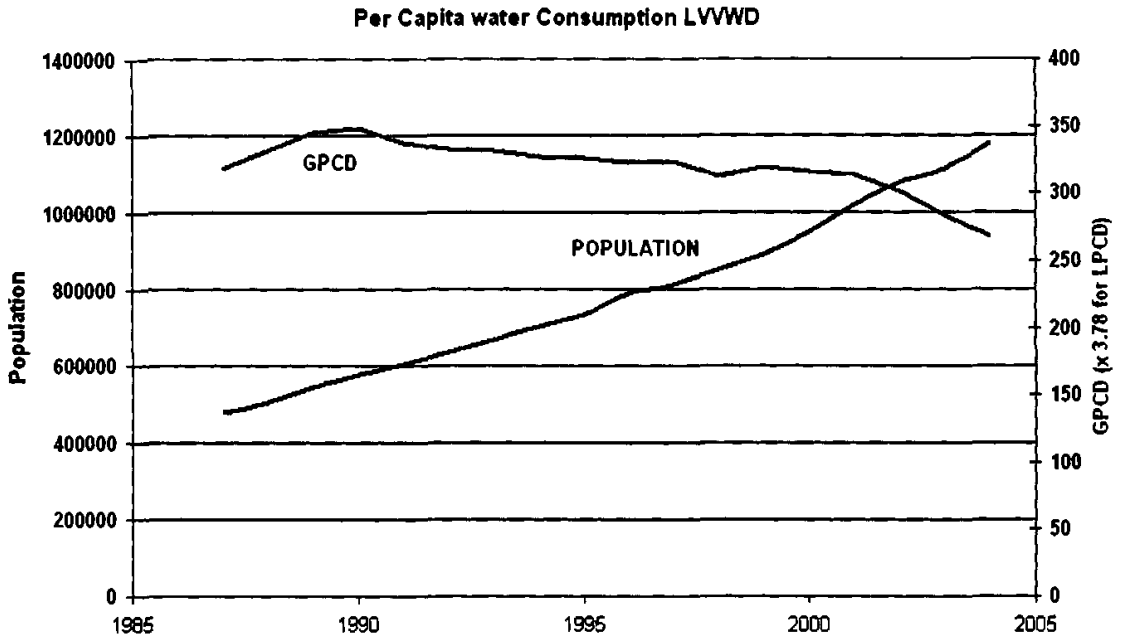


FIGURE 8 Per capita water consumption Las Vegas (Las Vegas Valley Water District)

Geographical and climatic effects also exert a strong influence in usage habits. In this regard, the data for the Las Vegas Valley Water district shown in Figure 8 should be considered. This utility serves a growing population in an arid location with higher per capita usage than New York City. This also shows the effect of conservation efforts as the total water usage has only doubled over the recorded period while the population has tripled.

Location-specific effects and usage trends like these are difficult to predict and show the extensive nature of the preliminary studies on plant and system size.

Demand and Daily Fluctuation The average consumption rate is the primary sizing factor but does not provide the full set of parameters for the system design. There are large fluctuations in the demand rate based on season, month, day of the week and hour of the day. The relationship of these parameters to the base average is critical to the ability to meet all demand possibilities. The usage and demand rates are as follows:

1. **Average daily demand** It may be expressed as gallons (liters) per capita per day or million gallons (liters) per day (the total year's usage divided by 365). The average daily demand rate is generally used as the yardstick by which all other demand rates are measured.
2. **Maximum monthly demand** This is the average daily use during the month of greatest consumption. It is determined by dividing that month's usage by the number of days in the month.
3. **Maximum daily demand** This is the total amount of water used during the day of heaviest consumption in the year. Experience shows this demand rate occurs on from three to five consecutive days during the year.
4. **Maximum hourly demand** This is the rate of use during the hour of peak demand on the day of maximum demand. This demand rate normally establishes the highest rate of design for distribution systems and peak pumpage.

Of these rates, the most difficult to determine accurately is the hourly demand because it is a nearly instantaneous snapshot of all the components of the system including pumping rates and storage levels. The development of sophisticated monitoring devices like Supervisory Control and Data Acquisition (SCADA) systems have made this a much simpler and accurate process. Figure 9 shows the hourly fluctuations for the Las Vegas Valley Water District.

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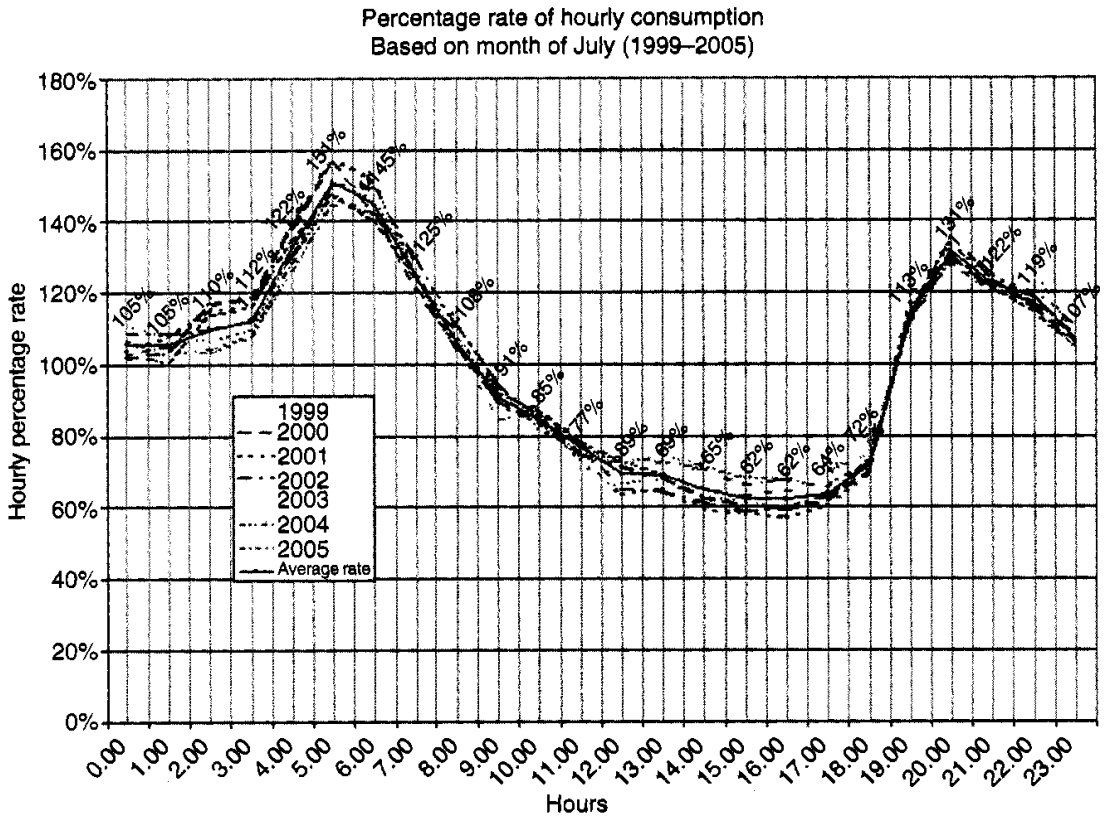


FIGURE 9 Hourly demand fluctuations (Las Vegas Valley Water District)

Demand Rates as Basis of Design The fluctuations of demand expressed as a percentage of some base value must be developed and used as the base for the system design. Performing a sort of worst-case analysis can do this. Historical data having been used to establish the four demand rates listed above, these can now be used for the process. In the example shown the base value chosen is the *average daily demand*. This is the most commonly used parameter although another may be used if shown by the data gathering activity to be more applicable to the usage pattern. The other three demand rates are then expressed as a percentage of this value. The analysis proceeds by manipulating the data as if the maximum day occurs during the maximum month and further as if the maximum hour occurs during the maximum day and month. The results of such an analysis are shown in Figure 10 based on data provided by the Las Vegas Valley Water District. The design rates suggested in the chart are averages of the results. The large variation in the hourly results may lead to a higher value than the 342% being selected by the designers.

WATER SUPPLY

"Pure water is the best of gifts that man to man can bring." (Anonymous)

Water Supply Basics Meeting the demand for water involves one or more steps in a set of processes that define the full scope of water supply. Not all systems will require every step and not all systems will utilize the same technology to achieve the end result of that step. The basic functions of a water supply system are these:

- *Screening*: Prior to entering the actual treatment process there may be a rough screening of the raw water to remove floating debris and large particles that could damage or clog equipment.

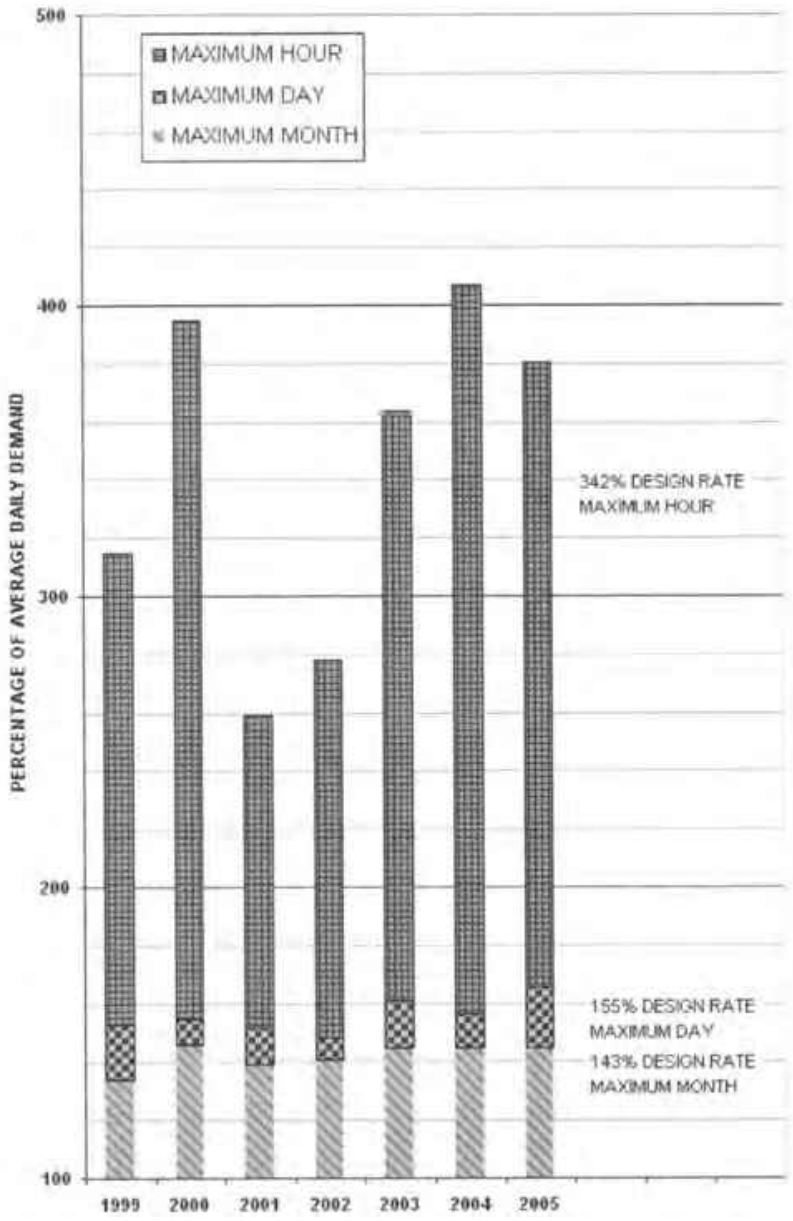


FIGURE 10 Comparison of maximum month, day, and hour to average daily demand (Las Vegas Valley Water District)

- *Acquisition*: This is the withdrawal of the feed supply, usually termed "raw water," from the source.
- *Transmission*: Depending on the location of the source it may be necessary to transport the raw water over significant distances to the plant.
- *Coagulation*: This is the collecting of minute particles of a solid dispersed throughout a liquid into a mass followed by the precipitation or separation of the solid mass from the liquid.
- *Clarification*: Also called *sedimentation*, this step requires the retention of the treatment flow in a large tank called the clarifier or sedimentation basin. This is to allow the coagulants to settle to the bottom where they can be removed as sludge. Exit from the clarifier is generally over a weir so only the water furthest from the sediment goes on into the system.
- *Filtration*: After clarification the water passes through a filter to remove unsettled sediment and any other particles. Several different filtration technologies exit including sand filters, pressure filters, and membrane "ultrafilters."

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- *Disinfection*: A final step in the purification process, disinfection is used to destroy any pathogens remaining after filtration. Forms of disinfection include chlorination, ozonation, and UV radiation.
- *Fluorination*: The addition of fluorides to the water is accomplished by the addition of dihydrogen hexafluorosilicate. This compound decomposes in water, releasing fluoride ions.
- *Storage*: Following the disinfection process the water is held in temporary storage to complete the action and then goes on to finished water storage.
- *Distribution*: The final step in the water supply system is the distribution of the water to the user.

Source of Water Quality The levels of treatment depend largely on the source of the raw water. The quality of the water can vary greatly based on the local climate, watershed and geology. Surface water problems influencing the treatment can include solids; suspended and dissolved; color, odor, microorganisms and taste. Groundwater may be pure enough for domestic use without chemical treatment; and domestic withdrawals amount to about 3000 MGD (131,000 L/s), with much being consumed as pumped. Groundwater problems could include hardness, acidity, and dissolved minerals that can cause objectionable taste and odor.

Pump Services for Water Supply The reliable supply of high quality water in sufficient quantities to support the lifestyle of the user is one of the engineering hallmarks of our society and the heart of the system, from source to destination, is the pump. At each step of the process, pumping machinery of all types transforms raw energy into the flow necessary to feed the system.

Raw Water The delivery of the primary feedstock into the treatment process begins with the extraction of the raw water from the source. The raw water system is highly dependent upon the source of the water, the style of the intake structure and the location relative to the treatment plant. The primary determinant of the raw water pump station design is source of the water, surface water from a lake, river or reservoir, or groundwater from a well. Surface water sources can have a further effect on the pump selection. Where the source of water is from a reservoir at an elevation above the pump station the normal selection would be a horizontal centrifugal pump of the general design, shown in Figure 11. A typical station layout for a pump of this type is shown in Figure 12. The basic reason for this style of pump being used is that the elevation of the source water above the pump provides the energy (net positive suction head) to fill the pump and support the energy requirements of the impeller. The station and the pump picture both show the bottom suction feature that simplifies the piping of the station.

Other surface water sources such as lakes or rivers place the water below the surrounding grade level where it would take a significant amount of costly excavation to place the pumps below the water level. For these sources pump stations are typically built along the shoreline or riverbank with an intake structure extending out into the water. If horizontal pumps were to be used then a priming system to evacuate the pump and lift the water into the impeller prior to starting would be required. To avoid the extra cost and maintenance of this system vertical pumps are used with the pump suspended down into the water.

Figure 13 shows a vertical raw water pump station where the design not only allowed for the pumps to be suspended in the suction source to provide the NPSHR for the impeller but also to accommodate a drop in the suction source water level. This installation was also designed to allow the number of stages in the liquid end pumps to be increased and the pumps lengthened if the level of the suction continued to drop due to climatic conditions and upstream usage.

Figure 14 shows a typical vertical raw water pump station. The only part of the pump in the picture, barely visible behind the piping, is the discharge head. This arrangement can save floor space in the pump station but can be more costly to maintain.

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FIGURE 11 Horizontal, axially-split, single-stage, bottom-suction sump (Flowserve Pump Division)

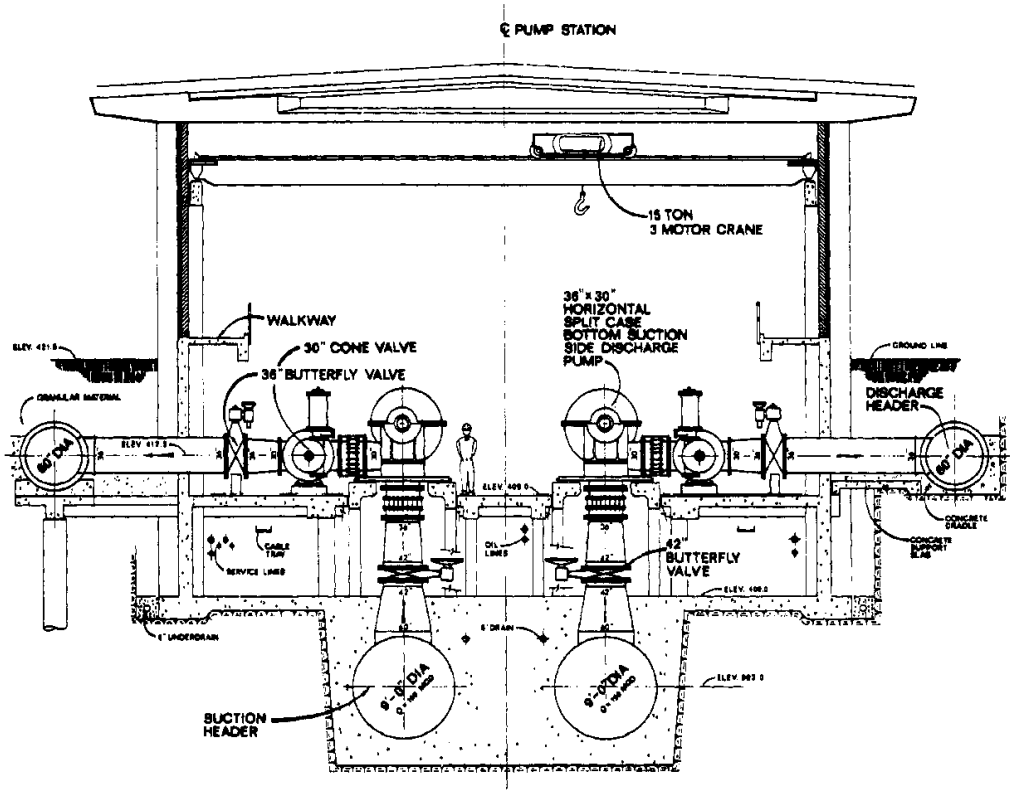


FIGURE 12 Typical section of the Forney raw water pumping station in Dallas, showing pump suction and discharge piping (1 in = 2.54 cm; 1 ft = 0.3048 m) (URS/Forrest and Cotton)

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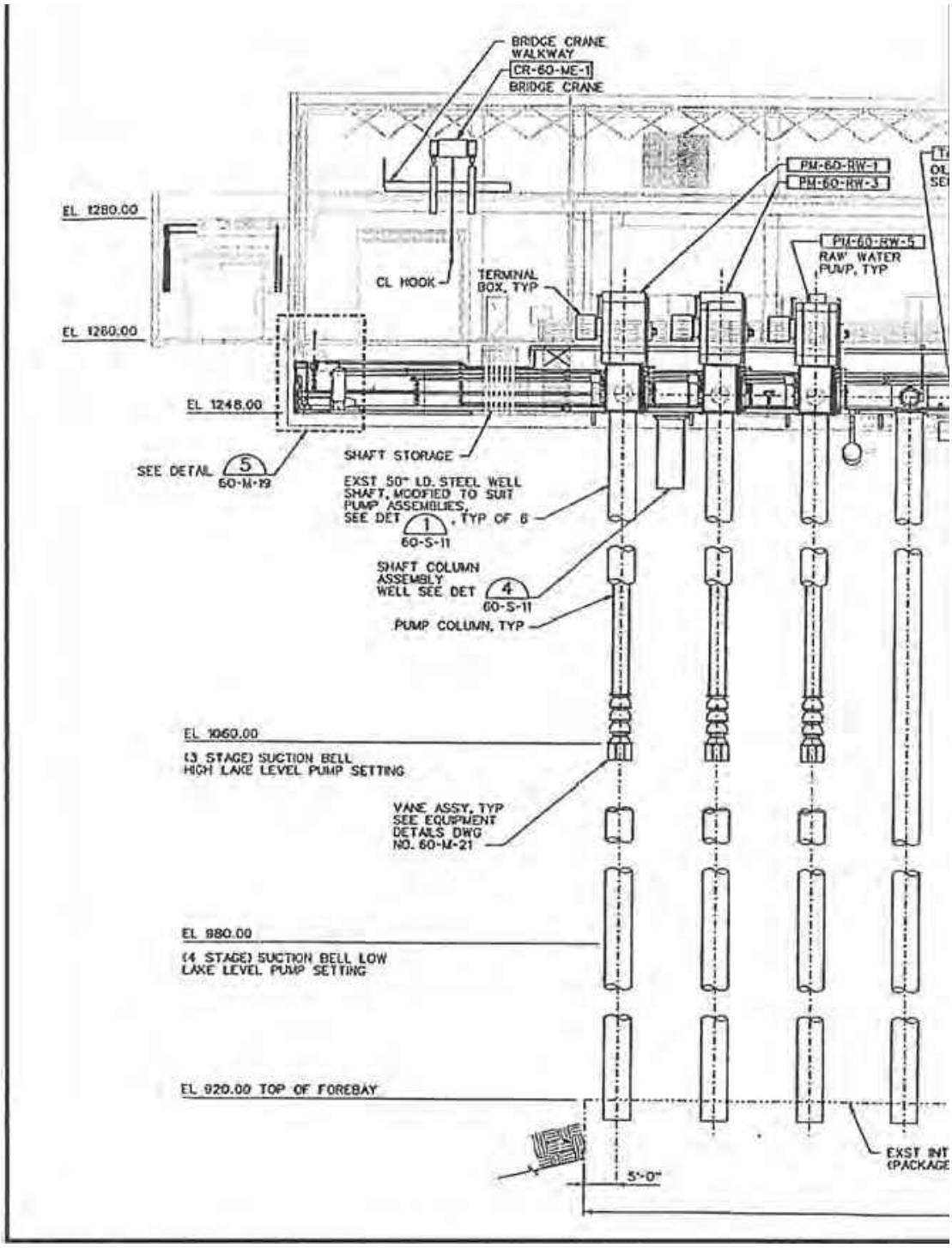


FIGURE 13 Typical section of a raw water pump station using vertical pumps (Southern Nevada Water Authority)

Groundwater is becoming a more attractive source for raw water as the supply of readily available surface water shrinks with demand. It will be recalled from the water inventory that fresh groundwater is much more abundant than surface water, yet it represents only 25% of the public water supply. Also, groundwater frequently requires less treatment than surface water. Although some groundwater sources are available at the surface in the form of artesian wells or springs, aquifers are chiefly accessed through wells drilled or dug down below the local water table. The pumping rate of wells is determined by the aquifer

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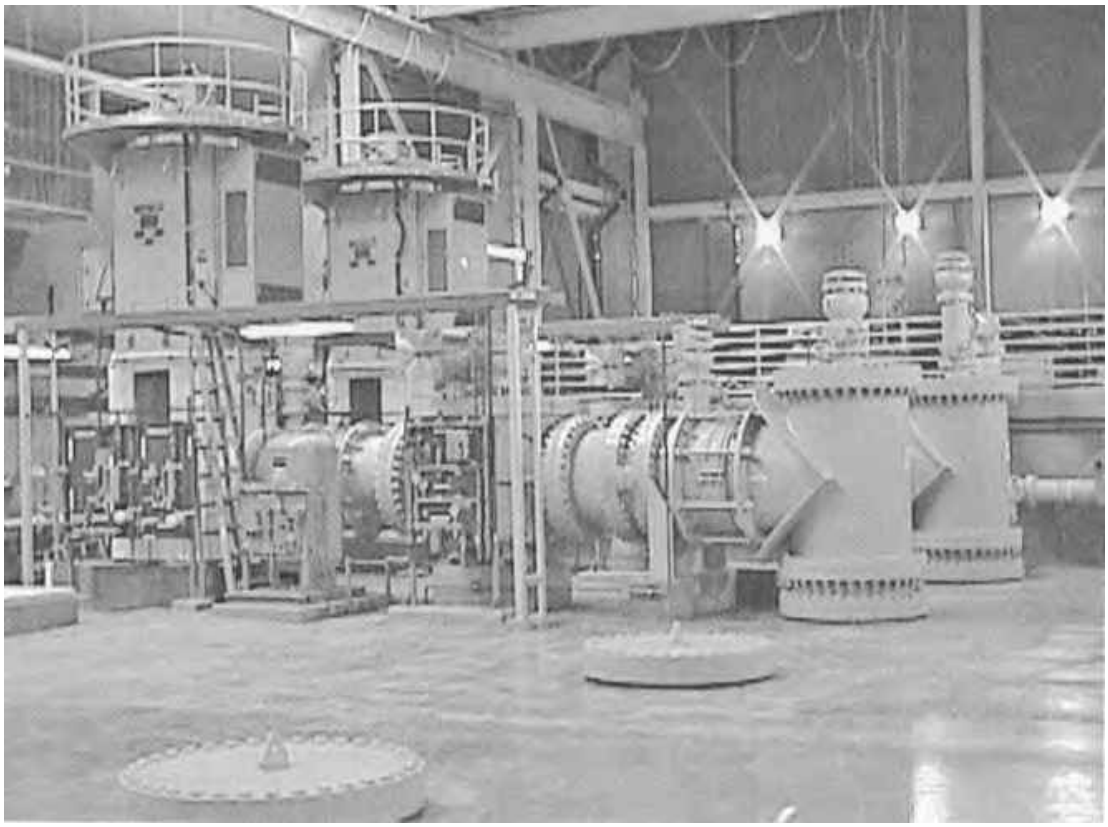


FIGURE 14 Typical raw water pump station with vertical pumps

activity, especially the recharge rate. If a well is pumped at a faster rate than the aquifer around it is recharged by precipitation or other underground flow, then water levels around the well can be lowered. For this reason, wells are drilled well below the "water table" to provide some protection against the lowering of the supply. Deeper wells also provide for some ready storage in the casing above the pump element.

Pumps for public supply groundwater are almost always of the multistage vertical turbine design. Smaller private wells may use the jet pump design, but the flow rates and lifts are limited (Chap. 7). The depth of public supply wells has been increasing recently with depths of 1000 ft (305 m) now becoming a more frequent occurrence. Depending on the geology of the field wells of this depth can present significant problems with straightness, both in drift (deviation from vertical) and spiral.

Vertical turbine pumps (VTPs) for deep wells come in two distinctly differing designs. The defining characteristic is the location of the drive motor. The actual pumping element of both types is essentially the same; a set of stages consisting of casings (bowls) and impellers mounted is series on a single shaft. The power can be delivered to the pump shaft either from below the element as in a submersible pump, so called because the motor is submerged, or from above as in a lineshaft pump, so called because the power is delivered from a motor mounted on the surface through a slender shaft.

Figure 15 shows a typical lineshaft design consisting of the submerged pumping element, the column pipe/shaft assembly, the discharge head and the surface mounted motor. The lineshaft pump design has been used for this application for many years, in fact the VTP design originated with well pump applications, but the deeper settings have associated problems that must be considered for each application. As with all pump designs the impeller creates an axial force that is applied to the pump shaft. In the case of lineshaft VTPs this axial force is transmitted to the surface and supported by a thrust bearing in the motor. The result of this force applied to a slender shaft of significant length is a growth in the length of the shaft. In deep-well applications this elongation can be several inches and special designs for the casings and impellers must be used to allow for this

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FIGURE 15 Vertical turbine, lineshaft-driven pump (Flowserve Pump Division)

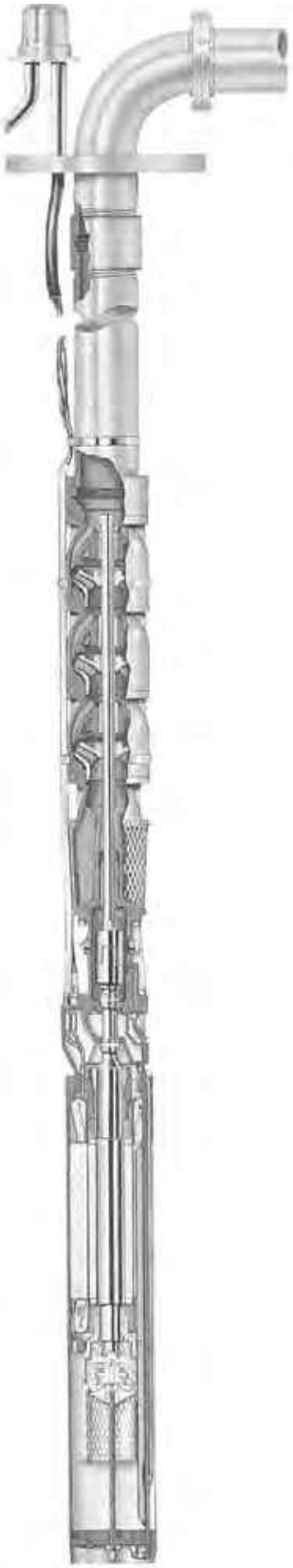


FIGURE 16 Vertical turbine, submersible motor driven pump (Flowserve Pump Division)

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movement of the rotor within the pumping element. The calculation of the shaft elongation is simple, as seen by the following formula:

$$Elong. = (TBH \times L_{col} \times K)/(A_{shaft} \times E)$$

where TBH = total "bowl head" = total developed head of multistage pump
L_{col} = length of column
K = impeller thrust constant [8,9]
A_{shaft} = shaft cross sectional area
E = shaft modulus of elasticity

This calculation will closely represent the elongation of the shaft itself, but other factors come into consideration. The column is subject to a portion of the weight of the water in the pump and will elongate some itself reducing the relative elongation of the shaft. Also, depending on the depth to the water level the head produced by the pump can vary affecting the value of the TBH.

The second variation of the VTP design and one that is growing in popularity with users is the submersible or submerged motor design. In this, as the name implies, the motor is located down in the well along with the pumping element, as shown in Figure 16. The motor is located at the bottom of the assembly with a special connector between it and the first stage pump case. A long cable from the surface transmits energy to the motor and the water is carried to the surface through standard, threaded pipe. This VTP variation, since it eliminates the lineshaft and absorbs the thrust in the motor bearing, does not require special case and impeller designs. The elimination of the lineshaft also reduces the number of maintenance parts and makes the unit less susceptible to bearing rubs and wear that can come from less than straight wells. As with any design choice, there are trade-offs that must be considered when deciding on the course to take. The location of the motor can cause higher maintenance costs if problems occur.

Hydraulic Gradient Whatever the source or pump style the primary task of the raw water pump is to withdraw the water from the source and deliver it to the treatment plant. Often this is a simple task of lifting the water to the plant where much of the process is performed using gravity flow. Raw water pumps in this service may be called *low lift* because the head required to feed the plant is not high. In other cases, such as river water intakes, the source may be at a lower elevation and the plant located at a higher elevation. In cases like this, the raw water system must be designed carefully to be certain the pumps will meet all the needs of the plant. The development of a hydraulic gradient plot (described in Section 11.1) is one method used to size these pump stations, sometimes referred to as high lift. A plot of hydraulic gradient for one system is shown in Figure 17. In this plot, the X-axis represents the distance from the source, a river pump station; and the Y-axis is the elevation of the transmission line from source to the delivery point, a reservoir at the treatment plant. Two lines are plotted on the chart. First is the profile of the raw water transmission line as it passes over the terrain. The upper line shows the head needed in the system to reach the reservoir with a positive head margin. The most important concept to be derived from this is that the head developed by the pumps must meet the difference between the minimum source level and the hydraulic gradient line for the selected line size at the entrance station to the pipeline. It cannot be stressed too strongly that the hydraulic gradient line must be above the system profile at all points. One pitfall of using a system head calculation that looks only at the source and delivery levels and the line friction without considering this chart was a real experience for one system designer. The pump station was complete and the day came to push the button. Everything seemed to go well, but no water was being delivered to the plant. The pumps were running, power was being consumed but the discharge gauge was reading much higher than expected. The problem was that between the pump station and the plant was a mountain, and the elevation of the pipeline as it passed over the mountain imposed a head on the pumps greater than that available, even at zero flow rate.

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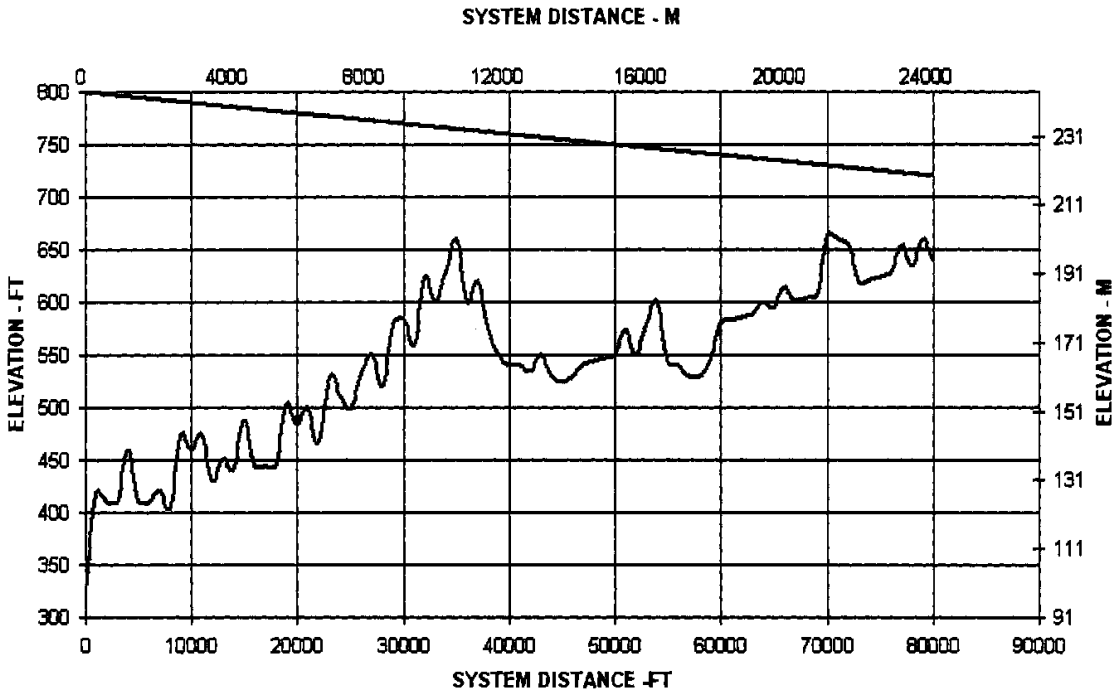


FIGURE 17 Typical hydraulic gradient plot

Transmission At first it was simple: population centers grew next to the best source of water, and when treatment plants were built they were also close to the water source. Cities grew and in many cases outgrew the capability of the local source to meet all the water needs. The need arose to locate, develop, and transmit water from new sources, often over long distances to the point of use. Many times the source of water was at a higher level than the point of use and gravity alone could deliver the flow. This is as old as the aqueducts of the Roman Empire and as new as New York City's Tunnel Number Three. Unfortunately, this is not always the case. Referring back to Figure 17, consider the profile being both longer and having multiple peaks. This is the prospect facing many utilities today, and the system needs far exceed the capability of a single pump station design. It becomes necessary to break up the raw water transmission line into two or more segments, adding a booster pump station at the junctions. Raw water booster pump stations can take many forms, but they commonly consist of a surge tank as a delivery point, a forebay to serve as the pump suction source, and the booster pumps. The layout of a typical raw water booster station using horizontal centrifugal pumps is shown in Figure 18.

Chemical Feed Pumps At the inlet to the treatment plant and continuing throughout the process, chemicals are added to the water for various purposes. Among these chemicals are the following:

POTASSIUM PERMANGANATE It is added to the water coming into the plant to remove iron, manganese, and objectionable tastes and odors. Because raw water is often drawn from below the surface of the source dissolved oxygen is low. When the oxygen level is low, iron and manganese are kept in solution. When iron and manganese are oxidized, they become insoluble in water and can then be removed by sedimentation and filtration. Potassium permanganate is also used to oxidize the organic contaminants that cause taste and odor problems.

ALUM Alum is added to the incoming raw water to support coagulation and flocculation reactions that bind together very fine suspended particles into larger particles that can be removed in the settling and filtration steps. The end result is to remove objectionable color and turbidity (cloudiness) from the drinking water.

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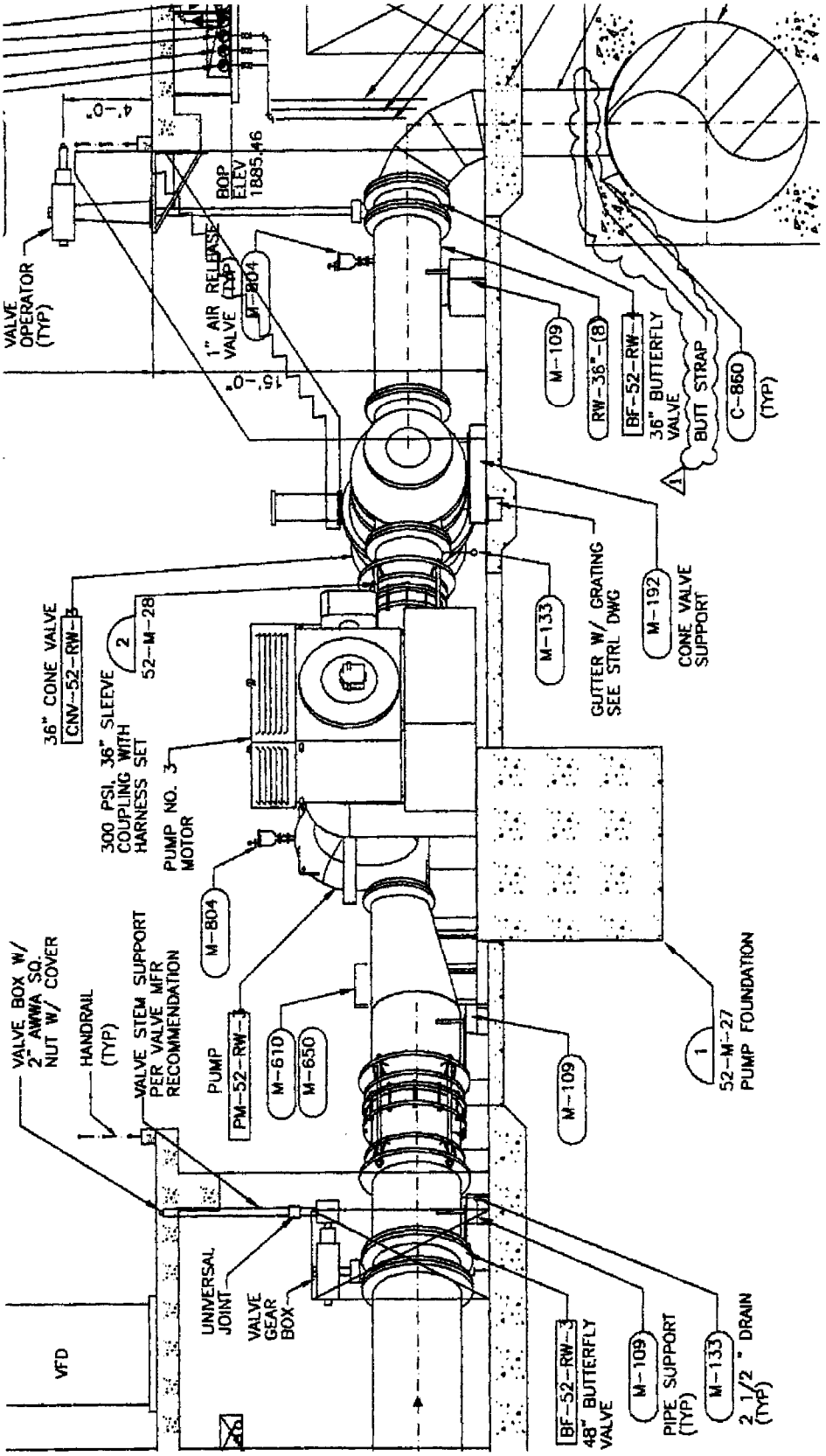


FIGURE 18 Typical section of a water booster pump station (Southern Nevada Water Authority)

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HYDRATED LIME Hydrated lime is an alkaline compound that is added to the water for pH adjustment. Since filter alum is an acidic salt, it lowers the pH of the water, and lime is used to neutralize this effect.

CHLORINE It is added to water to disinfect it or improve its quality. Chlorine destroys pathogenic microorganisms, oxidizes undesirable elements, such as iron and manganese, and reduces some tastes and odors. After these needs have been met, some residual chlorine remains in the water to protect it from further contamination until it reaches the customer's tap.

PHOSPHATES Phosphates are added to the water to isolate iron, manganese, and calcium to keep them in solution. It also prevents corrosion of water lines by coating them as the water passes through. Some utilities use this to prevent the leaching of lead from older distribution systems and domestic plumbing.

FLUORIDE Fluoride is added to water following the disinfection process for the purpose of preventing tooth decay. Fluoridation is usually accomplished by the addition of dihydrogen hexafluorosilicate, which decomposes in water, yielding fluoride ions.

CARBON Carbon is added as slurry to remove objectionable tastes and odors. The tiny carbon particles are tremendously absorbent, like sponges. The longer they stay in the stream, the more thorough their dispersion; the better the job they do in making the water pleasant and palatable.

POLYPHOSPHATE Polyphosphate, the last additive, is a material that coats the surfaces of pipes as it passes through them, and prevents the lead in old plumbing from leaching into the drinking water.

All of these chemicals must be added at the correct step in the process and at a closely controlled rate. The pumps used for this purpose are normally of the positive displacement type, so that the discharge rate can be controlled by the speed and operating time. The specific type of displacement pump used is a matter of preference, the plunger, diaphragm, and peristaltic pumps being frequently used. Examples of each type can be found in Chap. 3.

Filter Pumps Conventional rapid gravity sand filters with their multilayer technology are a low-pressure process and depend only on gravity flow from the clarifier. Some treatment plants use a pressurized form of these filters, which work on the same principle as rapid gravity filters, except that the filter medium is enclosed in a vessel and the water is forced through it under pressure.

Newer filtration processes also use media that depend on pressure differential to force the flow through the media. The principal new methods are membrane technologies such as reverse osmosis, mentioned earlier. In a reverse osmosis system, pressure is applied to water on one side of a semipermeable membrane. The process is termed reverse because the natural migration of water in an osmotic cell would be from the area of lower to higher concentration. Pressure must be applied to the "dirty" side of the membrane to produce the reverse flow and produce clean water. For instance, the osmotic pressure of ocean water is about 27 atm or 27 bar (400 psi). Reverse osmosis desalinators use pressures around 50 atm (750 psi) to produce freshwater from ocean saltwater. Multistage pumps of either the horizontal-type or closed-suction vertical "can" pumps are used in the service.

Wash Water Pumps Conventional sand filters do rely on pumps for one important function, washing the trapped solids from the filter media. Wash water pumps may be either horizontal or vertical depending on the configuration of the wash water tank. During normal operation a portion of the filtered water is diverted to a wash water holding tank. At regularly scheduled intervals, the wash water pumps are energized and pump water through a distribution system in the bottom of the filter causing an upward flow through the media and out into a collection system. The backwash rate is determined by the filter

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manufacturer and is based on the type of media and area of the filter surface. Typical backwash rates are 15 to 20 gpm/ft^2 of bed area (37 to 49 m^3/h per m^2), and backwash should be carried out for 8 to 10 min. In some systems air is first injected into the lower layer of the filter to agitate and loosen the filter media to make the backwash process more effective. Pump selection is important to the stability of the filter media. Backwash services generally require high flow rate and low head; so, pump performance curves must be selected to prevent any unsteady hydraulic behavior that might lead to the upset and mixing of the layers of the filter media.

High Service Pumps High service pumps are the final pumps at a water treatment plant placing the water from the clearwell into the distribution system. Either horizontal or vertical pumps can be used, based on the system requirements and the placement of the clearwell within the treatment plant. Horizontal pumps are normally used when the required head is low enough to be supplied by a single-stage pump. Pump stations for horizontal pumps must be located so that a positive head is delivered to the pump inlet that is sufficient to meet the NPSH required by the pump. Often this means excavating so that the pumps are below the existing grade level as shown in Figure 19.

In this pump station the suction well is pressurized by the level of the water in the clearwell. The bottom suction horizontal pump allows for a simple piping arrangement on the station main floor with all maintenance performed in accessible areas.

Vertical pumps are typically used when heads required are more than that available from single-stage pumps. The flexibility of the vertical, multistage design adapts well to the varying demands placed on high service pumps. A typical vertical pump installation is shown in Figure 20.

Vertical pumps also allow the station designer to maintain a simple piping arrangement. The one drawback to this design is the removal of the pump for maintenance. The station must be designed with sufficient overhead space and crane capacity to remove the pump from the suction well. Costs related to maintenance can be higher than those for horizontal pumps.

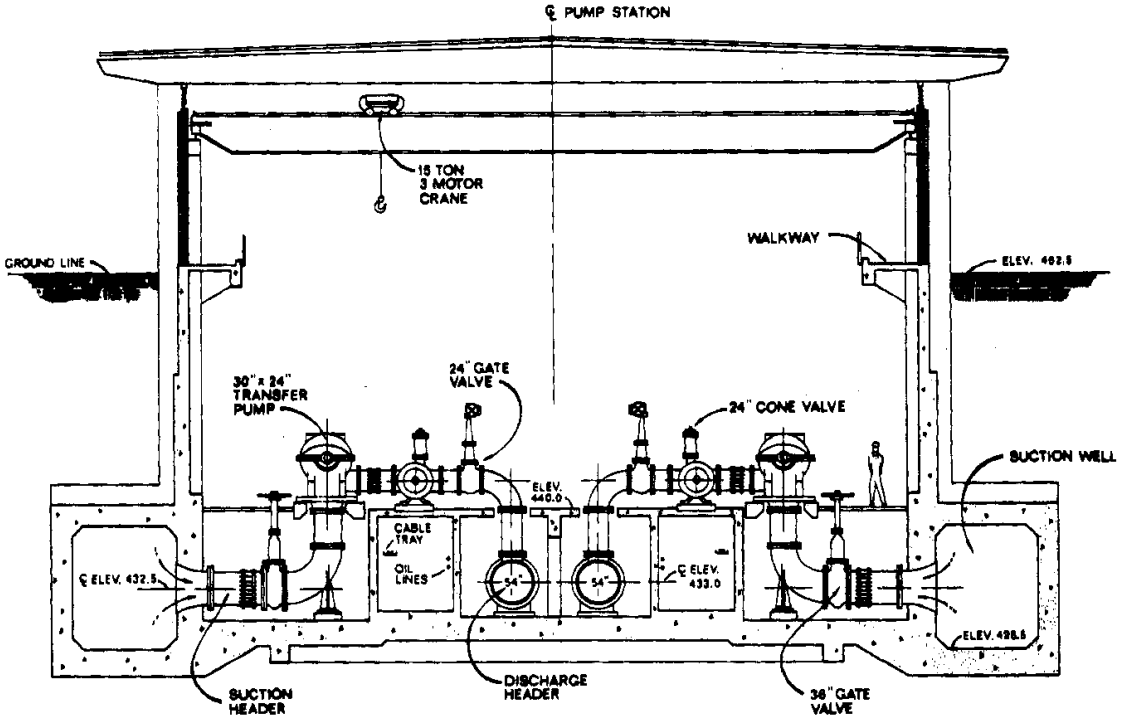


FIGURE 19 Typical section of the high service pumping station of the East Side water treatment plant in Dallas (1 in = 2.54 cm; ft = 0.3048 m) (URS/Forrest and Cotton)

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FIGURE 20 High service pump station with vertical pumps (Flowserve Pump Division)

The design conditions for high service pumps are not easily determined. In some smaller systems all the user needs can be met by a single set of pumps operating at a constant speed with demand variations being met by selecting the number of pumps on line and by elevated storage tanks and water towers in the system. As systems become more complex and user patterns more varied the nature of the analysis needed for the high service pump sizing becomes much more complicated. Meeting the peak hourly demand, which may exceed 350% of the average daily demand rate (see Figure 10) and be sustained for several hours (see Figure 9), calls for a balance of treatment plant sizing, system storage and pumping station capability. The primary choice required is the balance between the portion of the peak hourly demand to be met by direct pumping from the treatment plant and that supplied from storage. This choice not only affects the high service system sizing but, projected back upstream, the design capacity of the treatment plant also. The need for accurate input into this decision cannot be overstressed. The most common process for providing this information has been a system head analysis looking at all of the possible variables. (See Sections 11.1 and 11.2 in this book.) The result of one such analysis is shown in Figure 21. In this example, three system head curves are shown for the "equalization" hour, demand and design rates equal, "peak" hour, demand above design, and "refill" hour, demand below design. Superimposed on these three curves are additional curves reflecting the performance of a set of six pumps driven by two-speed electrical motors.

An upgrade to the pumping system to provide more complete system coverage would be to step up from two speed motors to a variable speed system. This could be done to one, two, or all of the pumps and makes the system much more flexible, allowing more than one selection to meet the critical operating conditions. By careful selection of the pump design conditions the designer can cover all of the possible combinations needed to meet the system demand. This type of analysis is good for one given design and, while the numbers of pumps can be varied with relative ease, requires a redevelopment of the system data for

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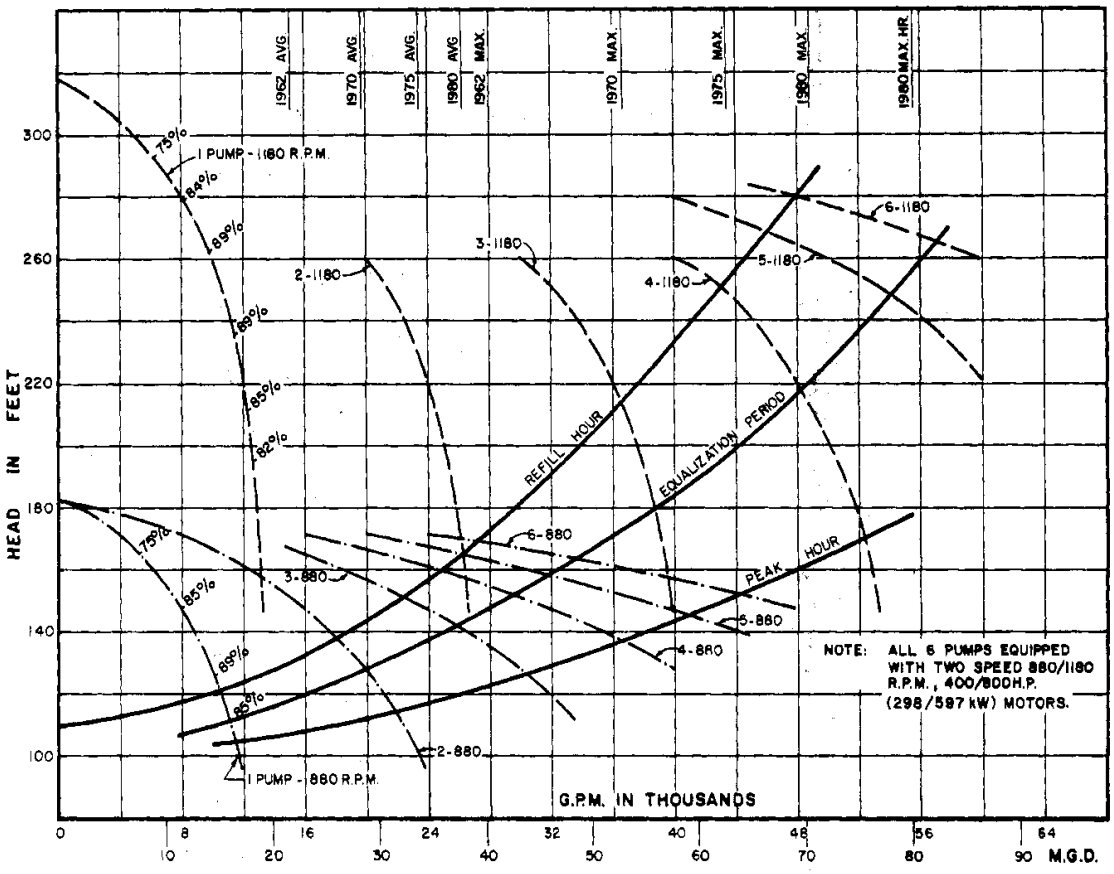


FIGURE 21 Typical system head curves for high service pumpage to water distribution system (1 ft = 0.3048 m; 1 gpm = 0.227 m^3/h)

any proposed system modifications. The importance of this information and the complexity of the analysis along with the need for flexibility in quickly evaluating system options have led to the development of computer software for distribution system design. As a quick search of the Internet will confirm there are many choices available to the system designer. Some programs are available free of charge and provide a basic analysis capable of an extended period hydraulic analysis package which can perform the computations described in Section 11.1, including the ability to:

- handle systems of any size
- compute friction head loss using the Hazen-Williams, Darcy-Weisbach, or Chezy-Manning formulas
- include minor head losses for bends, fittings, and so on
- model constant or variable speed pumps
- compute pumping energy and cost
- model various types of valves including shutoff, check, pressure regulating, and flow control valves
- allow storage tanks to have any shape (e.g., diameter can vary with height)
- consider multiple demand categories at nodes, each with its own pattern of time dependence
- model pressure-dependent flow issuing from emitters (sprinkler heads)
- base system operation on simple tank level or timer controls as well as on complex rule-based controls

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FIGURE 22 Exterior and interior of a typical community booster station (Flowserve Pump Division)

Evaluation of the various packages is beyond the scope of this document and left to the user. The basic functions appear to be the same with many options in the area of output and graphics.

Distribution Booster Pumps The addition of branches to a distribution system (cf. Section 11.2) as it reaches out to new housing developments often results in a loss of service pressure. The utilities goal should be to maintain normal service pressure delivery to the user is between 40 and 90 psig (2.8 and 6.2 barg). Where the existing system falls below this value a booster pump station must be installed. Typically booster stations include a storage reservoir to accept the incoming flow and a set of pumps to supply the new branch system. The pumps are controlled by the upstream pressure and meet the varying demand through variable speed or multiple pump operation. The design of these stations is often very sensitive to the location. Figure 22 shows the exterior design of one such station that was designed to conform to the general local architecture. An interior view of this building is also shown in Figure 22.

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SECTION 12.2
SEWAGE TREATMENT

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Sewage is defined as the spent water of a community. Although it is more than 99.9% pure water, it contains wastes of almost every form and description. Raw sewage, when fresh, is gray and looks something like dirty dishwater containing bits of floating paper, garbage, rags, sticks, and numerous other items. If allowed to go stale, it turns black and becomes very malodorous. About 25% of the waste matter of normal domestic sewage is in suspension; the remainder is in solution.

Sewage contains many complex organic and mineral compounds. The organic portion of sewage is biochemically degradable and, as such, is responsible for the offensive characteristics usually associated with sewage. Sewage contains large numbers of microorganisms, most of which are bacteria. Fungi, viruses, and protozoa are also found in sewage, but to a lesser extent. Although most of the microorganisms are harmless and can be used to advantage in treating the sewage, the viruses and some of the bacteria are pathogenic and can cause disease.

Sewage flow generally averages between 50 and 200 gallons per capita per day (gpcd) (190 and 760 lpcd). In the absence of better information, an average figure of 100 gpcd (380 lpcd) is generally used for design purposes. The rate of flow usually varies from minimum in the early morning to maximum in the later afternoon. Minimum flow ranges from 50 to 80% and maximum dry-weather flow from 140 to 180% of average flow. The extent of variation decreases as the size of the system increases. Wet-weather flows can be 600 gpcd (2270 lpcd) or more because of the extraneous water entering sewers from roof drains, areaway drains, footing drains, and so on.

SEWAGE SYSTEMS

In most instances, sewage systems are divided into two parts: collection systems and treatment systems.

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Collection Collection systems consist of a network of sewers that collect and convey sewage from individual residences, commercial establishments, and industrial plants to one or more points of disposal. Pumping stations are often needed at various points in the system to pump from one drainage area to another or to the treatment plant. The judicious location of pumping stations enhances the economy of the overall design by eliminating the need for extremely deep sewers.

Small-system pumping stations are frequently built underground and may be factory-built. For larger stations, superstructures should be in keeping with surrounding development. It has been said that people smell with their eyes and their ideas as well as their noses, and for this reason aboveground structures should be attractive, with landscaped grounds, to overcome the popular prejudices against sewage works. Stations can be and have been designed and constructed in residential areas where the neighbors apparently are not aware that the stations are not homes.

Treatment Treatment facilities can be many and varied, with the extent and nature of the treatment determined to a large degree by the proposed use of the receiving stream and its ability to assimilate pollutants. Most conventional treatment plants being built today can be classified as either primary, biological, or advanced waste treatment. Other alternatives, such as physical-chemical or chemical-biological treatment, are also used on occasion but on a lesser scale. The treatment needs of smaller communities are sometimes satisfied by package treatment plants or by waste stabilization lagoons.

Primary treatment involves removal of a substantial amount of the suspended solids but little or no colloidal or dissolved matter. Primary treatment facilities normally include screening, grit removal, and primary sedimentation. The sewage is often chlorinated during primary treatment in order to sterilize the wastes.

Biological treatment uses bacteria and other microorganisms to break down and stabilize the organic matter. Trickling filters and the many variations of the activated sludge process are the most popular biological treatment concepts presently in use. Biological treatment is generally followed by final sedimentation of the solids produced by the microorganisms.

Advanced waste treatment is a very complex subject, and it can range from a limited objective, such as phosphate removal, to whatever additional treatment is necessary for water reuse purposes. Advanced waste treatment usually follows conventional primary and biological treatment and can include phosphate removal, nitrate removal, multimedia filtration, carbon absorption, and ion exchange. Where zero discharge is required, it may be necessary to follow advanced waste treatment with spray irrigation of the plant effluent or other methods of disposal.

Combined primary and biological treatment using the activated sludge process is perhaps the most commonly used treatment concept currently in use. A schematic drawing of a typical activated sludge treatment plant is shown in Figure 1. In the example, liquid treatment is accomplished by coarse screening, grit removal, fine screening (or communication), and primary settling, followed by aeration, final settling, and chlorination. Sludge processing includes thickening, dewatering, incineration, and liquid disposal of ash. There are many variations to this layout, but the one shown includes most of the pumping applications normally encountered in treatment plant design. Pumping requirements will of course vary from plant to plant, depending on the process used, the site size and topography, and the relative location of the various structures and equipment.

PUMP APPLICATIONS

Most of the pumping applications associated with the collection and treatment of sewage can be classified according to the general nature of the liquid to be handled. The primary classifications are (1) raw sewage, (2) settled sewage, (3) service water, and (4) sludge. There are also, however, a number of specialized applications involving the handling of abrasive materials, such as grit and ash. The types of pumps recommended for sewage applications are indicated in Table 1. Although included in the table, chemical pumps are not discussed in this section. They are covered in Section 12.7.

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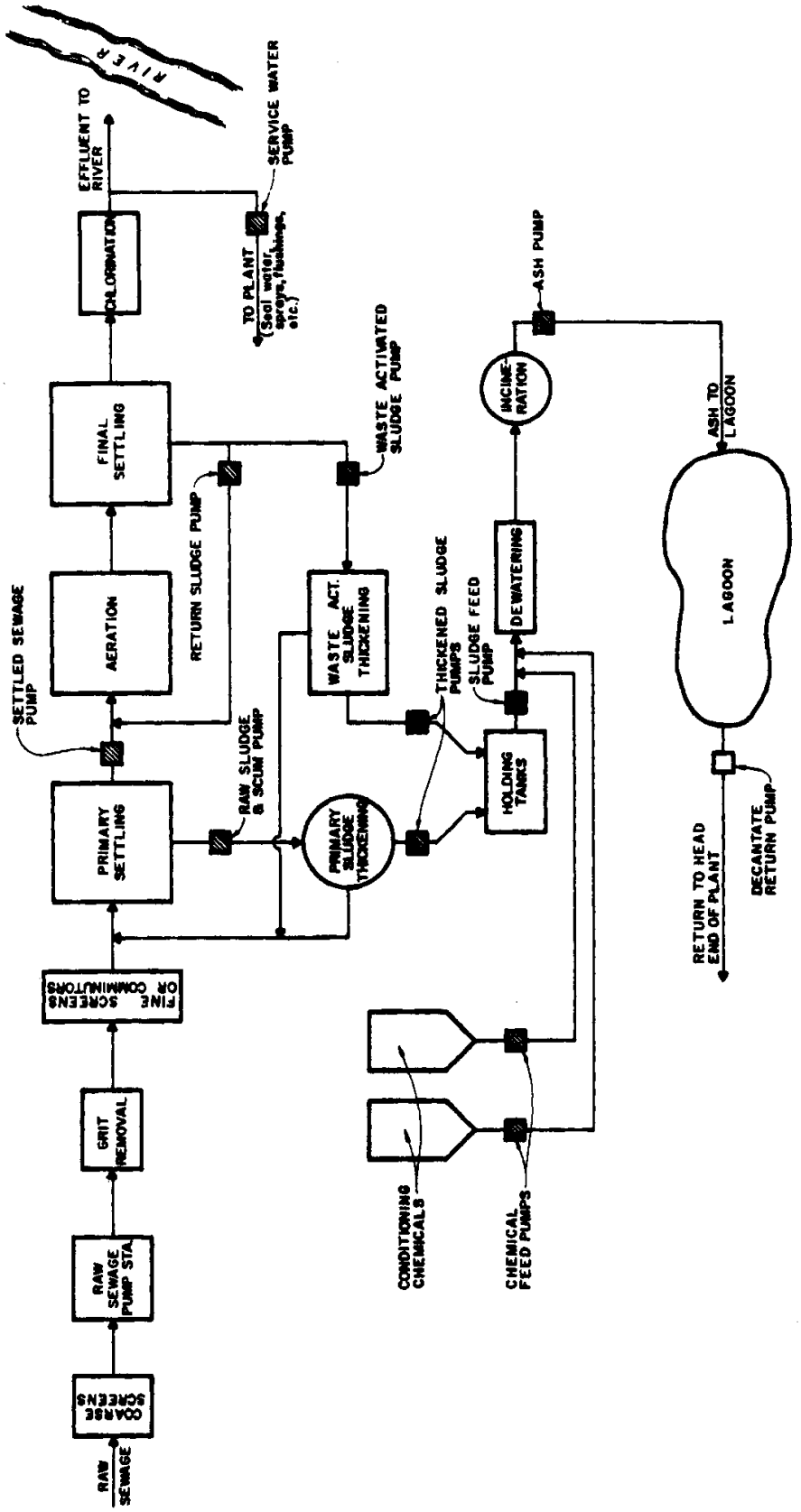


FIGURE 1 Typical activated sludge plant, with dewatering, incineration, and liquid disposal of ash

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TABLE 1 Types of pumps generally used in sewage applications

Application	Conventional sewage	Diffuser	Torque flow	Clear-water volute	Ash	Screw	Pneumatic ejector	Air lift	Positive displacement	Chemical
Raw Sewage	X	...	...	...	...	X	X	...	...	...
Grit	...	...	X	...	...	...	...	X	...	...
Primary sludge										
Less than 2% solids	X	...	X	...	...	...	...	...	...	...
More than 2% solids	...	...	...	...	...	...	...	...	X	...
Normal primary scum	...	...	...	...	...	...	...	...	X	...
Diluted scum	X	...	...	...	...	...	...	...	...	...
Biological sludge	X	X	...	...	...	X	...	X	...	...
Thickened biological sludge	...	...	...	...	...	...	...	...	X	...
Digested sludge, recirculation	X	...	X	...	...	...	...	...	...	...
Settled sewage	X	X	...	...	...	X	...	...	...	...
Plant effluent	X	X	...	...	...	X	...	...	...	...
Service or nonpotable water	X	X	...	X	...	...	...	...	...	...
Ash sluice	...	...	...	...	X	...	...	...	...	...
Decantate or supernatant liquor	X	X	...	...	...	...	...	...	...	...
Chemical solution	...	...	...	...	...	...	...	...	...	X

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Raw Sewage Raw sewage pumps are used to lift liquid wastes from one level of the collection system to another or to the treatment plant for processing. Regardless of where the pumps are located, the basic design considerations remain the same.

Even though the sewage is normally screened at larger installations before entering the suction wet well, it still contains a large quantity of problem material, such as grit, rags, stringy trash, and miscellaneous solids small enough to pass through the coarse screens. Screens are often omitted from smaller installations because large objects are not as much of a problem because of the smaller size of the incoming sewers.

Raw sewage pumping installations are usually sized so their firm capacity either is equal to a future maximum flow rate of the incoming sewers or can be expanded to accommodate this level. *Firm capacity* is defined as total station capacity with one or more of the largest units out of service.

Pneumatic ejectors are sometimes used where the required capacity is less than that provided by the smallest conventional sewage pump. This type of unit, however, should not be used where more than 50 connections are expected.

Conventional sewage pumps are, by far, the most common pumps used for the handling of raw sewage. A conventional sewage pump is more specifically described as an end-suction, volute-type centrifugal with an overhung impeller of either the nonclog (Figure 2) or the radial- or mixed-flow type (Figure 5), depending on capacity and head.

Nonclog pumps are all based on an original development by Wood at New Orleans. Actually, no pump has been developed that cannot clog, either in the pump or at its



FIGURE 2 Nonclog dry pit sewage pump (Flowserve Corporation)

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appurtenances. Experience shows that rope, long stringy rags, sticks, cans, rubber and plastic goods, and grease are most conducive to clogging.

Nonclog impellers are used almost exclusively today for pumps smaller than 10 in (25 cm). These pumps differ from clear-water pumps in that they are designed to pass the largest solids possible for the pump size. The conventional nonclog impeller contains two blades, although some manufacturers are now offering a single-blade ("bladeless") impeller. The two-blade impeller has thick vanes with large fillets between the vanes and the shroud at the vane entrance. The bladeless impeller has no vane tips to catch trash. On the other hand, it is inherently out of balance because of its lack of symmetry.

The larger raw sewage pumps are equipped with either mixed-flow or radial-flow impellers, depending on head conditions. Both have two or more vanes, depending on pump size and the size of solids to be handled. The vane tips are sharper than for the nonclog impeller, resulting in a higher operating efficiency. The heavier vanes are not necessary because the vane openings can be larger than on the smaller pumps. Experience indicates that stringy trash will not clog an impeller with vane openings larger than 4 in (102 mm) in diameter.

Conventional solids handling volute-type sewage pumps may be of the dry pit type (Figure 2), wet pit type (Figure 3), or submersible type (Figure 4). Dry pit pumps are by far the most popular and widely used type due to their accessibility for observing mechanical operation and ease of maintenance when necessary.

Wet pit solids handling pumps are gaining in popularity especially on return activated sludge services. They are similar in design to a vertical turbine or mixed-flow diffuser pump except the bowl contains a limited number (usually two) of well-rounded diffusers,



FIGURE 3 Solids handling wet pit pump (Flowserve Corporation)

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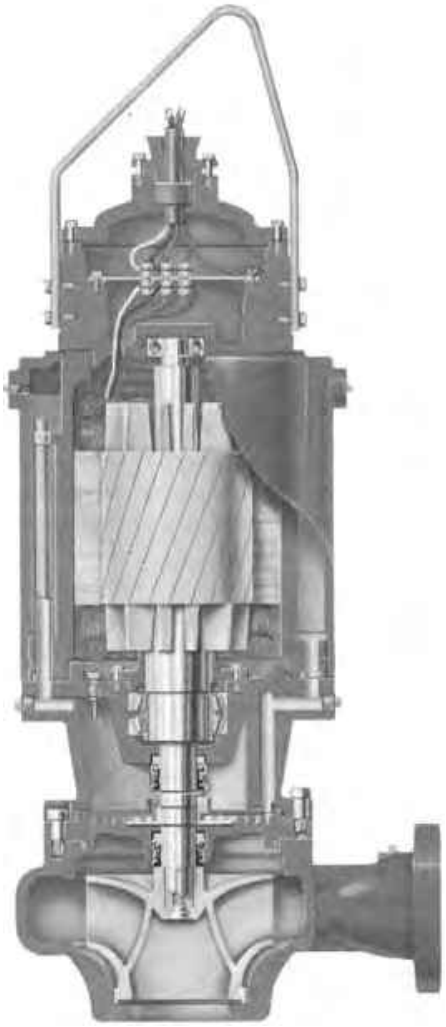


FIGURE 4 Nonclog submersible sewage pump (Flowserve Corporation)

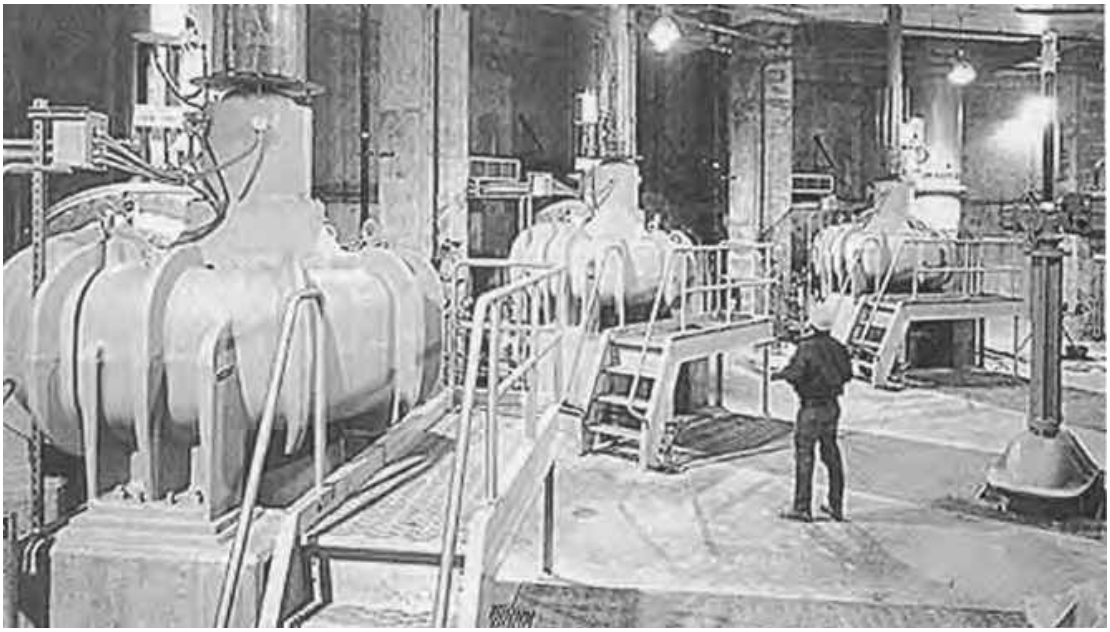


FIGURE 5 Vertical sewage pumping units at the South System Pump Station, Deer Island in Boston, MA, containing (8) 36 in (915 mm), 46,300 gpm (10,510 m^3/h) pumps driven by 1250 hp (932 kW) variable speed motors (Flowserve Corporation)

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the impeller is of the solids handling type, and there is no lower suction bell bearing. Therefore, they are not nearly as likely to clog on the debris normally found in raw sewage. Single-stage models available at the time of this publication are limited to operational total heads under 100 ft (30 m).

Submersible solids handling pumps are commonly used in small raw sewage lift stations. They are most widely used in sizes 12 in (305 mm) and smaller with total heads under 100 ft (30 m) and motor ratings under 150 HP (112 kW).

Although nonclog pumps 8 in (203 mm) and small are available with self-priming (Section 2.2.2), most conventional sewage pumps are located so the impeller is always below water level in the suction wet well. This eliminates the need for specialized priming systems.

Self-priming pumps have been used successfully to pump raw unscreened sewage, particularly in the southern part of the United States. The self-priming feature eliminates the dry-pit cost and gives the centrifugal pump the gas-handling advantage of positive displacement pumps. Operating costs are higher, though, because the design efficiencies generally run about 10 to 15% lower than for the conventional nonclog units.

Archimedean screw pumps (Figure 6) are occasionally used for raw sewage pumping applications. These units are advantageous in that they do not require a conventional wet well, and they are self-compensating in that they automatically pump the liquid received regardless of quantity as long as it does not exceed the design capacity of the pump. This is done without the need for variable speed drive equipment. Also, as shown by Figure 6, the total operating head of a screw pump installation is less than for those pumps that require conventional suction and discharge piping. Screw pumps, however, have a practical limitation as to pumping head. Generally speaking, they are not used for lifts in excess of 25 ft (7.6 m).

Settled Sewage Settled sewage pumps are used to lift partially or completely treated waste from one part of the plant to another or to the receiving stream. In Figure 1, these pumps are the settled sewage pump, service water pump, and decantate return pump.

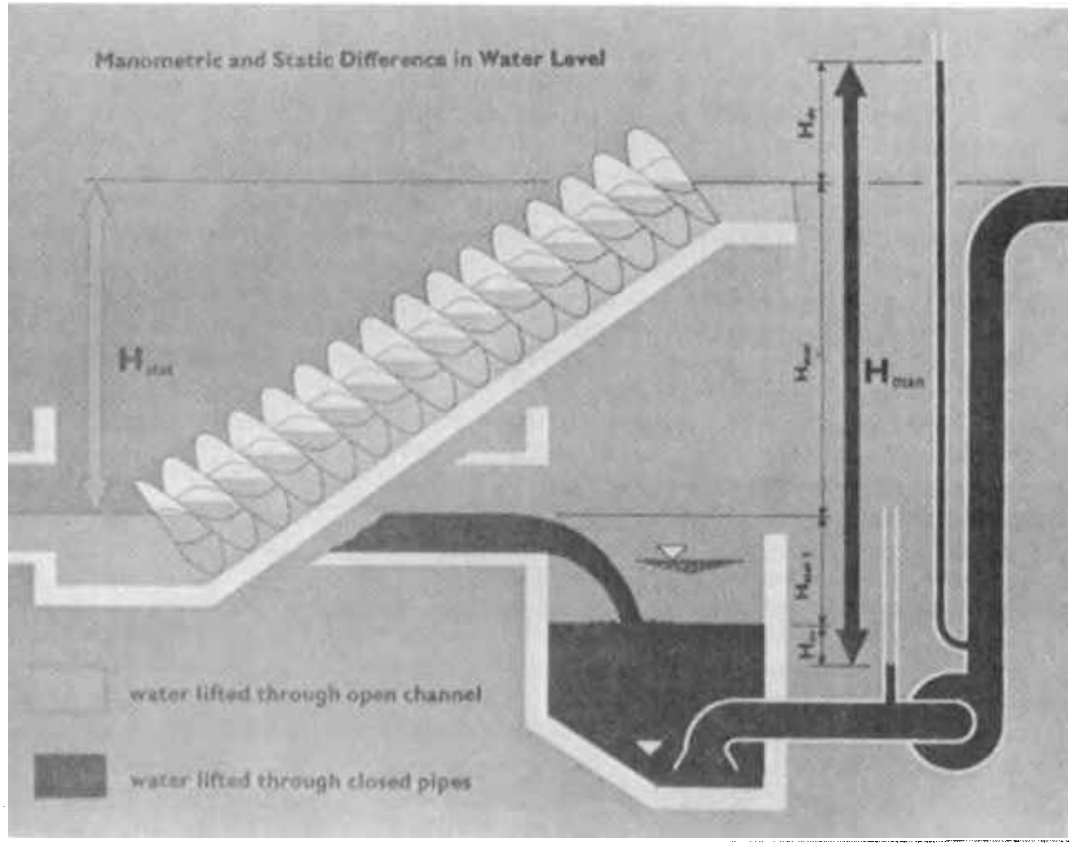


FIGURE 6 Archimedean screw pump (U.S. Filter/Zimpro)

The liquid to be handled usually contains some solids, but grit and most of the rags and other stringy material have already been removed. Sufficient firm capacity should be provided to meet peak flow requirements. In no case should fewer than two units be provided.

Wet pit solids handling or diffuser pumps (Section. 2.2.1) are commonly used for the pumping of settled sewage. Depending on the total head conditions and degree of solids removal, a diffuser pump selection may be of either the propeller or mixed-flow design. However, solids-handling wet pit pumps are gaining in popularity due to their greater freedom from clogging. Although normally installed in wet-pit applications, these units are sometimes mounted on suction piping and installed in a dry pit. Either type of application is acceptable, although economics usually dictates a wet-pit installation. Head and capacity conditions will determine which type of unit is applicable.

Conventional sewage pumps may also be used to pump settled sewage. They may be of the dry pit, wet pit, or submersible type. It is not usually as economical to design a dry pit for this application, but it is acceptable as far as suitability of equipment is concerned.

Archimedean screw pumps can be used to pump settled sewage, provided the lift is not excessive. As previously noted, this type of pump has certain inherent advantages.

Service Water Plant effluent water is frequently used for flushing, gland seal, foam control sprays, chlorine injector operation, lawn sprinkling, fire protection, and various other services in a waste-water treatment plant. Except for the fact that some solids must be contended with, this application is much the same as that found in building-water supply and small distribution systems.

Screening of solids is normally required; this can be accomplished either before or after the pumps, depending upon various circumstances. Pipeline-type strainers are recommended as they are not only economical but require a minimum of space, can be automatically backflushed, and are much easier to operate than alternative equipment.

Any type of conventional volute or diffuser clear-water pump can be used on service water applications, provided the effluent water is screened prior to entering the pump. Pumps capable of handling some solids should be used in those instances where prescreening is not practical.

Sludge and Scum This classification is divided into two separate categories, based on the concentration of solids in the liquid to be handled. Specialized pumping equipment is required for more concentrated sludges, whereas pumping of dilute sludge and scum is somewhat comparable to the handling of settled sewage.

DILUTE SLUDGE OR SCUM For the purposes of this discussion, *dilute sludge* and *scum* is defined as having less than 2% solids. An exception is digested sludge recirculation, which generally exceeds the 2% limit. This is included along with the more dilute sludges because the same type of pumping equipment is used.

Normally, the handling of dilute sludge is limited to the transfer of biological sludge back to the treatment process or to some other point for further concentration or dewatering and disposal. When digesters are used as part of the treatment facilities, sludge is often recirculated through external heat exchangers in order to maintain temperatures conducive to anaerobic bacterial action. This recirculation also helps keep the contents of the digester mixed. Occasionally, primary sludge and scum are handled in diluted form.

The firm capacity of dilute sludge pumping facilities should be equal to anticipated peak loading. Biological sludge return pumps should have a capacity range from 25 to 100% of average design raw sewage flow to the plant. Digested sludge recirculation pumps should be sized to turn over the contents of the digester frequently enough to maintain the desired temperature. Diluted primary and waste biological sludge pumps should have sufficient capacity to handle peak sludge loading at conservative solids concentrations.

Conventional sewage pumps are suitable for handling dilute sludge and scum. Either the nonclog or mixed-flow impeller may be used, depending upon capacity requirements.

Diffuser pumps are particularly suitable for handling biological sludge that does not contain any appreciable amount of trash or stringy material. They are not recommended, however, for handling diluted scum or for recirculating digested sludge. Depending on capacity requirements, diffuser pumps may be of either the mixed-flow or propeller design.

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Wet-pit applications are most common, although dry-pit installations are occasionally used.

Torque flow (or vortex) pumps are often used to handle dilute sludges that contain some grit. These units are particularly suitable for this type of service because their design is such that close running tolerances are not required; this allows the use of specially hardened materials, such as high-nickel iron, which are not easily machined. The most common applications of torque flow pumps are for the pumping of nondegritted dilute primary sludge to gravity thickening and the recirculation of digested sludge.

Screw pumps can be used in certain instances for handling biological sludge. Use of screw pumps is generally limited to low to medium lifts and to those instances where the point of discharge is close to the sludge source.

Air-lift pumps are suitable for transferring biological sludge where the lift is small and the point of discharge nearby. A typical air-lift pump installation is shown in Figure 7. Total head should not exceed 4 to 5 ft (1.2 to 1.5 m). The ability of an air-lift pump to vary capacity is somewhat limited, ranging from about 60 to 100% of the rated amount. These pumps are inexpensive in first cost but have an operating efficiency of only about 30%. They are very easy to install, and maintenance is minimal because there are no moving parts. Air-lift pumps are commonly used to transfer sludge at package treatment plants.

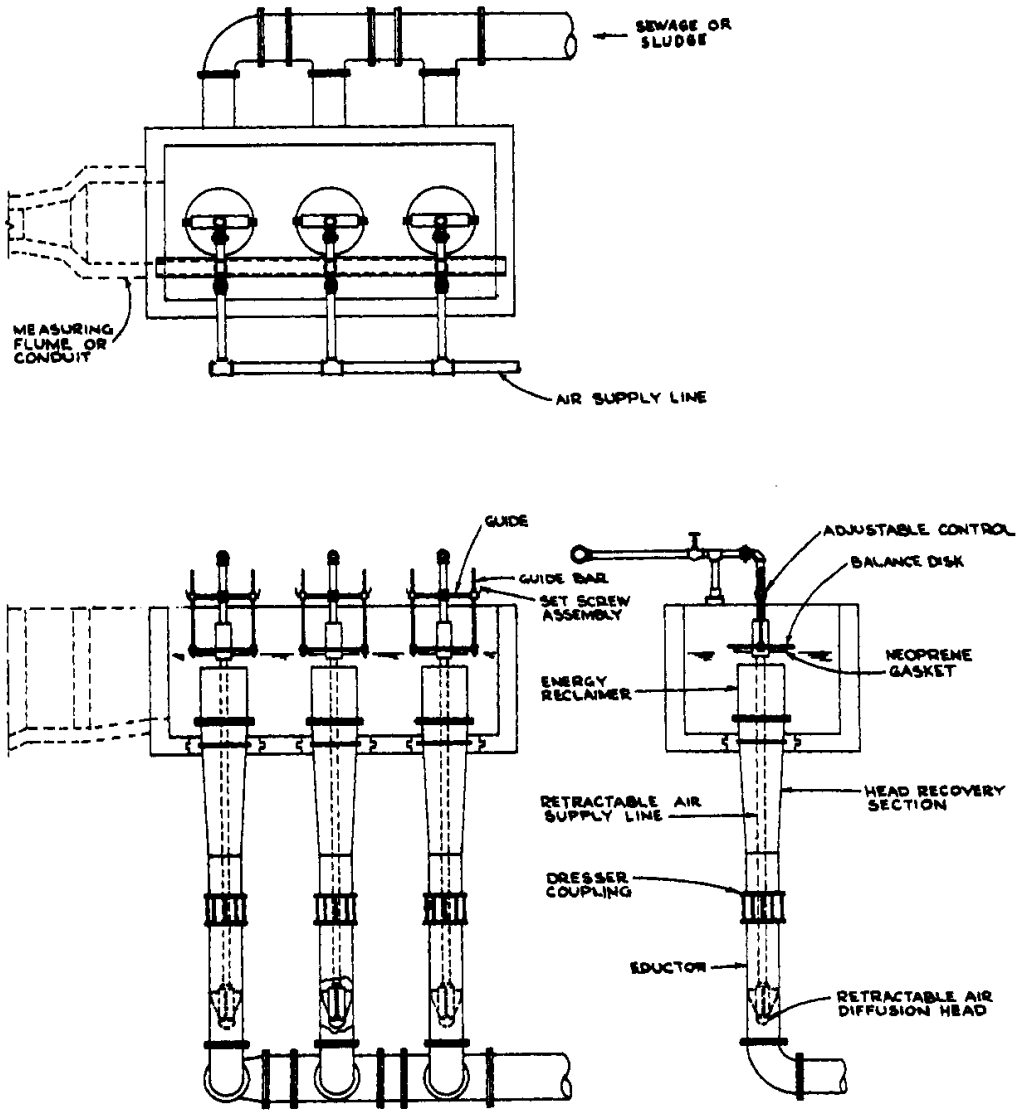


FIGURE 7 Typical air-lift pump installation (Walker Process Equipment Division, McNish Corp.)

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CONCENTRATED SLUDGE OR SCUM Concentrated sludge or scum is defined as having more than 2% solids. The single exception is in the case of the recirculation of digested sludge. As previously discussed, this has been included in the dilute sludge classification.

Each pumping installation should have enough firm capacity to handle peak design sludge quantities while operating part-time. The proportion of operating time at peak loading should vary from about 25% for primary sludge pumps to close to 80% for pumps feeding dewatering equipment.

Only positive displacement pumps are recommended for handling concentrated sludge and scum, mainly because they can pump viscous liquids containing entrained gas without losing prime. Also, these materials are thixotropic, and conventional formulas for frictional losses are not always valid. An arbitrary allowance of at least 25 lb/in^2 (170 kPa) should be added to the pumping head calculated by conventional methods to allow for changes in viscosity and partial clogging of pipelines. Positive displacement pumps are able to maintain a relatively constant capacity regardless of variations in discharge head.

For most applications, positive displacement pumps may be of either the plunger or the progressing cavity design (Figure 8). The performance of both depends upon close running clearances; consequently they have a high incidence of maintenance, especially where gritty substances are encountered. Even so, they represent the best pumping equipment currently available, and both designs have been used with success. Lobe-type gear pumps have been used for specialized applications. These are to be avoided, however, where there is any possibility that the material to be pumped will contain even a small amount of grit.

Plunger pumps should be of the heaviest design available and should be rated for capacity at about one-half of full stroke. The shorter the stroke, the more stable the operation and the less maintenance required. Heads as high as 80 to 100 lb/in^2 (550 to 690 kPa) are available and should be specified in order to give as much flexibility as possible.

Specially designed progressing cavity pumps are available for handling sewage sludges. Wear increases along with pump speed, and so excessive speed should be avoided. Ideally, the maximum speed of a progressing cavity pump should not exceed 350 rpm. These units are readily available with head capabilities up to 50 lb/in^2 (345 kPa) and should be so specified.

Certain of the newer sludge conditioning and dewatering processes, such as heat treatment and pressure filtration, require pumps having a head capability in excess of 500 lb/in^2 (3450 kPa). This is extremely difficult service, and special care should be taken in selecting the type of equipment to be used. So far, this area of application has received very little consideration from pump manufacturers.

Other Uses Grit may be handled with reasonable success with either a torque or an air-lift pump. Considerable flushing water is required with a torque flow pump, and to a lesser extent with an air-lift unit. A special ash pump is required where it is necessary to dispose of incinerator residue in a liquid form. These units are especially designed for ash sluicing service and are made of special hardened metals. No other pump should be considered for this service.

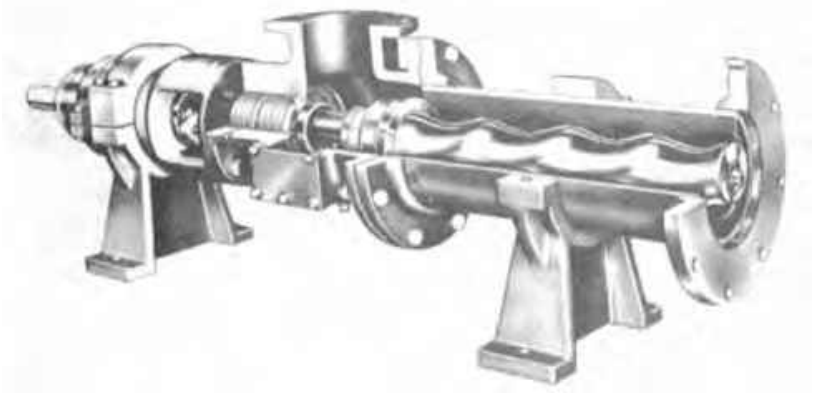


FIGURE 8 Progressing cavity sludge pump (Mono Pumps Ltd.)

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PUMP SELECTION

Various factors should be considered when selecting pumping equipment. These include the number of units to be installed, operating frequency, and station reliability requirements. After these factors have been fully evaluated, head-capacity curves should be prepared in order to match the pumps properly with system requirements. This is necessary because the capacity of most pumps varies with the total head at which the unit operates. When a pump is referred to as having a certain capacity, this capacity applies to only one point on the characteristic curve.

Number of Pumps The number of pumps to be provided at a particular installation depends largely on the required capacity and range of flow. In considering capacity, it is customary to provide a total pumping capability equal to the maximum expected inflow with at least one of the largest pumping units out of service. A minimum of two pumps should be installed in any installation except where pneumatic ejectors are used to serve fewer than 50 houses. Two pumps are customarily installed where the maximum inflow is less than 1.0 mgd (160 m^3/h). At larger installations, the size and number of units should be such that the range of inflow can be met without starting and stopping pumps too frequently and without requiring excessive wet-well storage capacity. Variable-capacity pumps can be used to match pumping rate with inflow rate.

Where variable-capacity pumps are used, a minimum of two units should be installed. In those cases where more than one variable-capacity unit is required to handle peak flow, three units should be installed. In this manner, it is possible to maintain a reasonable rate of flow through each pump. Operation of a single variable-capacity pump in parallel with a constant-capacity pump requires the variable-speed unit to operate at almost no capacity whenever total inflow barely exceeds the rating of the constant-capacity unit. This is extremely difficult service and should be avoided. As a general rule, pumping rates of less than 20% of the rated capacity for which a pump is designed will result in excessive internal recirculation and unstable operation. Recirculation can occur in some pumps at more than 50% of rated capacity. See Section 2.1.3.

Operating Frequency Pump size should be coordinated with wet-well design in order to avoid frequent on-off cycling of pumps. Excessive starting will cause undue wear on the starting equipment. Also, standard motors should not be started more than six times an hour. Where more frequent starting is required, special motors should be provided. Inflow into the wet well without pumping should not exceed about 30 min if septicity is to be prevented.

Cycle time is defined as the total time between starts of an individual pump. It can be determined by comparing the volume between the on and off levels in the wet well with the pump capacity. Cycle time is computed as follows:

In USCS units $$CT = \frac{V}{D - Q} + \frac{V}{Q}$$

In SI units $$CT = 60\left(\frac{V}{D - Q} + \frac{V}{Q}\right)$$

where CT = cycle time, min

V = wet-well volume between on and off levels, gal (m^3)

D = rated pump capacity, gpm (m^3/h)

Q = wet-well inflow, gpm (m^3/h)

With a given wet-well volume and pumps having a uniform pumping rate, minimum cycle time will occur when the rate of inflow is equal to one-half the discharge rate of the individual pump under consideration. The formula for cycle time simplifies to

in USCS units $$CT = \frac{2V}{Q}$$

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in SI units $$CT = \frac{120V}{Q}$$

An effective wet-well volume of at least 2.5 times the discharge rate of the pump under consideration is required in order not to exceed the six starts per hour recommended above for pumps having a uniform pumping rate.

Reliability With its increased awareness of and concern for environmental matters, the public has little tolerance for the bypassing of sewage equipment because of power outages, equipment failure, insufficient pumping capacity, or any other reason. Reliability is of extreme importance, and the design of pumping facilities should be premised on providing continuous service. Where electric motors are used, two incoming power lines from separate sources with automatic switching from the preferred source to the standby source are the minimum required for reliability. Standby engine-driven pumps, engine-driven right-angle gear drives, or standby engine-driven generators should be provided where dual electric service cannot be obtained or where the degree of reliability provided by two feeds is not considered adequate. Raw sewage pumping installations are particularly critical. Plant pumping installations usually can be out of service for as long as 4 hours without adversely affecting the treatment process, provided the liquid will flow by gravity through the plant.

Speed The maximum speed at which a pump should operate is determined by the net positive suction head available at the pump, the quantity of liquid being pumped, and the total head. When specifying pumps, especially those that are to operate with a suction lift, the speed at which the pumps will operate should be checked against limiting suction requirements as set forth by the Hydraulic Institute.

In general, it is not good practice to operate sewage pumping units at speeds in excess of 1750 rpm. This speed is applicable only to smaller units. Larger pumps should operate at lower speeds.

Preparation of Head-Capacity Curves Pump selection generally involves preparation of a system head-capacity curve showing all conditions of head and capacity under which the pumps will be required to operate. Frictional losses can be expected to increase with time, materially affecting the capacity of the pumping units and their operation. For this reason, system curves should reflect the extreme maximum and minimum frictional losses to be expected during the lifetime of the pumping units as well as high and low wet-well levels.

Where two or more pumps discharge into a common header, it is usually advantageous to omit the head losses in individual suction and discharge lines from the system head-capacity curves. This is advisable because the pumping capacity of each unit will vary depending upon which units are in operation. In order to obtain a true picture of the output from a multiple-pump installation, it is better to deduct the individual suction and discharge losses from the pump characteristic curve. This provides a modified curve that represents pump performance at the point of connection to the discharge header. Multiple-pump performance can be determined by adding the capacity for points of equal head from the modified curve. Figure 9 shows a typical set of system curves, together with representative individual pump characteristic curves, modified pump curves, and combined modified curves for multiple-pump operation. Intersection of the modified individual and combined pump curves with the system curves shows total discharge capacity for each of the several possible pumping combinations. A typical set of system curves consists of two curves with a Williams-Hazen coefficient of $C = 100$ (one for maximum and one for minimum static head) and two curves with a Williams-Hazen coefficient of $C = 140$ (for maximum and minimum static head). These coefficients represent the extremes normally found in sewage applications.

Pumps should be selected so the total required capacity of the installation can be delivered with maximum water level in the wet well and maximum friction in the discharge line. Pump efficiency should be maximum at average operating conditions. In the case of Figure 9, assuming that the total capacity of the installation is to be obtained by operating pumps 1, 2, and 3 in parallel, the total head required at the discharge header would be approximately 51 ft (15.5 m). Projecting this point horizontally to the individual modified

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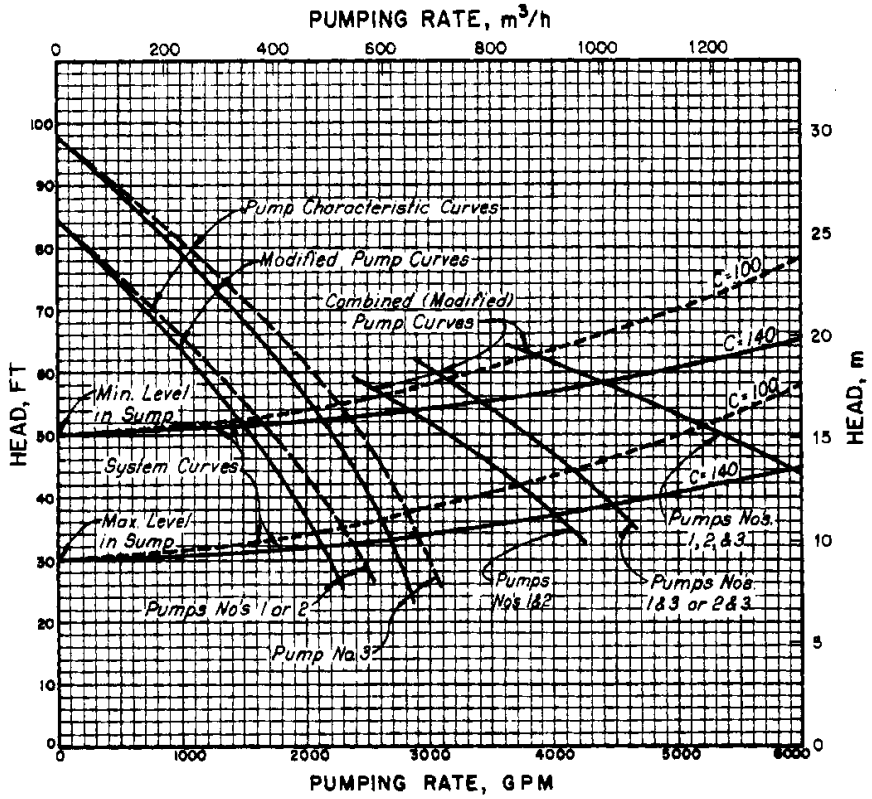


FIGURE 9 Typical head-capacity curves

pump curves and thence vertically to the pump characteristic curves, the required head for pumps 1 and 2 should be 54 ft (16.5 m) and for pump 3 approximately 57 ft (17.4 m). The difference between the head obtained from the pump characteristic curve and the modified curve is the head loss in the suction and discharge piping for the individual pumping units.

Figure 9 also shows the minimum head at which each pump has to operate, approximately 39 ft (11.9 m) for pumps 1 and 2, and about 42 ft (12.8 m) for pump 3. These minimum heads are important and should be made known to the pump manufacturer because they will usually determine the maximum brake power required to drive the pump and the maximum speed at which the pump may operate without cavitation.

PUMP DRIVERS

In the majority of cases, pumps are driven by electric motors. Sometimes, however, they are driven by gasoline, gas, or diesel units where firm power is not available or where pumping is required only at infrequent intervals. Variable-speed drivers are used extensively in sewage applications. These units generally consist of variable-speed motors or constant-speed motors with adjustable slip couplings of either the eddy-current or the fluid coupling type. Selection of the type of variable-speed driver to be used is usually based on space considerations, initial cost, operating cost over the expected life of the equipment, and customer preference. Emphasis is increasingly being placed on operating cost over the expected life due to government and environment requirements. See Section 9.2 for these and other types of speed-varying devices.

Variable-speed drivers are particularly appropriate for raw sewage installations that discharge to a treatment plant. Use of this equipment allows the treatment facilities to operate continuously instead of intermittently surging the plant at incremental pumping rates. Variable-speed drivers are used to pump settled sewage and biological sludge where

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intermittent surging would adversely affect the process. Also, sludge pumps used to feed dewatering equipment are often equipped with variable-speed drives because it is necessary to vary the rate of discharge with the dewatering characteristics of the sludge.

For dry-pit applications, the choice between horizontal- and vertical-drive motors depends considerably upon the station arrangement and available space. Horizontal motors are usually preferred, provided there is space and no potential flooding problem. Horizontal pumps are more easily maintained, and they are generally less expensive in first cost. Vertical drivers are generally used, however, for the pumping of raw sewage because of their smaller space requirements. Also, vertical units are advantageous in that the motor is located higher and is less susceptible to flooding.

In vertical dry-pit installations, intermediate shafting is normally preferred. This allows the drivers to be located above a potential flood level in the station. However, the use of intermediate shafting can become quite expensive in terms of initial installation costs and maintenance costs. Additionally, they may be prone to vibration problems if proper precautions are not taken in the design of the station. Vertical drive motor can also be mounted directly above a pump without the use of intermediate shafting. In doing so, it is connected to the pump by means of a suitable coupling. A separate support or bracing of the motor may also be required to provide for adequate installed stiffness.

PUMP CONTROLS

Some means of controlling pump operation is required at most pumping installations. This is usually done from either wet-well level or flow.

Level Control With level control, each pump is turned on and off at specific water levels in the suction wet well; in the case of variable-capacity pumps, the level control attempts to maintain a preset level, once started. Pumps turn on with a rising level and off as the level lowers. Level control is generally used in raw sewage pumping applications. In this manner, it is possible to match discharge with incoming flow.

Flow Control Flow control is used sometimes where there is no limitation on the availability of flow to the pump suction and where it is desirable to maintain a predetermined rate of discharge. Where flow control is used, a flowmeter is used as the primary instrument to measure flow and to serve as a basis for varying pump speed, which in turn controls capacity. The speed can be changed either manually or automatically through closed-loop instrumentation.

Additional Level Control Low-water pump cutoff and high-level alarm are provided on most pumping installations. The low-level cutoff is required to prevent the pumps from running dry, and the high-level alarm notifies the operator in the event the pump should fail to operate. When the pump stops because of low-water cutoff, there is usually some means of indicating this to the operator.

MISCELLANEOUS DESIGN CONSIDERATIONS

In addition to the matters discussed previously, there are certain other items that should be given consideration in the design of pumping installations.

Piping and Valves Suction and discharge piping should normally be sized so the maximum velocities do not exceed 5 and 8 ft/s (1.5 and 2.4 m/s), respectively. Higher velocities, however, may be justified by economic analysis for particular installations. Lines less than 4 in (102 mm) in diameter should not be used for raw sewage. Preferably, sludge lines should be at least 6 in (152 mm) in diameter; 4-in (102-mm) lines are sometimes used for dilute biological sludge.

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Valves should be installed as required on the suction and discharge sides of each pump to allow removal and maintenance of individual pumping units without disturbing the function of the remainder of the installation. It is customary to use either ball or plug valves on raw sewage and concentrated sludge applications. Either plug or butterfly valves can be used for settled sewage or for dilute sludge.

Piping should be designed with sufficient flexibility to avoid stress on the pump flanges. Flange-coupling adapters are sometimes used for this purpose on both the suction and discharge sides of the pump.

Surge Control Careful attention should be given to surge control wherever a pump discharges into a force main of appreciable length. Generally this is a problem only in the design of raw sewage pumping stations located within the collection system. Changes in fluid motion caused by starting or stopping of pumps or by power failure can create surge conditions.

Surges caused by normal starting and stopping of pumps driven by electric motors may be controlled by (1) selecting individual pump capacities such that the change in velocity in the system when a single pump starts or stops will not result in excessive surges, (2) using variable-speed drives to bring pumps gradually on- or off-line, or (3) using power-operated valves that are controlled so the pumps are started and stopped against a closed valve.

Surges caused by power failure can be controlled by devices designed to open on an increase in pressure, by devices that will exhaust sewage from the system upon sudden pressure drop in anticipation of surge, or by a surge tank.

Pump Seals Most sewage and sludge pumps can be obtained with either mechanical seals or packed stuffing boxes. Conventional mechanical seals have the disadvantage of requiring a pump to be disassembled so the seal can be repaired. Present mechanical seal technology offers a solution to this in the form of a split mechanical seal. Such a seal can be removed and repaired or replaced without the necessity of disassembling the pumping unit.

Often it is easier to replace the seal rather than repair it, and it is desirable to keep a spare on hand for this purpose. Packed stuffing boxes provided with water-sealed lubrication are still the most common choice for non-submersible sludge and sewage pumps. Grease seals are sometimes used for some of the smaller sewage pumps that do not run continuously.

Water serves multiple purposes as a sealing medium: it seals, lubricates, and flushes. Flushing is particularly important where abrasive material is involved in that it helps prevent this material from entering the seal. Grit and ash are very abrasive, and either will cut the shaft sleeves in a relatively short time. Where pumps are controlled automatically, a solenoid valve interlock with the pump starting circuit should be provided in the seal water connection to each pump. A manual shutoff valve and strainer should be provided on each side of each solenoid valve, and a bypass line should be provided around it.

Mechanical seals are normally lubricated with a clean external water source supplied at a pressure and flow rate as recommended by the seal manufacturer. Seals can be lubricated by the product being pumped provided it is filtered and can provide a reliable pressure and flow rate to the seal cavity. In such cases, a connection is normally provided between the pump discharge and the seal with a 0.10 to 0.20 in (2.5 to 5 mm) in-line filter to prevent foreign material from entering the seal cavity.

Occasionally, a pump station is so remote that sealing water is not readily or economically available. Mechanical seals can be provided constructed of special materials to withstand such adverse conditions. Also, several types of "formed-in-place" packing are available that do not require a smooth surface on which to seal as the packing develops its own sealing surface. However, these packing materials are not normally recommended for temperatures over 130°F (54°C) or pressures over 60 lb/in^2 (4 bar).

Pump Bearings Pump bearings must be adequate for the service and should be designed on the basis of not less than a minimum life of 5 years in accordance with the American Bearing Manufacturers Association life and thrust values. The larger sewage pumps are usually equipped with both case and impeller rings of bronze or chrome steel.

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Cleanout Ports Pumps should be provided, where possible, with cleanout ports on both the suction and discharge sides of the impeller. These are desirable for inspection and maintenance purposes.

Wet-Well Design Raw sewage wet wells should not be so large that sewage is retained long enough to go septic. It is usually desirable to limit storage to a maximum of 30 min. Shorter retention time is desirable. With the variable-speed controls now available, many stations can be designed so that the pumping rate matches the inflow rate and the inherent difficulties of frequent pump cycling or long retention times in wet wells can be avoided.

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SECTION 12.3
DRAINAGE AND IRRIGATION

JOHN S. ROBERTSON
JIM BIRD

"Water is a good servant, but it also is a cruel master." (John Bullein, 1562)

INTRODUCTION

The truth of John Bullein's insight is nowhere more evident than in the topics discussed here. Water can be the greatest of blessings or the tool of destruction, based simply on its quantity and location. Water has the power to turn deserts into gardens but it also can lay waste to years of building and development in a matter of hours. Harnessing the power of water is key to creating a durable infrastructure for humans within their environment, and the key invention for achieving this is the pump. It is virtually certain that the first efforts at moving large quantities of water were directed at solving the problems of too much or too little water at the desired location. The history of the earliest civilizations document devices that are the predecessors of today's pumping equipment. The shadoof, noria, and inclined screw were all developed to move water from rivers into fields for farming and into rudimentary public water supply systems. Floods presented another challenge and similar devices were used to drain low lying areas. The same challenges of moving water exist today, drainage systems exist to provide protection from the rages of water, and irrigation systems open up new areas to agriculture to spread the benefits of water over larger areas.

DRAINAGE PUMPS

Drainage pumps are used to control the level of water trapped in a protected area. Entrapment occurs when high lake levels, stream stages, and tides preclude the normal discharge of streams, storm runoff, and seepage from the protected area. These high-water conditions are created by floods and hurricanes or impoundment.

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Floods and hurricanes occur infrequently and are relatively short-lived. However, as normal drainage is not possible at such times, all interior drainage that cannot be ponded must be pumped over the protective works if the protective works is a levee or through it if it is a concrete flood wall. Pumps for this purpose are strategically located at the edge of pending areas and streams, in sewer systems, or in the protective works. These pumps are operated continuously or are cycled on and off as necessary to maintain the water level in the protected area below the elevation at which damage would be experienced. As their use is required only during emergency situations and usually under adverse weather conditions, reliability of operation is essential.

The history of one such installation chronicles the history of drainage systems in the United States. Many early settlements were established along waterways in an effort to provide both a ready source of water and access to transportation. This was true of a small French settlement established in 1718 on the Mississippi River. The initial location was on high ground, 14 ft above river level. Growth of the settlement moved the population away from the river and into a natural basin as much as 6 ft below normal water levels. Heavy rains, high tides, and floods produced rivers in the streets. Trapped water resulted in threats to the health of the residents. In the 1890s the Sewerage and Water Board was established to implement a drainage plan to meet the current and future need for a safe and secure New Orleans. The first installations were driven by windmills and ineffective. In 1906, A. B. Wood, an engineer, employed by the Sewerage and Water Board developed a pump with a 6-ft-diameter (1.8 m) rotor capable of moving larger quantities of water than possible before. He developed larger pumps, 12- and 14-ft (3.7- and 4.3-m) diameters, using the same principle. A photograph of one installation is shown in Figure 1. Wood's pumps were tested many times over the years by floods, rain, and tropical storms and performed admirably in resolving the city's drainage problems. In 1974, the A.B. Wood Screw Pump at the No. 1 Pump Station in New Orleans was designated as a National Historic Mechanical

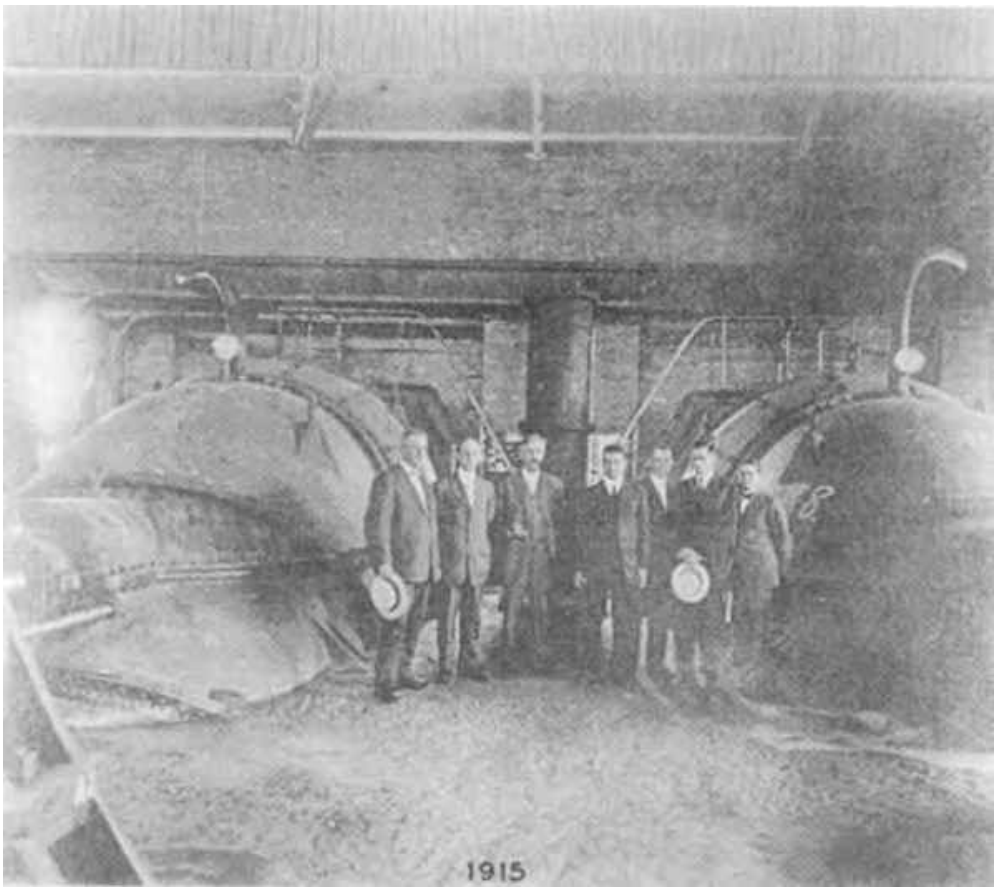


FIGURE 1 Wood's drainage pumps for New Orleans (ASME, 1974)[1]

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FIGURE 2 A recent New Orleans drainage pump (Flowserve Corporation)

Engineering Landmark.[1] Other pump stations were built over the years with installations and designs very similar to the Wood's pumps. One of these is shown in Figure 2.

In 2005, the New Orleans drainage system was put to its greatest test by Hurricane Katrina. The pumping stations feeding the drainage systems have been designed to lift water out of the basin and into the drainage canals moving large quantities of water at low pressures. The drainage canals, secured by levees, stretch into the inner city; and the failure of the canal system placed the pump stations under a load of fast, massive flooding that they were not designed to resist. New proposals for innovative designs capable of pumping high flows against heads necessary to utilize closed piping systems and avoiding the open canal system are being considered to replace the Wood's pumps and their offspring.

IRRIGATION PUMPS

From the annual floods on the Nile River that the Egyptians attributed to their gods the benefits of water on agriculture have been well known. The problem faced was how to maximize the benefits. Annual floods deposited rich silt for top soil but also took so long to dry that the growing season did not allow for multiple plantings or crops taking longer times form planting to harvest. Irrigation systems with pumps as the prime deliverer of water play an important part in making vast areas of arid and semiarid land agriculturally productive. As another benefit, proper irrigation management can make fertile areas capable of producing multiple crops throughout the year. Two types of irrigation systems exist differentiated by the source of the water that is to be brought to the crops.

In areas where surface water is plentiful and available from rivers, streams, lakes, and ponds, it can be pumped directly into the distribution system or into a conveyance system. Many systems use large pump stations to lift water from these sources into distribution canals that move water by gravity throughout the planting areas. At specific locations along the canals booster pumps remove water from the canals to delivery systems such as sprinkler systems. Most of the pumps used in these installations are mounted permanently and are arranged to operate as necessary without constant attendance during the growing season.

Ground water, water from subsurface sources, is lifted up to the surface by well pumps. The pumps normally provide energy to lift the water out of the ground and also to deliver it to the irrigation system. In such installations, a well is drilled in the vicinity of the area

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FIGURE 3 Center pivot irrigation pumping system

to be irrigated and is fitted with the proper size pump. Center pivot systems as shown in Figure 3 each have a pump at the center. The traveling sprinkler system is driven though the circular path by hydraulic motors. In most instances, more than one well is needed to provide the required capacity. Well pumps require care in sizing to be certain that as water is removed from the local aquifer the draw down of the water table does not fall below the pump intake. For this reason wells and well pump settings are becoming much deeper as the rate of withdrawal increases. Pumps with settings of 1000 ft (300 m) below the surface are not unusual.

PUMP TYPES

Centrifugal pumps are used almost exclusively in drainage and irrigation installations. Because of the large selection of propeller, volute, turbine, and portable pumps manufactured today, there is little difficulty in finding a pump that will meet the conditions encountered in these fields. Pumps in the sizes needed to meet the requirements of the majority of installations are available as standard items. Specialized pumps and extremely large pumps are designed and built to meet the needs of individual projects.

Propeller Pumps These units are used for low-head pumping. As most of the pumping in drainage and irrigation is low-head, the propeller pump is the most widely used type. In general, vertical, single-stage axial- and mixed-flow pumps as shown in Figure 4 are used; however, there are instances where two-stage axial-flow pumps should be considered for economic reasons. In systems that are designed for high flow rates the casing of the pump may be formed in concrete with a pull out pumping element. A typical design for one such station and a detail of the pumping element are shown in Figure 5.

Horizontal axial-flow pumps are used for pumping large volumes against low heads and usually employ siphonic action when not of the submersible type. When higher heads are involved, these pumps can be arranged to operate with siphonic action until the back pressure places the hydraulic gradient above the pump.

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FIGURE 4 Vertical, single-stage pumps of the propeller type (left) and the mixed-flow type (right) (Flowserve Corporation)

Volute Pumps Volute pumps are used when pumping from surface sources and in general when the total head exceeds approximately 45 ft (14 m). Such pumps are available in many types, such as vertical and horizontal shaft, end suction, bottom suction, and double suction with semi-open or closed impellers, etc., as illustrated in Figure 6. These pumps are mounted in dry pits when located below grade and on slabs or floors when located above grade. The particular type used depends on the capacity and kind of service to be performed.

Deep-Well Turbine Pumps Deep-well turbine pumps (vertical-shaft, single-suction pumps having one or more stages, as shown in Figure 7) are used in irrigation primarily to pump water from a subsurface source into a distribution system. The head against which the pump will operate determines the number of stages that must be provided. For large flow rates, more than one pump will be needed.

Submersible Turbine Pumps In these deep-well units, the motor is close-coupled to the pump and submerged in the well as illustrated in Figure 8. This type of pump is used for high-head applications where long intermediate shafts are undesirable.

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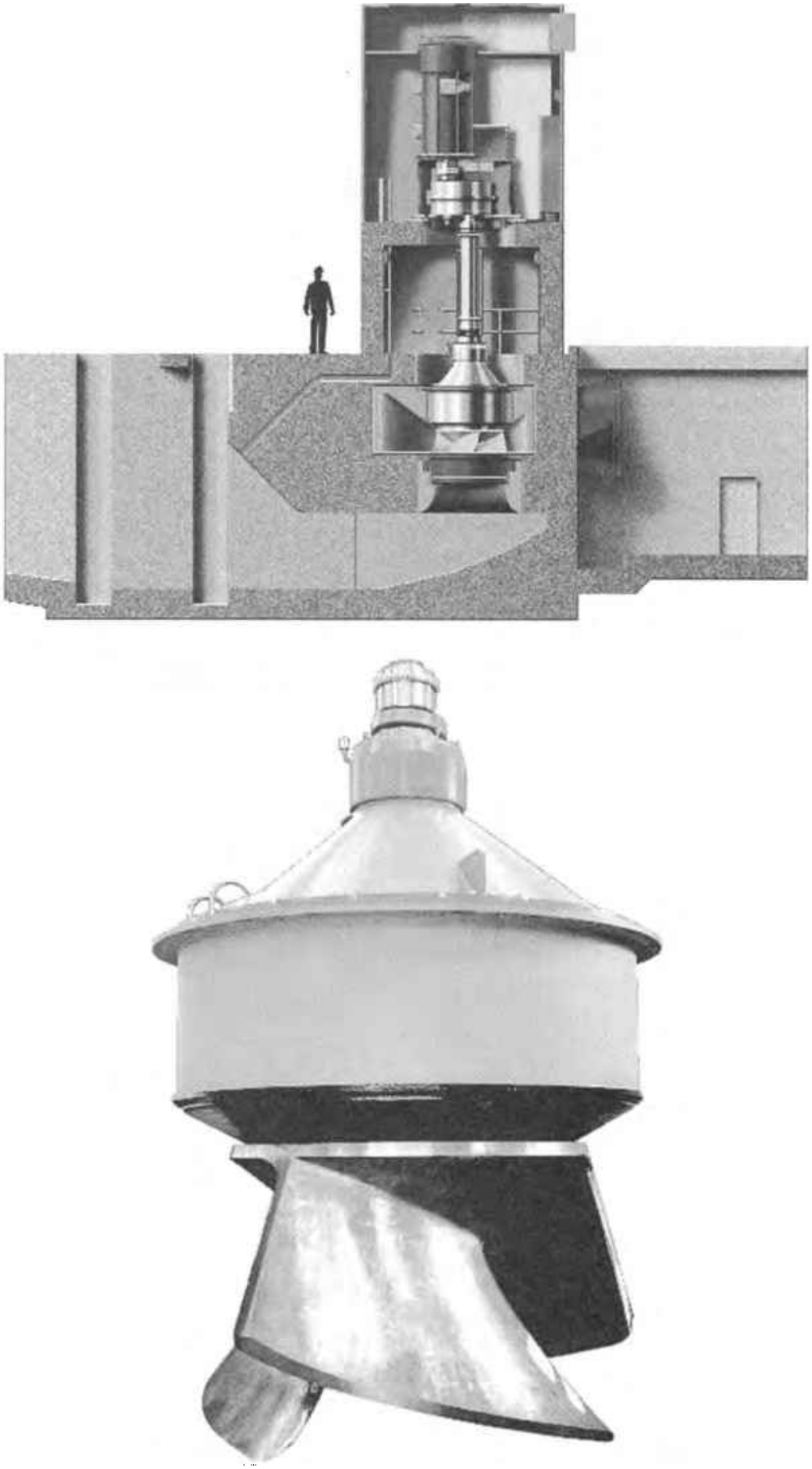
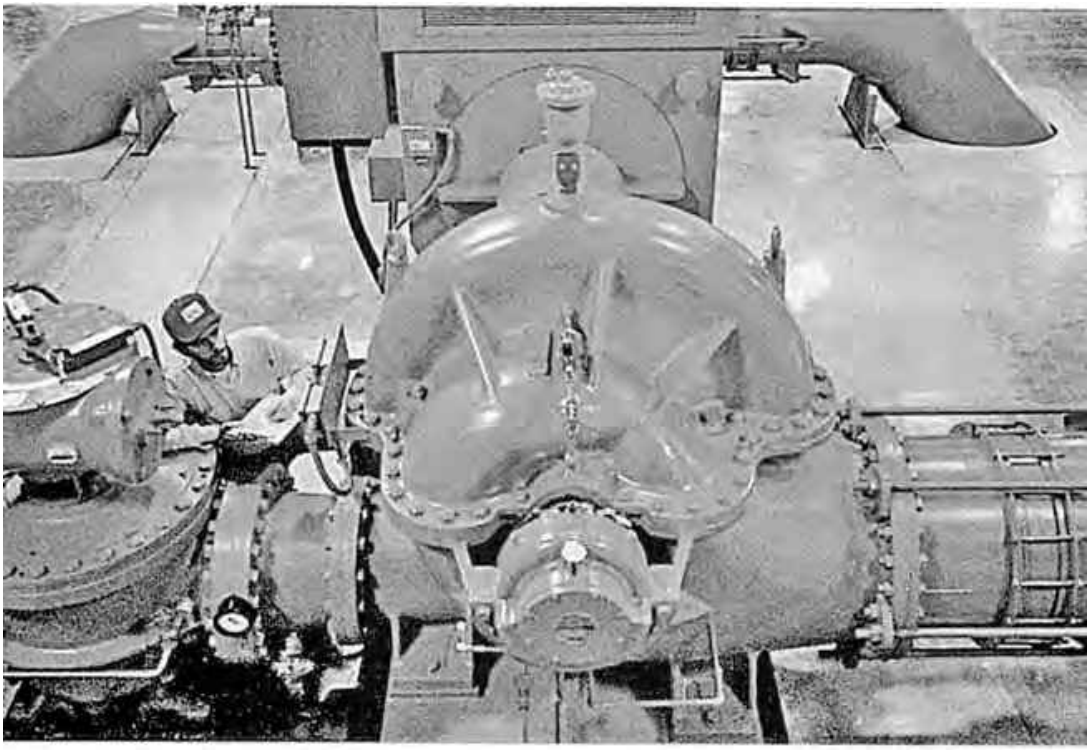


FIGURE 5 High-flow-rate pull-out pump and concrete casing (Flowserve Corporation)

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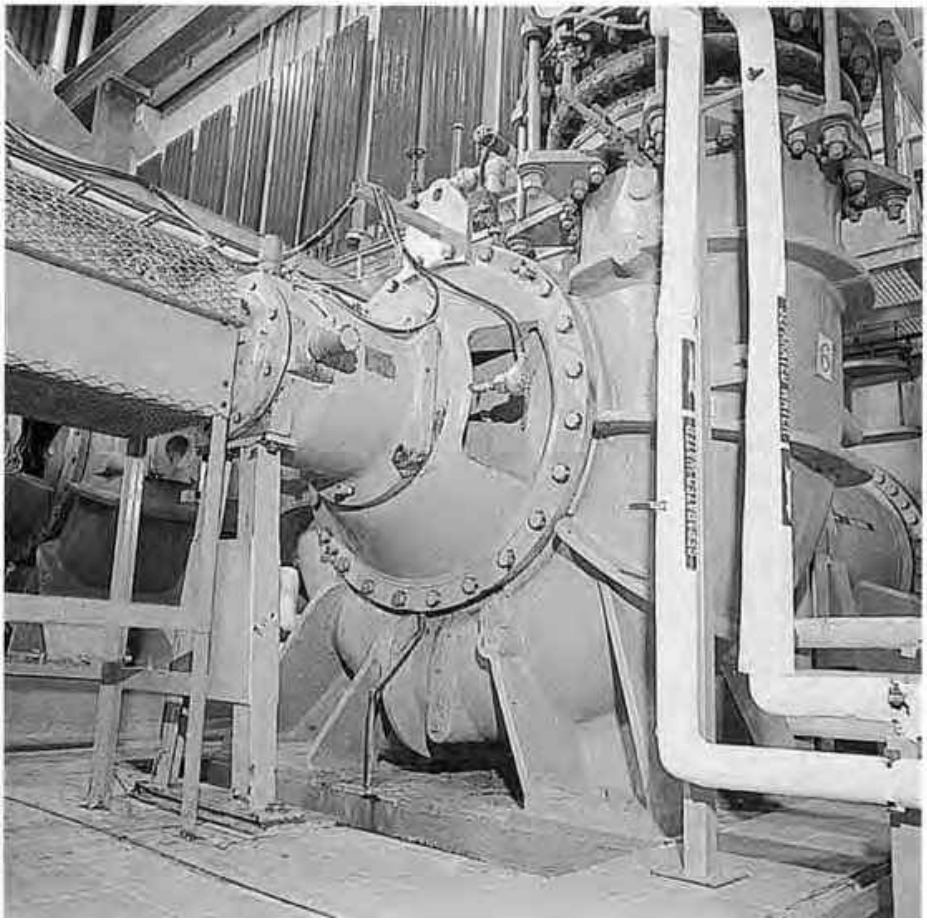


FIGURE 6 Horizontal-shaft volute pumps: double suction (above) and end suction (below) (Flowserve Corporation)



FIGURE 7 Multistage, deep-well, vertical turbine pump (Flowserve Corporation)



FIGURE 8 Submersible vertical turbine pump (Flowserve Corporation)

Portable Pumps In drainage work, portable pumps are used as emergency equipment to control ponding elevations where mobile equipment, such as tractors and trucks, having power takeoffs are available to drive them. In irrigation work, they are usually used to irrigate from a surface source. Such pumps are small-capacity, low-head, economical pumps that must be submerged in order to be used.

PUMP SELECTION

A pump selection study should always be made, and its importance cannot be emphasized enough. Such studies permit selection of the type of pump and discharge system best suited to the project and provide the information needed to proceed with the design of the installation.

Before initiating such a study, all previous studies made to determine the total pumping requirement or station capacity, pertinent water-surface elevations, terrain, utility locations, proposed station or well locations, points of discharge, and the proposed method of operation should be reviewed.

Number of Pumps First costs are generally of more concern than operating costs in drainage and irrigation work because the operating period for the majority of installations is relatively short and occurs only once a year. Costs can be minimized by using as few pumps as possible. However, one-pump installations are seldom used except in the case of wells. For reliability, a minimum of two pumps should be installed in drainage pumping stations, where the loss of even one pump during an emergency situation could result in considerable damage. Three or more pumps are preferred. Standby units are provided only in those installations where continuous operation precludes taking a pump out of service for maintenance.

The number of pumps ultimately used should be consistent with the demands of the project. For instance, when the installation is located in an agricultural area or is a part of an urban sewer system, a standby pump should be provided because these installations must always be capable of discharging project requirements during periods of blocked drainage. In this way, considerable damage may be avoided. When the installation is used to pump storm water from pondage or irrigation water from a lake, the loss of a pump is not critical and so a standby pump is not needed.

For installations requiring the use of large pumps, foundation conditions become important. To prevent the installation from being relocated to a less desirable site or the necessity of providing a more expensive pile foundation because the bearing pressures at the selected site exceed the allowable limit, the number of pumps should be increased, provided the loading can be reduced to an acceptable amount and the resulting installation continues to be the most economical.

Inadequate depth on the suction side of the pumps may necessitate the use of more pumps. Should the water not be deep enough to provide the submergence needed by the contemplated pump, more pumps of a smaller size may have to be used or the sump and approach channel may have to be excavated to the needed depth. The latter alternative could cause operational and maintenance problems and might be the more expensive solution.

Capacity The capacity of a pump (i.e., the flow rate which it is designed to handle) is a function of the total pumping requirement, the number of pumps, and, in the case of wells, the capacity of the well. Whenever possible, all pumps in a multiple-pump installation should be of the same capacity. This is advantageous from a cost standpoint as well as from a maintenance standpoint. In drainage installations, three pumps are generally provided in order to have the capability of pumping not less than two-thirds of the project requirement with one pump inoperative. Each pump would then have a capacity equal to one-third of the total capacity. When more than three are used, the capacity should be the total required capacity divided by the number of pumps being used. If two pumps are used, each pump should be sized to pump not less than two-thirds of the total capacity.

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In irrigation installations utilizing multiple pumps, the capacity of each pump should be the same. In single-pump installations, the pump should be sized to meet project requirements. When wells are involved, the capacity of the pump will be determined by the capacity of the well. Often several wells are needed to give the capacity to satisfy the established requirements.

When a standby pump is to be provided, its capacity should be equal to that of the largest pump being furnished.

Head Total head is the algebraic difference between the total discharge head and the total suction head. In drainage and irrigation work total suction head can usually be determined, but this is not always the case for the total discharge head. The losses in the discharge systems often must be determined by hydraulic model test rather than by calculation and therefore must be estimated for preliminary selection purposes. The head specified will have to be some head other than total head; generally pool-to-pool head is used.

Total discharge head is defined in the Hydraulic Institute Standards, and its value is to a great extent determined by the type of discharge system used. A number of the many possible discharge systems used are shown in Figure 9. The losses in systems *A, B, C, D,* and *E* can be calculated. Thus the performance for pumps discharging into these systems can be put on a total head basis.

System A is an "over the levee" siphonic discharge line, the use of which can seldom be justified. However, continuous operation over long periods could effect a savings that would be sufficient to justify the additional cost of the installation and the taking on of the operational hazards usually encountered with such lines. The total discharge head for this system is equal to the height of the discharge pool, stream, or lake above the impeller plus the exit loss and the discharge line losses from the pump discharge nozzle to the line terminus. The absolute pressure at the high point of the line should be not less than 9 ft (2.7 m). Lower values have been used successfully, but this should be the exception rather than the rule. For additional information on the determination of heads in a siphonic system, refer to "Siphon Head" in Section 11.1.

Check valves should be installed on the discharge end of all lines subjected to a cycling operation. The closing of these valves following pump shutdown prevents reverse flow into the protected area and permits development of the pressures needed to keep the lines full (primed) during those short periods when pumps are idle. These pressures will be less than atmospheric pressure, with the minimum pressure (absolute) being at the high point. Should there be significant leakage at the joints or in the valves, the pressures will rise and the water level at the high point will drop. If the water level drops below the invert of the line at the high point, the two legs of the line will be separated by an air space and priming will be necessary when the pump is started.

Air-release valves should be installed at the high point of all siphonic discharge lines to provide an escape for the air being compressed as the water rises in the line during priming, and vacuum-break valves should be used to prevent reverse flow into the protected area. For lines equipped with flap valves, the vacuum-break valve should be manually operated and used when the flap gate fails to seat properly or to provide a rapid means of draining the line when pumping is no longer required. Units for almost any size line can be obtained from manufacturers who specialize in such equipment or they can be assembled, using swing check and angle valves.

System B in Figure 9 is an "over the levee" nonsiphonic discharge line. To preclude siphonic action, this line should be vented at the high point with a vent having a diameter that is approximately one-fourth that of the line. The invert of this line at the high point should be placed at the same elevation as the top of the protective works so the pumped flow can discharge from the down leg of the line under gravity without backwater effects for all discharge pool elevations up to maximum. Thus the total discharge head will have a constant value because it will not be affected by changes in the level of the discharge pool. To ensure adequate prime mover capacity when using this system, it is the practice in all cases to use the top of the line at the high point in lieu of the hydraulic gradient when determining the total discharge head. Therefore, total discharge head is the height of the top of the line at the high point above the impeller plus the velocity head in the line at the high point based on a full pipe and the losses in the line from the pump discharge nozzle to the beginning of the down leg of the line.

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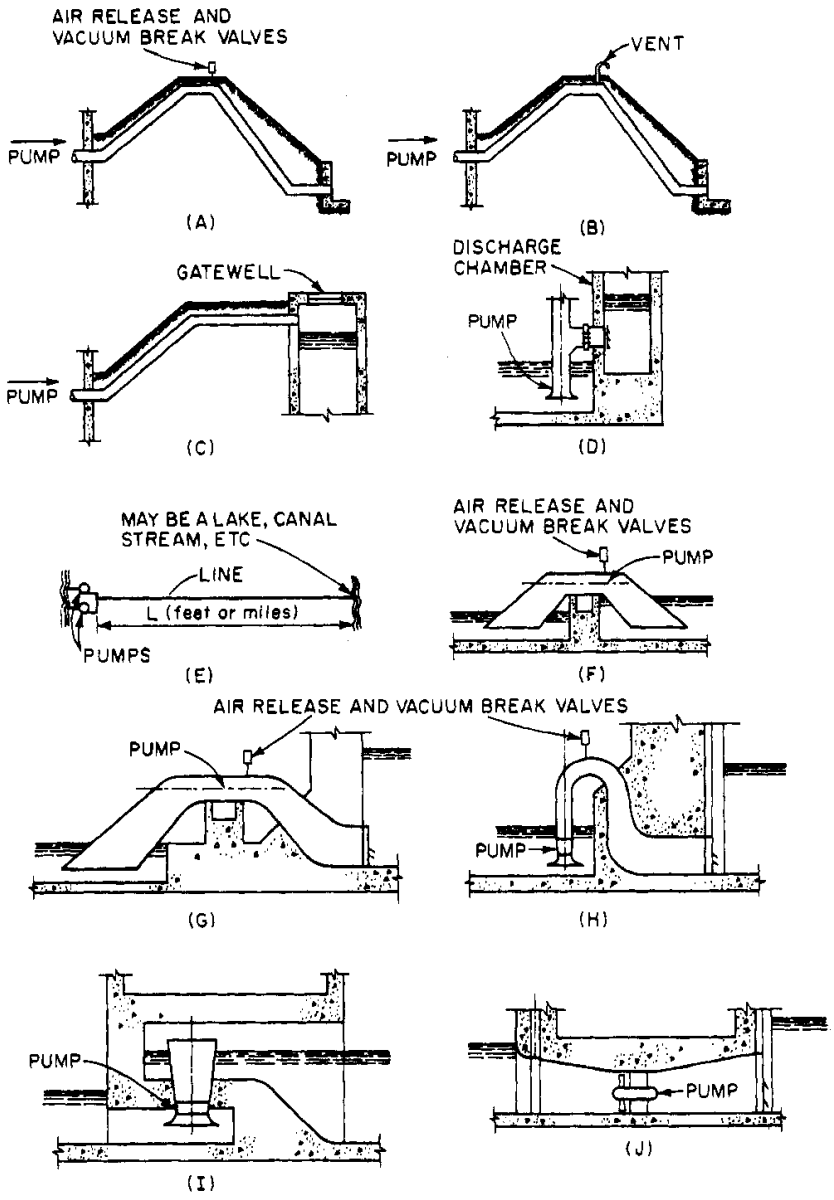


FIGURE 9A through J Discharge systems (adapted from Reference 2)

System C is used when there is a conduit carrying the normal gravity discharge under the levee adjacent to the station that must be valved off against reverse flow into the protected area during periods of high water. The closure gate is located in a gate well constructed on the stream or lake side of the levee to prevent subjecting the gravity conduit to high-water conditions. The pump discharge lines go over the levee and terminate in the gate well above the maximum water level. This shortens the lines and reduces the cost. The total discharge head for this system is equal to the height of the top of the line at the terminal end above the impeller plus the exit loss and the losses in the line between the pump discharge nozzle and the terminal point. The total discharge head for this system, as in system *B*, is independent of the discharge pool and therefore constant. Neither flap valves nor vents are required on these lines.

System D is used when the pumping station is constructed as an integral part of the levee or flood wall. The invert of the pump discharge line is placed at an elevation that is above the stream or lake level that will prevail approximately 70% of the time or as is dictated by the physical dimensions of the pump. Owing to the extreme turbulence in the discharge chamber, gates with multiple shutters, which are less likely to be damaged, should be used instead of flap gates on discharge lines that are larger than 36 in (914 mm) in diameter.

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When the water level in the discharge chamber is below the top of the discharge line, the total discharge head is determined in the same manner as for system *C*. For higher discharge water levels, the total discharge head is equal to the height of the water level in the discharge chamber above the impeller plus the exit loss and the losses between the pump discharge nozzle and the chamber side of the flap valve.

System E is perhaps the most common discharge system in use today. It is used to connect one pump or several manifolded pumps with a lake, canal, stream, ditch, reservoir, or sprinkler system. For short lines and low static heads, valve and fitting losses, frictional losses, and exit losses are very important, whereas in long lines or very high static head installations only frictional losses are given consideration. In manifolded installations using propeller pumps, a check valve and gate valve are installed immediately downstream of the pump. The gate valve should always be opened before the pump is started because the motors provided are not usually sized to operate against shutoff head. Positive shutoff valves are placed immediately downstream of volute or turbine pumps because these pumps are usually started and stopped against a closed valve. They also prevent reverse flow into the sump when one of the pumps is inoperative.

Pool-to-pool head is the difference in elevation between the sump and discharged-water surfaces and is used instead of total head in drainage work because the losses in the discharge system are not easily determined. Installations of this type are exemplified by systems *F*, *G*, *H*, *I*, and *J* in Figure 9. For such installations, it is best to specify the pumps on a pool-to-pool basis, to have the pump manufacturer design the pump and the discharge system, and to verify the predicted performance by model test. It should be noted that in such installations the discharge systems are usually constructed within the confines of the pumping station structure.

System F is operated as a siphon with the pump supplying energy equivalent to the pool-to-pool head plus the system losses. The invert of the pump discharge pipe at the highest point is located above the maximum river stage, and vacuum pumps are generally used to aid in priming the pump. This system is used for pool-to-pool heads of up to approximately 6 ft (1.8 m) and where the physical dimensions of a vertical pump would make it necessary to operate against higher heads. In estimating the losses, entrance losses, which are small [approximately 0.14 ft (4 cm)], should be neglected. The centerline of the suction piping should be assumed to make an angle of 45° with the horizontal, and the diameter of the discharge piping as measured at the discharge flange of the discharge elbow should be such that the velocity at maximum discharge is approximately 12 ft/s (0.037 m/s) or less.

System G is used for pool-to-pool heads of up to approximately 15 ft (4.6 m). The water passages change in cross-section from round at the pump bowl to rectangular at each end. The width at the suction end is the same as or less than that of the suction bay, and the height is such that the entrance to the suction passage is always submerged when the pump is in operation. The discharge velocity should be kept to approximately 6 ft/s (1.8 m/s). Multiple-shutter gates (Figure 2) arranged to be raised when the pump is in operation are provided to prevent prime mover overload when the pump is started and to prevent reverse flow when the pump is stopped or inoperative. These pumps are usually large and slow and require substantial prime movers. Vacuum priming equipment is used so additional power will not be required for priming.

System H is essentially the same as system *G*, except that the former is used for pool-to-pool heads up to approximately 26 ft (7.9 m). Also, a splitter may be required in the discharge water passage for structural purposes as well as for keeping the multiple-shutter gates to a reasonable size. The pumps used with this system are vertical and in general smaller than those used with system *C*.

System I can be constructed with the lip of the pump column either above or below the design flood elevation or the maximum surge, if it is being provided for hurricane protection. If the lip of the pump column is above and the chances of reverse flow through the pumps are extremely remote, gating of either the pump column or the discharge water passage is unnecessary. If the lip of the pump column is below, a decision of whether to gate both the pump column opening and the discharge water passage or just the water passage must be made. In general, if the pumps will be in operation continuously during high-water conditions and if reverse flow through an inoperative pump will not have substantial detrimental effects, only the multiple-shutter gate at the end of the discharge water passage

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need be provided. Reverse flow could occur should the flap for some reason fail to close. Like system *C*, splitter walls may be required if pumps are large. Also, the clearance between the lip of the pump column and the ceiling of the discharge water passage is critical and should be determined by model test.

The shape of the pump column lip will also affect pump efficiency and should be determined by test.

System J could have many configurations but, regardless of arrangement, would have to be gated on the suction side with a positive shutoff gate, such as a pressure-seating slide gate, and on the discharge side with a positive shutoff gate and a multiple-shutter gate or a flap (check) gate if the installation is small and a pipe is used in place of formed water passages. When formed water passages are used, transition sections between the pump and the gates sections will be needed and should be designed by the pump manufacturer.

Total Suction Head The practice for propeller pump installations is to dimension the station sump or sump bays in accordance with Hydraulic Institute standards and to make the distance between the sump floor and the lip of the suction bowl conform to the standards or the recommendations of the pump manufacturer. The approach and entrance velocities resulting therefrom are small enough to be disregarded when calculating total suction head. Total suction head in vertical propeller pump installations is the height from the centerline or eye of the propeller to the water surface in the sump. For drainage installations using vertical propeller pumps, a total suction head of zero is not uncommon. In submerged horizontal propeller pump installations, total suction head is the height from the centerline of the propeller shaft to the water surface in the sump, and the minimum value should be not less than $1.2D$, where D represents the diameter of the propeller.

Volute pumps equipped with a formed suction or suction piping may operate with either a suction lift or a suction head, depending on the suction water level. In either case, all the losses between the entrance and the eye of the impeller should be included in any calculation. Approach velocity is not a consideration in these installations, but entrance losses are.

TOTAL SUCTION LIFT Approach and entrance velocities can be disregarded in suction lift installations by proper dimensioning of the station sump or sump bays and by proper setting of the suction bell. The sump or sump bay dimensions used, as in suction head installations, should conform to the Hydraulic Institute standards or to the recommendations of the pump manufacturer. The suction bell for both horizontal propeller pump and vertical volute pump installations should be set with the lip located approximately $0.5D$ above the sump floor. For horizontal propeller pump installations, the minimum submergence of the suction bell should be approximately $0.25D$, where D is the diameter of the suction pipe; for volute pump installations it should be $1.5D$, where D is the diameter of the suction bell. In determining the total suction lift for these installations, it is assumed that the suction piping is a part of the pump and that the approach and entrance velocities can be disregarded. Total suction lift therefore is the height from the water surface in the sump to the centerline of the propeller shaft or to the eye of the impeller.

Setting The setting of the pump or the locating of the centerline or eye of the propeller or impeller with respect to the water surface should be given careful consideration when selecting the pump to be used. Some installations offer few if any problems in this regard, whereas in others setting may have a considerable effect on the size and number of pumps selected.

TURBINE PUMPS Drawdown is a consideration in any well installation. In setting the pump to prevent cavitation, sufficient NPSH at the eye of the first-stage impeller and/or sufficient depth over the suction bell lip or tailpipe to prevent vortexing should be maintained when maximum draw-down is being experienced.

VOLUTE PUMPS Volute pumps may operate with either a suction head or a suction lift. If with a suction lift and of the horizontal type, the pump should be set above the maximum anticipated elevation of the suction water source in order to avoid inundation. As priming

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will be necessary when starting, a suction lift would be considered a satisfactory arrangement when long periods of continuous operation are anticipated.

Suction head installation is the preferred practice (because priming is unnecessary) and should be used whenever conditions permit. In drainage work, vertical volute pumps are used for pumping small amounts of rainfall runoff and seepage flows. These pumps are usually located in a dry sump adjacent to the storm water pump sump, with motors and valve operators located on the operating floor above. The submergence of these pumps should be such that when discharging at maximum capacity, the total suction head is zero or above. In many instances, water surface fluctuations of just a few feet occur. When small volumes are involved, a cycling operation occurs. For this type of operation, the pumps are usually started at the maximum water level and stopped at the minimum level.

PROPELLER PUMPS Sufficient water depth does not always exist or cannot always be provided to give the submergence needed to permit the smallest and perhaps the most efficient pump to be used. Excavation is one answer. However, depending on the soil type, the location of the installation, the kind of construction used, the silt-carrying characteristics of the stream, and the frequency of operation, excavation may be an operational and maintenance headache and is impractical when sumps are to be made self-draining. Another alternative—and the one most frequently used in drainage work—is to set the centerline or eye of the propeller at or slightly below the minimum sump elevation and select a pump that will operate at this setting with little or no cavitation damage. This means that a larger pump operating at a lower speed should be used.

PUMPING STATION

Sump The sump is perhaps the most important element in the structure of the pumping station. Unless it is properly located, designed, and sized, the flow conditions within could have an adverse effect on the operation of the pump. There are many variations in sump arrangements that are acceptable; however, best results are obtained when the sumps or sump bays are oriented parallel to the line of flow. Flows approaching from an angle create dead spots and high local velocities, which result in the formation of vortices, nonuniform entrance velocities, and an increase in entrance losses. The flow to any pump should not be required to pass another pump before reaching the pump it is meant for. When sumps or sump bays are normal to the direction of flow, such as in sewer systems, the distance between the sump or sump bay entrance and the pump must be sufficient for the flow to straighten itself out before reaching the pump. For additional information relative to sump design and sizing, refer to Chap. 13. If the installation is large enough to warrant it, modeling of the sump to permit the best design to be determined is advocated.

Sumps in drainage installations pumping storm water should be either located above normal water levels, in which case they would be self-draining, or isolated from normal flows by gates. In small installations, motorized pressure-seating gates should be used; in large installations, roller gates raised and lowered with a crane or by some other suitable system should be used. These gates should be sized so the velocity through them will not exceed 5 ft/s (1.5 m/s) for any condition of flow. One gate should be located directly opposite each pump when all pumps are installed in a common sump or should be located at the entrance to each sump bay when pumps are separated.

Frequent cycling of pumps is encountered in installations that pump from sewer systems and that pump seepage. In such installations, the sump should be sized so the volume stored in it and in the ponding area or the sewer lines, as the case may be, within the limits of the operating range, will be sufficient to prevent the starting of the pumps more often than once every four minutes.

Superstructure A superstructure is provided on practically all drainage pumping stations but not on all irrigation pumping stations. The type of superstructure provided should be consistent with the surrounding area and should have a minimum number of windows and openings. In rural areas, corrugated sheet metal structures, which are

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inexpensive, have been used extensively. These provide adequate protection from the elements and from vandalism. To eliminate the need for an indoor crane, hatches in the roof over each pump can be provided to permit a truck crane to remove motors and pumps as a unit. For engine-driven pumps and larger pump installations, indoor cranes of the appropriate size and type should be provided for installation, removal, and maintenance.

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SECTION 12.4
FIRE PUMPS

MARIO DI MASI
RUDOLPH KAWOHL

Compelling reasons dictate the installation of fire protection systems driven by stationary fire pumps. Foremost among these reasons is protection—protection of lives, of equipment, possessions, and inventories, and of major assets, such as hospitals, hotels, office and residential buildings, and warehouses. There are two additional, less obvious but equally compelling, reasons: the reduction of costs and the protection of income-generating operations.

Costs are reduced in a simple manner. Over the projected life of a facility, the total of construction costs plus fire protection equipment costs plus fire insurance costs is lower than the total of construction costs plus fire insurance costs without fire protection equipment.

FIRE PROTECTION SYSTEMS

Protection may be provided by a combination of several complementary means, acting at various levels:

- First by a simple fire detection system activating an alarm
- On the second level, the detection and warning system is combined with a first degree of fire fighting; for example, with a *sprinkler installation*, the role of which is to extinguish the fire at the beginning or, at least to limit its extension so as to permit intervention with additional equipment
- The third level is a combination of detection, warning, sprinkler, and extinguishing systems (water, inert gases, foam)

The degree of protection is obviously a function of risk. Whereas a fire in a single-family-dwelling may be warned by phone and then efficiently contained by means of fire hydrants or fire fighting trucks, the risk of fire in a chemical plant requires a completely

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different approach. The rules for adequate protection vary from one country to another. There are, for instance, well known pamphlets published by the U.S. National Fire Protection Association (NFPA). In Europe, various national rules exist, based on the recommendations of the European Committee of Insurances, such as those of APSAD in France (Assemblée Plénière des Sociétés d'Assurances Dommages), V d S in Germany (Verband der Sachversicherer e;v), and LPC in the United Kingdom (Loss Prevention Council).

FIRE PUMP CRITERIA

The most common liquid available for fire protection and fire fighting continues to be water. The efficiency of a fire fighting installation depends to a great extent on a dependable water supply, which will supply the required flow at the required pressure and be continuously available during the time necessary to extinguish the particular type of fire in question.

The needs for fire fighting liquid vary considerably with the degree of danger involved. When these needs cannot be satisfied from local water sources, such as public water networks, high level reservoirs, or pressurized water tanks, pumping stations need to be designed and situated (located) to address the types of fire possible and the national codes and standards that exist in a particular country or geographical area.

A typical fire fighting station may be composed of the following:

- A jockey pump (Figure 1) that maintains the desired system pressure within the fire fighting system. Typical performance for a jockey pump is 10 gpm (38 l/min) at heads up to 300 ft (92 m).
- A first intervention fire fighting pump (Figure 2) capable of feeding a limited number (up to 5) of sprinklers. This pump is typically driven by an electrical motor and is fed from a reservoir capable of supplying at least one hour of liquid flow without interruption at required flow and pressure. Records indicate that as many as 95% of all fires are extinguished by this first intervention pump. Typical performance for a first intervention pump is 250 gpm (950 l/min) at a head of 200 ft (60 m).
- An emergency fire fighting pump, capable of feeding all available sprinklers, and other supplementary fire fighting equipment, such as deluge installations and fire-hose nozzles.

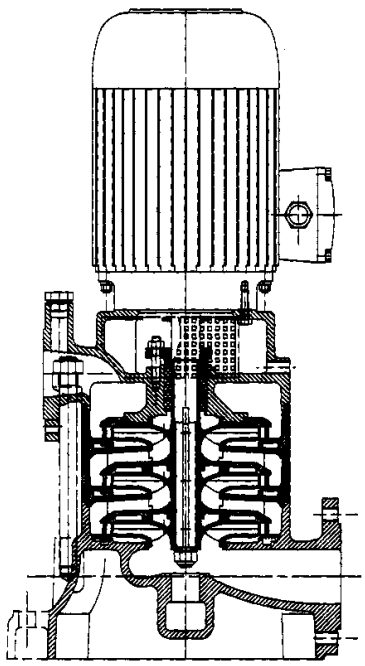


FIGURE 1 Small, vertical multistage jockey pump using an extended motor shaft (Flowserve Corporation)

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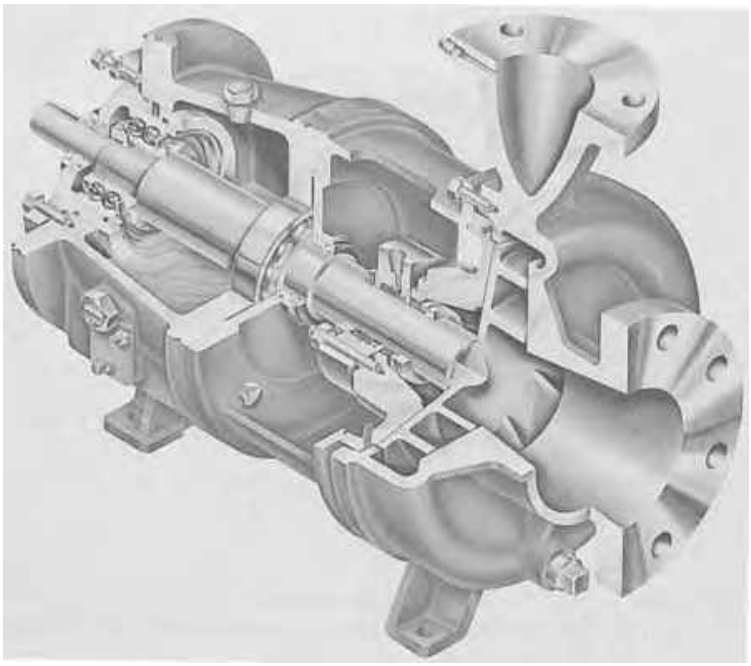


FIGURE 2 End suction, top discharge pump (Flowserve Corporation)

This pump is typically driven by a diesel engine and supplied with fire fighting water from a (practically) inexhaustible source. The emergency fire fighting pump standard in North America is a split case pump (Figure 3), whereas the European standard is an end suction pump (Figure 2). In each case, typical pump performance is from 500 to 5000 gpm (100 to 1200 m^3/h) at a head of 260 ft (80 m).

- High-rise buildings, protected by sprinklers or standpipes and hose systems, require an overpressure pump. This pump is generally a close coupled end suction pump (Figure 4) driven by an electrical motor, with typical performance of 150 gpm (34 m^3/h) and a head (according to the building height) of at least 200 ft (60 m), sourced from a public water network.

Although the various national authorities have different approaches to the details of fire fighting equipment and arrangement, they nevertheless have a similar basic philosophy. The pumps utilized in fire protection systems are usually classical designs. Specifications require skillful packaging of the complete fire protection system, including the pump(s), driver(s), starting and control devices, and other accessories.

Most European codes give considerable freedom of choice relative to pump type and design, whereas NFPA 20 (United States) is more restrictive. According to NFPA, the pumps must be of horizontal end suction or horizontal split-case design. Single stage close coupled vertical in-line pumps may be used for limited capacities, but these pump types

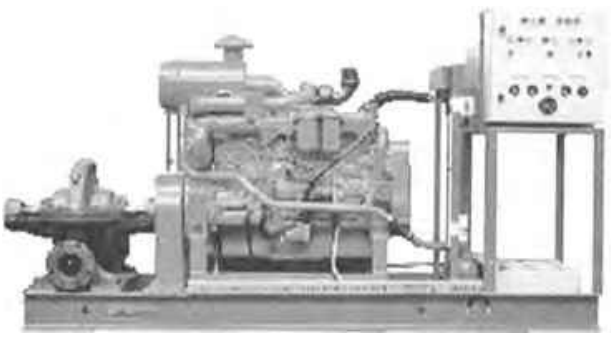


FIGURE 3 Split case fire pump (Flowserve Corporation)

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FIGURE 4 Close-coupled end suction pump (Flowserve Corporation)

may not be used where a static suction lift is required. If the pump must lift water from a well, panel, river, and so on, NFPA 20 requires the use of vertical pumps installed so the static water level is never below the bottom impeller.

Very high importance is given in all regulations to permanent availability of fire fighting water. Pumps must always be filled with water and must be ready for starting at any moment. This requirement is obviously satisfied when NFPA 20 compliance is mandatory. Although the same approach is preferred in Europe, some European codes nevertheless allow the use of horizontal pumps with a static suction lift. In such cases, the suction pipe must be fitted with a foot valve, and the pump set must be equipped with a controlled filling device.

Some rules common to both North America and Europe are

- The shape of the pump performance curves must conform to precise specifications.
- The design pressure of the pump casings is regulated.
- Hydrostatic tests are required.
- Certified performance test curves are required for most fire pumps.
- Fire fighting pumps must be approved, 'de jure' (according to written law or regulations) in North America and in most European countries; 'de facto' (according to well-established practices) in the others.

Stationary fire pumps can be purchased as packaged systems, which saves installation time and money. Single or multiple units of horizontal or vertical pumps can be packaged, and each packaged unit generally includes the pump(s), driver(s), controller(s), headers, accessories, and piping mounted on a common base. To the extent possible, all wiring and piping connections are made and the unit is factory-tested. Installation consists simply of positioning and leveling the package and making external piping and electrical connections.

FIRE PUMP DRIVERS

The principal objective of a fire pump driver is to provide the pump with motive power under any circumstances. Usual drivers are electric motors, diesel engines or, to a lesser extent, steam turbines. Although the reliability of the drivers themselves does not really pose problems, careful consideration must be given to the dependability of the power supply for electric motors and turbines, and to the fuel supply for diesel engines. Very complete regulations exist in every country which must be considered.

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Electric motors are the most economical driver type when a reliable power source is available. If this is not the case, power must be supplied by two or more independent sources, one of which may be an emergency generator set. Another choice might be to use one motor-driven pump and one diesel engine-driven pump. General guidelines for electric drives are

- Motor power and speed must be selected in accordance with the pump characteristics.
- Depending upon codes and available current, wound rotor, star delta, wye delta primary resistance, or part-winding start motors may be used.

Diesel engines are frequently used to drive stationary fire pumps. Equipped with battery packs and automatic controls, they rival electric motors for reliability and eliminate concern over the dependability of the source of electric power. In order to guarantee the fuel supply, the tank must be located above ground at an appropriate height. The prevailing philosophy of a diesel engine driven pump is that 'the pump must run' in actual fire conditions. All engine failures (oil pressure, cooling water temperature, speed) should be indicated only, and should not stop the engine. The only exception to this is the NFPA 20 requirement that a shutdown occur at an over speed of 120%.

Here are some general guidelines for diesel engines:

- Engine power and speed must be selected in accordance with the pump characteristics (diesel engines are very sensitive to altitude and ambient temperature).
- All energy sources for engine starting must be doubled: two batteries, or two air containers, or electrical and manual recharge of the hydraulic starting system, and so on.
- An instrument panel secured to the engine at an appropriate place must include a tachometer, an hour meter, an oil pressure gage, and a water temperature gage.

Occasionally, a fire pump is driven by a steam turbine. When it is desirable to use steam as the power source, details of the steam supply and exhaust need to be carefully planned to ensure that the required level of reliability exists.

FIRE PUMP CONTROLLERS

Regardless of the type of driver, most fire pumps are started automatically by a pressure signal from the pump discharge line. Each fire pump must have its own controller, including the jockey pump.

Depending upon application requirements, fire pump electric motor controllers must conform to the appropriate national regulations. Each controller must be arranged to match the starting characteristics of its motor and must include

- Manual disconnect switch
- Circuit breaker
- Starter without heaters or contactors
- Pressure switch (when NFPA is mandatory)
- Minimum-run timer to prevent motor cycling

The diesel engine controller is arranged to permit either automatic or manual start. The manual start is required to permit periodic run tests. The power source for the controller and for starting is a dual set of batteries or air containers. The controller is arranged to show the following engine conditions:

- Low lubrication oil pressure
- High water jacket temperature
- Failure to start automatically

- Shutdown due to overspeed (when NFPA 20 is mandatory)
- Battery and battery charger failure or, when equipped with one, an air system failure

The engine controller must also provide a means of relaying the following information to remote indicators:

- Engine running
- Engine switch off or manual position
- Trouble signal (activated by one or any combination of the engine or controller signals described above)

Approval of the control system for all pumps, except the jockey pump, is required in nearly all national codes.

GENERAL ENGINEERING PROCEDURE FOR FIRE PROTECTION SYSTEMS

As described earlier, fire protection systems can vary greatly depending on the type and significance of the fire risk, the site conditions and, finally, the various national regulations in force in the given country. Outlined below are engineering guidelines for stationary installations using water as protection agent. Figures 5 and 6 show typical installations for horizontal and vertical fire pumps.

Here are some project basics to keep in mind:

1. Refer to an insurance expert, or to a specialist of a certification board, to confirm that water may be used for the risks of concerned and to determine which kind of spray system is recommended: sprinklers, water jets, deluge installations, and so on.
2. Determine the appropriate governing regulations: NFPA (North America), APSAD (France), VDS (Germany), and so on.
3. Ensure the availability of sufficient water capacity, according to the type of risk. Preference should be given to a water supply above ground, which is generally more reliable and less expensive than a below-ground supply.

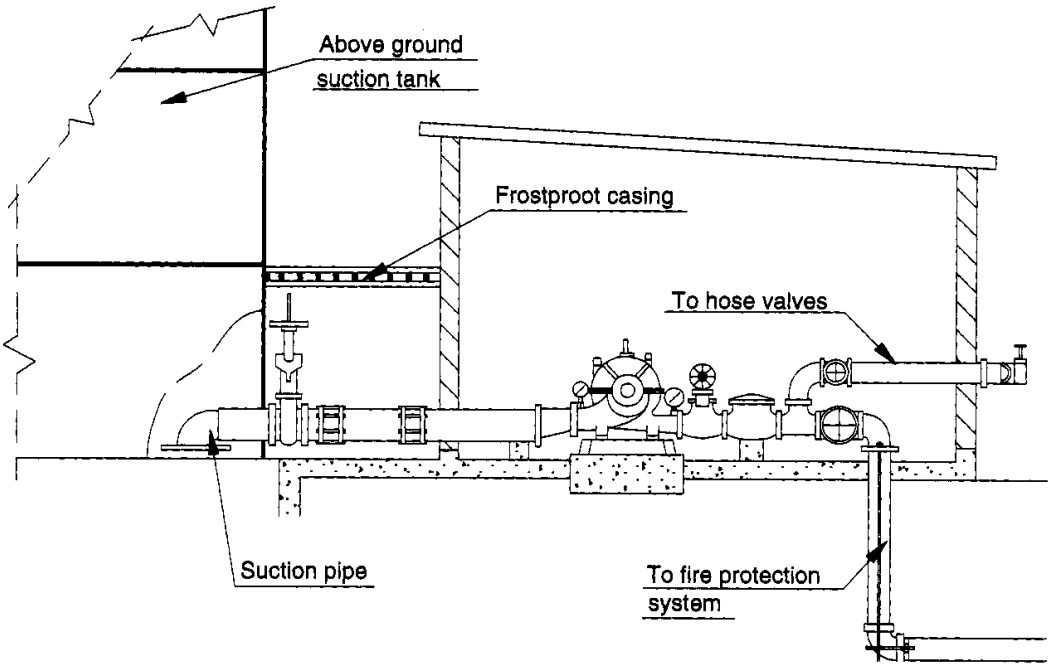


FIGURE 5 Typical installation for horizontal fire pumps

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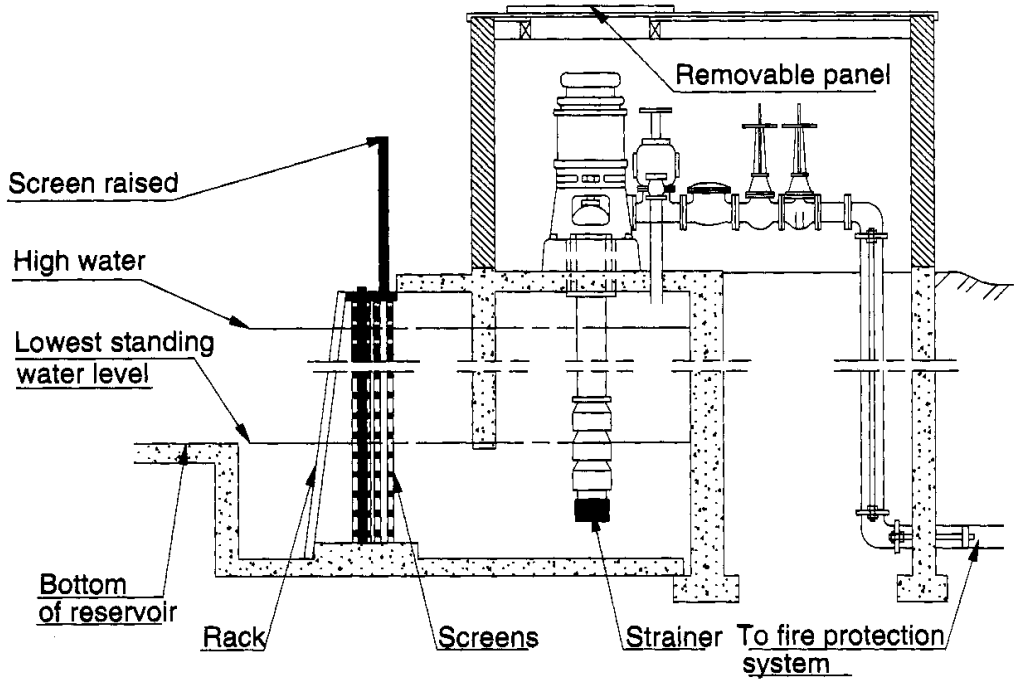


FIGURE 6 Typical installation for vertical fire pumps

4. Distinguish all the critical areas and functions to be protected on the site.
5. Determine the motive power of the pump drivers. The objective is to be independent from the site of risk. Diesel engine-driven installations many times provide a convenient, independent power source that addresses this concern. The costs associated with a diesel engine power source is usually equivalent to those of electrical installations; in some cases, they are even lower.
6. Provide a pump house, or a pump room, so the pump, driver and controller may be protected against possible damage or injury. In addition to the normal precautions and concerns of a pump installation, also consider the following factors:
   - Easy access for installation and maintenance
   - Sufficient ventilation for motors and engines
   - Risk of earthquakes
   - Risk of freezing
   - Noise protection, where required
7. Determine the pipe network and all required fittings and accessories in order to obtain the lowest project costs.
8. Smaller pipe sizes (less expensive) increase friction losses and may require more powerful (more expensive) pump sets.
9. Larger pipe sizes (more expensive) decrease friction losses and may allow the use of less powerful (less expensive) pump sets.

Keep in mind the following factors when selecting a pump:

1. Compute the total capacity and pressure required to feed the protection system.
2. Determine the NPSH available at the pump suction nozzle. This is an important selection criterion for establishing the speed and design of the pump. Reminder: U.S regulations forbid static suction lifts for fire pumps, whereas most European regulations will allow limited suction lifts.

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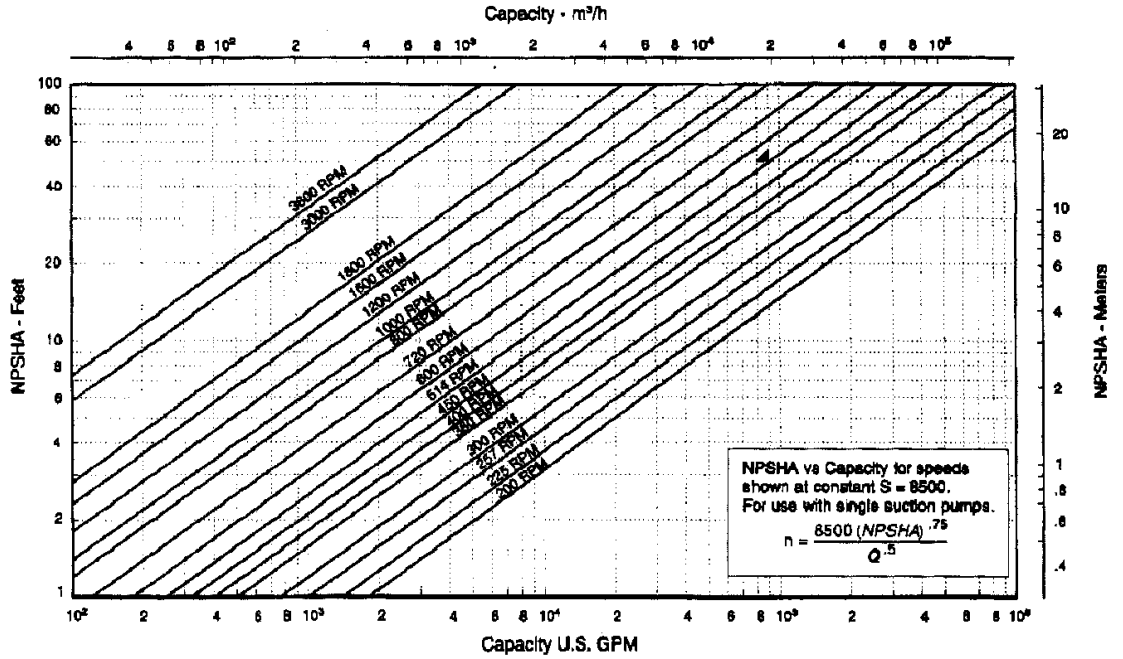


FIGURE 7 Recommended operating speeds for single suction pumps (Hydraulic Institute ANSI/HI 2000 Edition Pump Standards, Refernce 1)

3. Select the highest possible pump speed based on the required capacity and the available NPSH. The higher the allowable pump speed, the lower the cost of the pump set. Figures 7 and 8 give recommended operating speeds for single suction and double suction pumps.
4. Select pump size and design according to the governing (applicable) regulations.

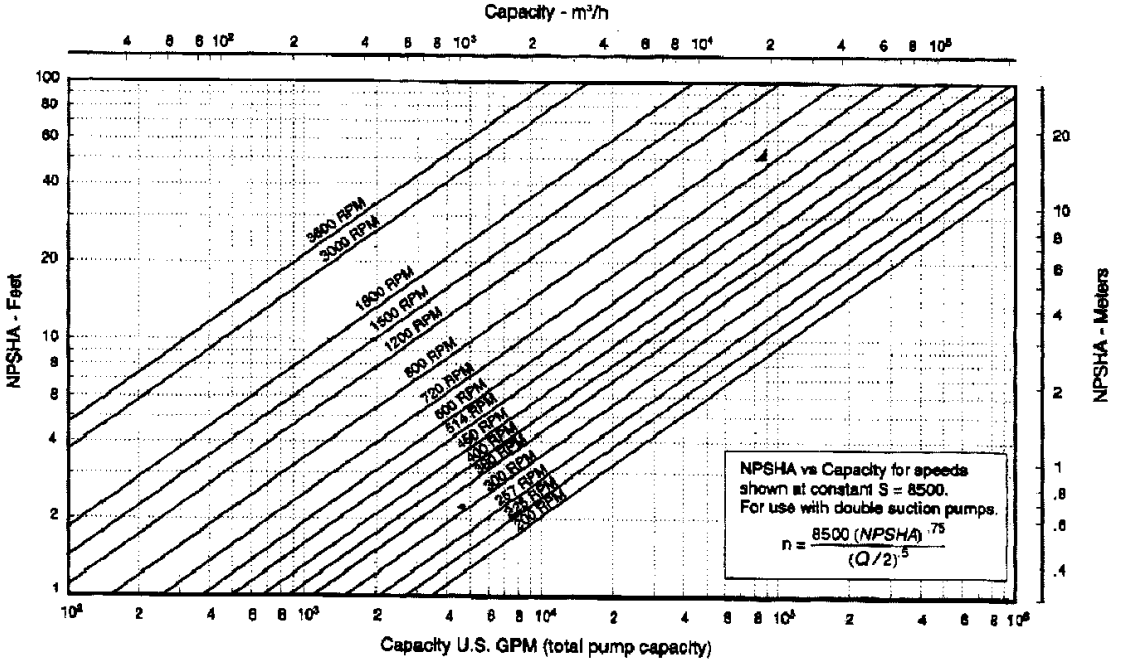


FIGURE 8 Recommended operating speeds for double suction pumps (Hydraulic Institute ANSI/HI 2000 Edition Pump Standards, Refernce 1)

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Drivers and Controllers As outlined earlier under 'Fire Pump Criteria,' it is recommended that a completely packaged fire pump set, including the pump(s), driver(s), controller(s), headers, accessories and piping, mounted on a common base, be furnished.

PUMP DESIGN AND MATERIALS

Constant operational readiness and a high degree of operational reliability are the most important aspects of fire pump designs. Although not mandatory, the features listed and described next are recommended for fire pumps.

- Shaft sealing should be by means of stuffing boxes (packing).
- Grease-lubricated rolling element bearings should be used on horizontal pumps. Water-lubricated (rubber) sleeve bearings (open line shaft) or oil-lubricated bronze sleeve bearings (enclosed line shaft) and, if so equipped, a grease-lubricated thrust bearing should be used on vertical pumps.
- Combinations of materials, running clearances, and rotor end plays should permit trouble-free operation, even if the pump has been idle (not run) for extended periods of time.
- Repairs should be achievable by means of standard tools.
- Corrosion-resistant materials must be utilized if the fire-fighting water is corrosive, such as brackish or salt water.
- Additional features or specific materials, as may be mandated by specific national standards and codes.

Because most fire pumps handle clear water, and because wear (from severe usage) is not usually a problem, fire fighting pumps should be of the simplest design possible and should be made of commonly used industrial materials. The most economical fire pump is either a single-stage, end suction, or vertical in-line pump. This class of pump is readily available to heads of approximately 260 ft (80 m). Unfortunately, capacities have been limited in the United States (by NFPA 20) to 750 gpm (200 m^3/h) for horizontal end suction pumps and to 500 gpm (115 m^3/h) for vertical in-line pumps. The 1996 edition of NFPA 20 eliminated these historical limitations. In Europe, however, end suction volute pumps are commonly employed up to 4000 gpm (900 m^3/h).

The jockey pump (Figure 1), used to maintain the pressure within the system, is generally a small, vertical multistage pump, utilizing an extended motor shaft (no shaft coupling). Recommended materials for jockey pumps are

Casings:	Cast iron
Impellers:	Bronze or thermoplastic resin
Diffusers:	Cast iron
Shaft:	Chromium steel
Bearings:	Grease lubricated rolling element bearings, with an additional product lubricated bronze line bearing in close coupled designs

A large end suction fire fighting pump, as shown in Figure 2, is typical of units used in Europe and allowed to be operated over the entire pump capacity range, from shutoff to the highest capacities. The pump exhibits the following features:

- Back pull out construction for ease of maintenance without disturbing the driver
- The absence of shaft sleeves, giving highest possible shaft stiffness
- A bronze bearing bushing located behind the impeller that acts as a product lubricated auxiliary line bearing
- A conical (tapered shaft) impeller fit to ensure no looseness between the mounted impeller and the shaft (critical when pumps are driven by a diesel engine)

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Recommended materials for end suction fire fighting pumps are

Casing:	Cast iron or ductile iron
Impeller:	Bronze
Shaft:	13% chromium steel
Bearings:	Grease-lubricated rolling element bearings

For larger capacities, in excess of 500—750 gpm (115–200 m^3/h) in the United States and above approximately 4000 gpm (900 m^3/h) in Europe, single stage double suction axially split case pumps (Figure 9) are required. These larger units are more costly than the simple single stage overhung designs, but they do offer some advantages that could make them desirable even at lower flow rates, such as

- The required NPSH is lower (double suction impeller), thus allowing higher operating speeds.
- The axially split case design provides easy access to the pump interior (neither the pump nor the driver need to be disconnected) for inspection and rotor removal.
- Axially balanced hydraulic axial forces, and bearings on each end of the rotor supports unbalanced radial loads.

Recommended materials for single stage double suction axially split case for fighting pumps are

Casing:	Cast Iron
Impeller:	Bronze
Shaft:	Carbon steel
Sleeves:	Bronze
Casing Rings:	Bronze
Bearings:	Grease-lubricated rolling element bearings

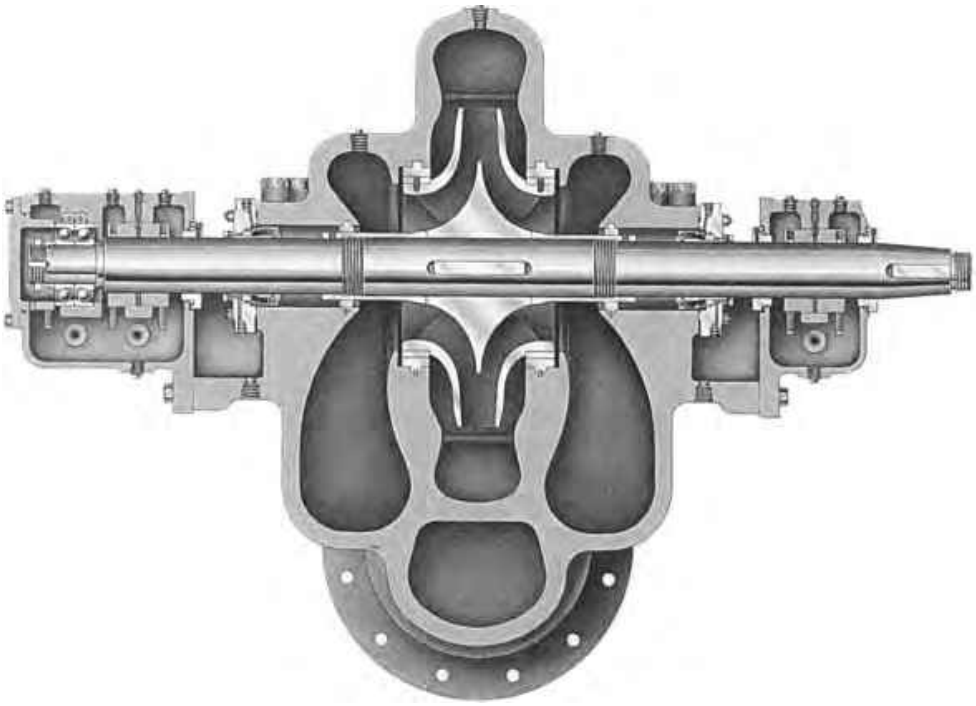


FIGURE 9 Single stage double suction axially split case pump used for larger flow rates (Flowserve Corporation)

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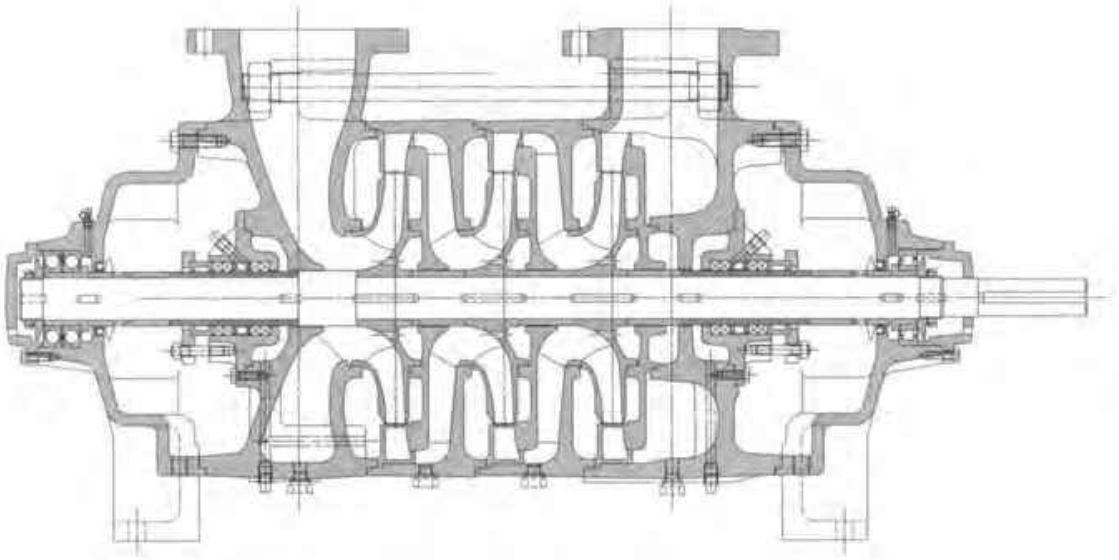


FIGURE 10 Multistage ring-section fire pump (Flowserve Corporation)

Normally, head requirements for fire pumps do not exceed 260 ft (80 m). For those systems that do require more head, end suction and axially split case pumps may be available up to 450 ft (140 m). Beyond this head range, multistage pumps are typically employed. Multistage pumps for fire services in Europe are usually of the ring-section design (Figure 10), whereas the axially split case design (Figure 11 shows a typical two-stage pump; Figure 12 shows a typical multistage pump) tends to be more popular in North America.

In North America, vertical pumps must be used when the water source is located below ground. In Europe, where the use of a limited static suction lift is generally permitted, such vertical pumps are required only when the NPSH available at the suction nozzle of a horizontal pump would exceed a certain limit, as may be established by national regulations in each country—for instance 16.4 ft (5 m) in France.

The submergence of vertical pumps require a particular care. As a minimum, the second stage (second impeller up from the bottom end of the pump) should always be submerged below the minimum water level.

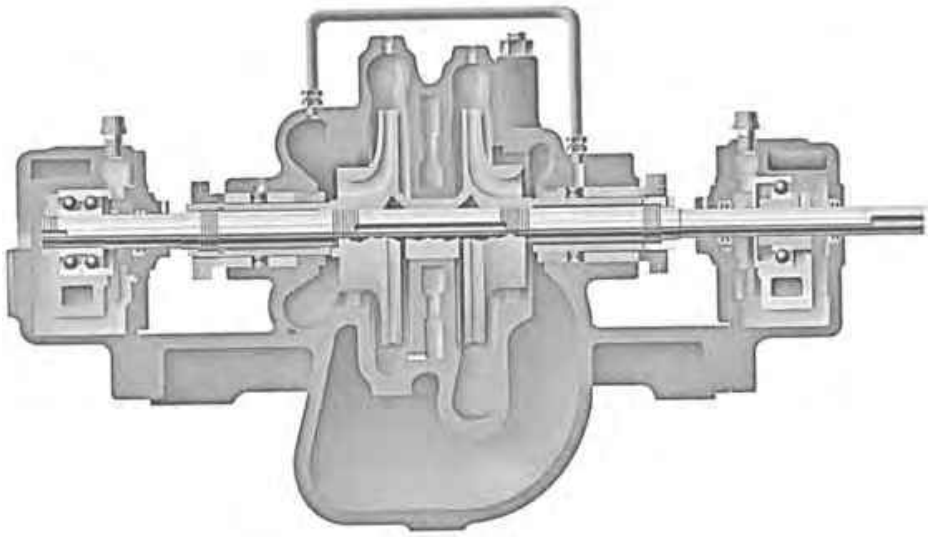


FIGURE 11 Two-stage axially split pump (Flowserve Corporation)

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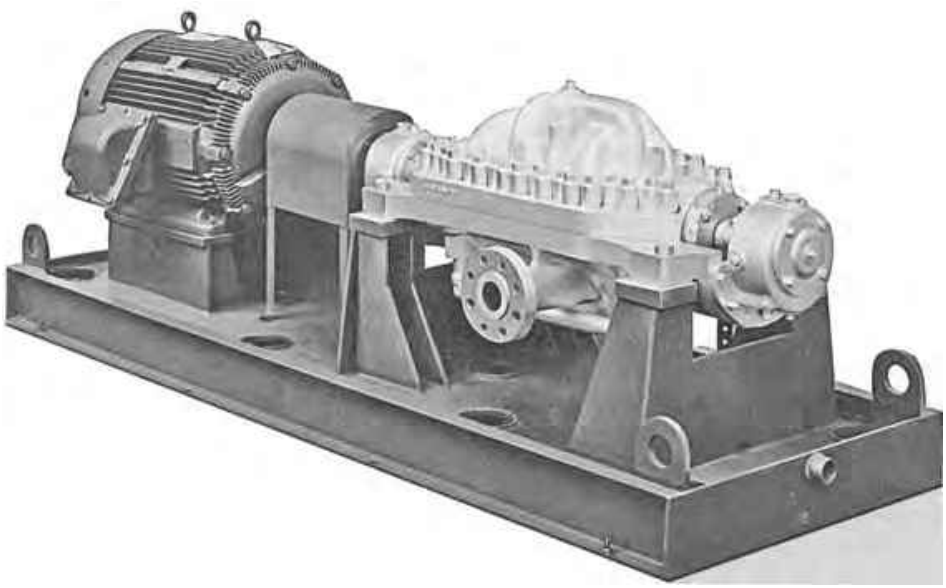


FIGURE 12 Typical axially split multistage pump (Flowserve Corporation)

The inner column (drive shafting) for vertical pumps is available in either of two configurations:

- Open line shaft (OLS) construction (Figure 13), within which a steel shafting rotates in water-lubricated rubber bearings, centered and stabilized by rigid bearing retainers, is used for static water levels 50 ft (15 m) or less below the pump discharge flange.
- Enclosed line shaft (ELS) construction (Figure 14), in which steel shafts rotate in oil-lubricated bronze sleeve bearings, is used for static water levels of more than 50 ft (15 m). In ELS construction, the outside of the sleeve bearings are threaded and tubes enclosing and supporting the bearings and shafts isolate the shafting from the water being pumped.

If an oil-lubricated line shaft design is utilized, the appropriate environmental protection or health department should be consulted with regard to special requirements for installation and protection.

Recommended materials for vertical pumps are

Discharge head and columns:	Cast iron or welded steel
Water-lubricated line shaft:	Carbon steel protected by nonrusting shaft sleeves through packing and water-lubricated bearings (Note: chromium steel shafts typically do not require shaft sleeve protection.)
Oil-lubricated line shaft:	Carbon steel possible, but chromium steel is preferred
Pump shaft:	Chromium steel
Bowls:	Cast iron
Impellers:	Bronze

OVERVIEW OF COMMONLY USED ACCESSORIES

See Figures 15 and 16 for typical fire pump systems.

1. Common to horizontal and vertical pumps:
   a. **Casing relief valve** To prevent a no-flow condition when system is running at shutoff



FIGURE 13 Vertical pump with open line shaft construction (Flowserve Corporation)



FIGURE 14 Vertical pump with enclosed line shaft construction (Flowserve Corporation)

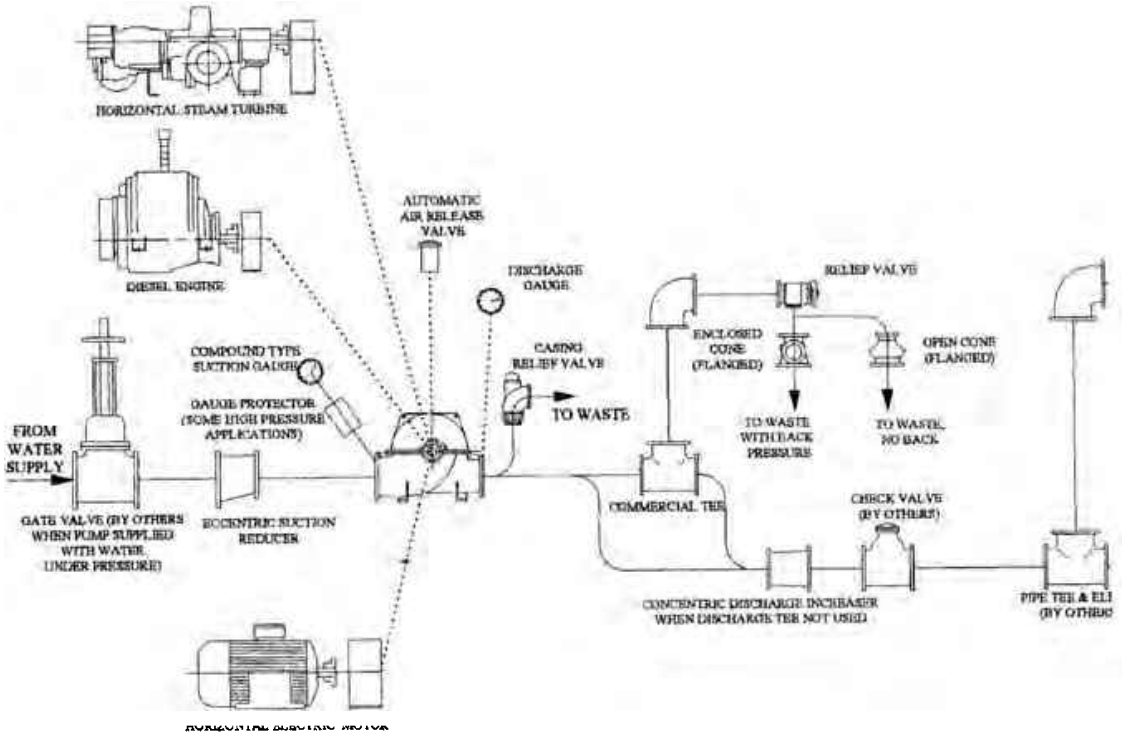


FIGURE 15 Typical horizontal fire pump system

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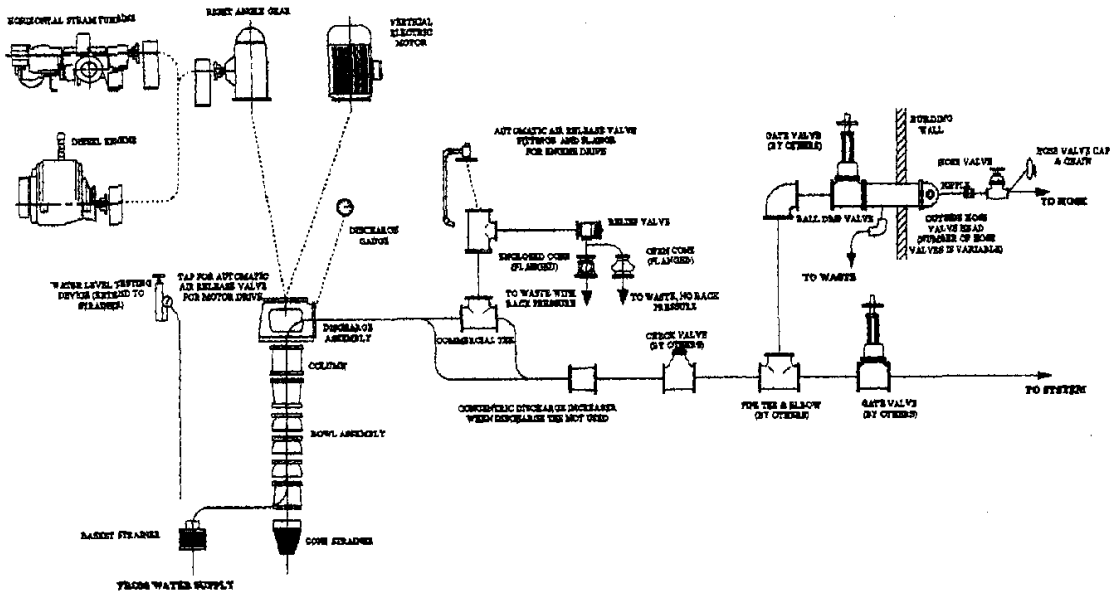


FIGURE 16 Typical vertical fire pump system

- **b. Automatic air release valve** To vent entrapped air in automatic systems
- **c. Umbrella cock** To vent air in manually operated systems
- **d. Hose valve head with hose valves, caps, and chains** To permit pump flow (capacity) tests (most modern systems use flowmeters in lieu of hose valve heads)
- **e. Ball drip valve** Installed upstream to prevent freeze damage to hose valve head installed outside
- **f. Overflow cone** To show whether the relief valve is open
- **g. Commercial discharge tee with (if required) 90° elbow** Used when main relief valve is required

2. Horizontal pumps only
   - **a. Eccentric suction reducer** Required when the size of the suction pipe does not match the size of the pump suction connection
   - **b. Concentric discharge increaser** Required when the size of the discharge pipe does not match the size of the pump discharge connection
   - **c. Main relief valve** Required when pump shutoff pressure plus suction pressure exceeds system design pressure and when engine or other variable-speed driver is used
   - **d. Splash partition** Used for motor-driven units where hose head are mounted indoors near pump
3. Vertical pumps only
   - **a. Main relief valve** Required when engine or other variable-speed driver is used
   - **b. Water level testing device** To determine distance to surface of water; required for well pump installations

REFERENCES AND FURTHER READING

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SECTION 12.5
STEAM POWER PLANTS

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STEAM POWER PLANT CYCLES

Power is produced in a basic steam power plant by pressurizing water, adding heat energy to produce steam under pressure, and transforming part of that steam's thermal energy into mechanical energy to do useful work. The pressurized water, called feedwater, acts as a conveyor of energy. Modern steam power plants use centrifugal pumps to deliver pressurized water to some heat source such as a boiler, steam turbines as heat engines to transform thermal energy into mechanical energy, and condensers to recover the steam in a continuous cycle. Most steam power plants produce electric power with generators driven by the steam turbines.

This basic cycle is improved by connecting a condenser to the steam turbine exhaust and by heating the feedwater with steam extracted from an intermediate stage of the main turbine. This results in an improvement of the cycle efficiency, provides deaeration of the feedwater, and eliminates the introduction of cold water into the boiler and the resulting temperature strains on the latter. The combination of the condensing and feedwater heating cycle (Figure 1) requires a minimum of three pumps: the condensate pump, which transfers the condensate from the condenser hot well into the direct-contact heater; the boiler-feed pump; and a circulating pump, which forces cold water through the condenser tubes to condense the exhaust steam. This cycle is very common and is used in most small steam power plants. A number of auxiliary services not illustrated in Figure 1 are normally used, such as service water pumps, cooling pumps, ash-sluicing pumps, oil-circulating pumps, and the like.

The required improvements in operating economy in the 1970s dictated further refinements in the steam cycle, and these created new demands for power plant centrifugal pumping equipment. This evolution involved a steady increase in operating pressures until 2400 lb/in^2 (165 bar*) steam turbines became quite common. Many plants are operating at supercritical steam pressures of 3500 lb/in^2 (240 bar). Several central station plants constructed in the late 1970s are operating between 4000 and 5000 lb/in^2 (275 and 345 bar).

Other refinements were directed toward a greater utilization of heat through increased feed-water heating, introducing a need for heater drain pumps-equipment with definite

*1 bar = 10^5 Pa.

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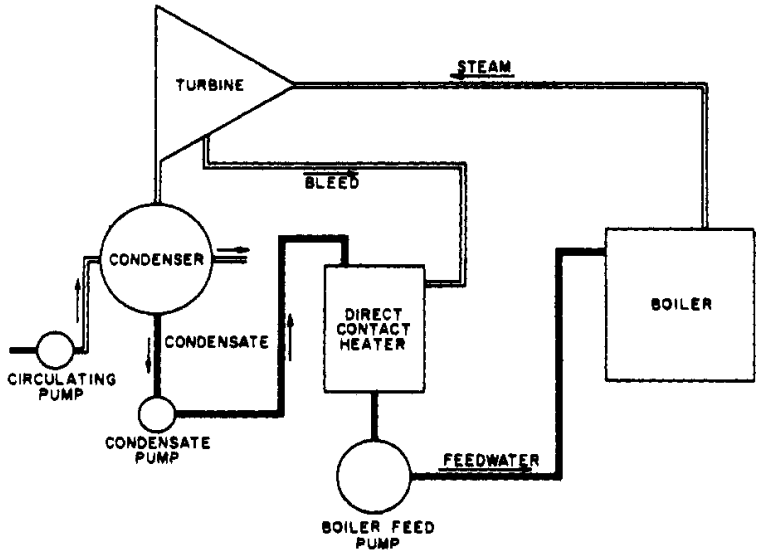


FIGURE 1 Simple steam power cycle

problems of its own. Finally, the introduction of forced or controlled circulation as opposed to natural circulation at 650°F (343°C) in boilers created a demand for pumping equipment of again an entirely special character.

Although direct-contact heaters would have thermodynamic advantages, a separate pump would be required after each such heater. The use of a group of closed heaters permits a single boiler feed pump to discharge through these heaters and into the boiler. The average power plant is based on a compromise system: one direct-contact heater is used for feedwater deaeration, whereas several additional heaters of the closed type are located upstream as well as downstream of the direct-contact heater and of the boiler-feed pump (Figure 5). Such a cycle is termed an *open cycle.* The major variation is the *closed cycle,* where the deaeration is accomplished in the condenser hot well and all heaters are of the closed type (Figure 6).

In the 1970s, steam power plant technology advanced with the implementation of nuclear fission reactors as heat sources in commercial electric power generation. Steam cycles with nuclear fission reactors utilize two concepts for producing steam: boiling water reactors (Figure 3) in which feedwater is delivered directly to the reactor and pressurized water reactors (Figure 4) in which reactor heat energy is transferred indirectly to the feedwater by steam generators. In both cases, the basic steam cycle is the same as in steam power plants with conventional fossil fuel-fired boilers. However, feedwater pressures for nuclear fission reactors are substantially less feedwater pressures for modern fossil fuel-fired boilers.

In the 1980s, steam power plant technology was reapplied to recover exhaust energy from gas turbines and thereby create an efficient form of electric power generation. These combined-cycle power plants have multiple electric generators driven by combinations of gas turbines and steam turbines (Figure 2). The heat source for the steam cycle is called a heat recovery steam generator (HRSG), which transfers heat energy from the hot exhaust gases of a gas turbine to the feedwater. The pumps in these steam cycles are similar to, but usually smaller than those typically found conventional fossil fuel-fired boilers.

In the 1990s, steam power plant technology advanced yet again with new boiler designs and improved emission controls. Once-through supercritical designs improve the heat rates and commercial viability of fossil fuel-fired boilers. Fluidized bed-type boilers and flue gas desulfurization subsystems, such as wet-limestone scrubbers, reduce emissions from coal-fired plants. The reemergence of these coal-fired plants has required larger, higher-energy feedwater pumps and new slurry-type pumps.

Presently, the world's electric power producers are utilizing and constructing all three types of power plants: combined-cycle, nuclear reactor, and fossil fuel-fired boiler. This

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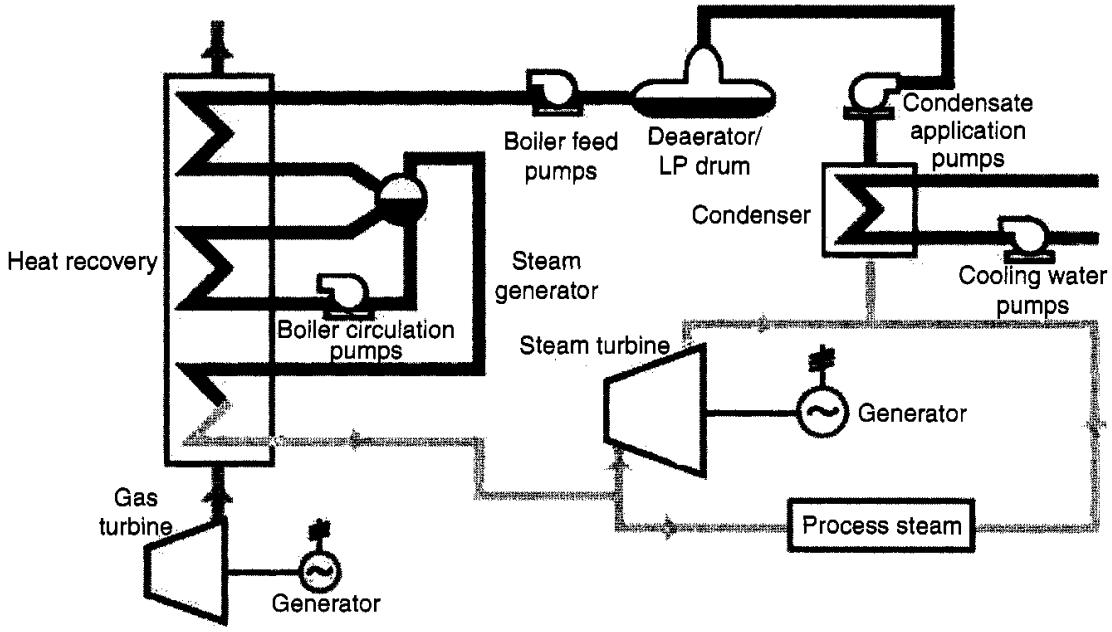


FIGURE 2 Natural gas and steam, combined cycle power

multiple-type approach is expected to continue as power producers worldwide strive to serve ever growing electric energy demands and as changes in technologies, local fuel resources, environmental needs, and commercial factors emerge. Going forward, pumps will continue to serve both traditional and specialized requirements in new steam power plants.

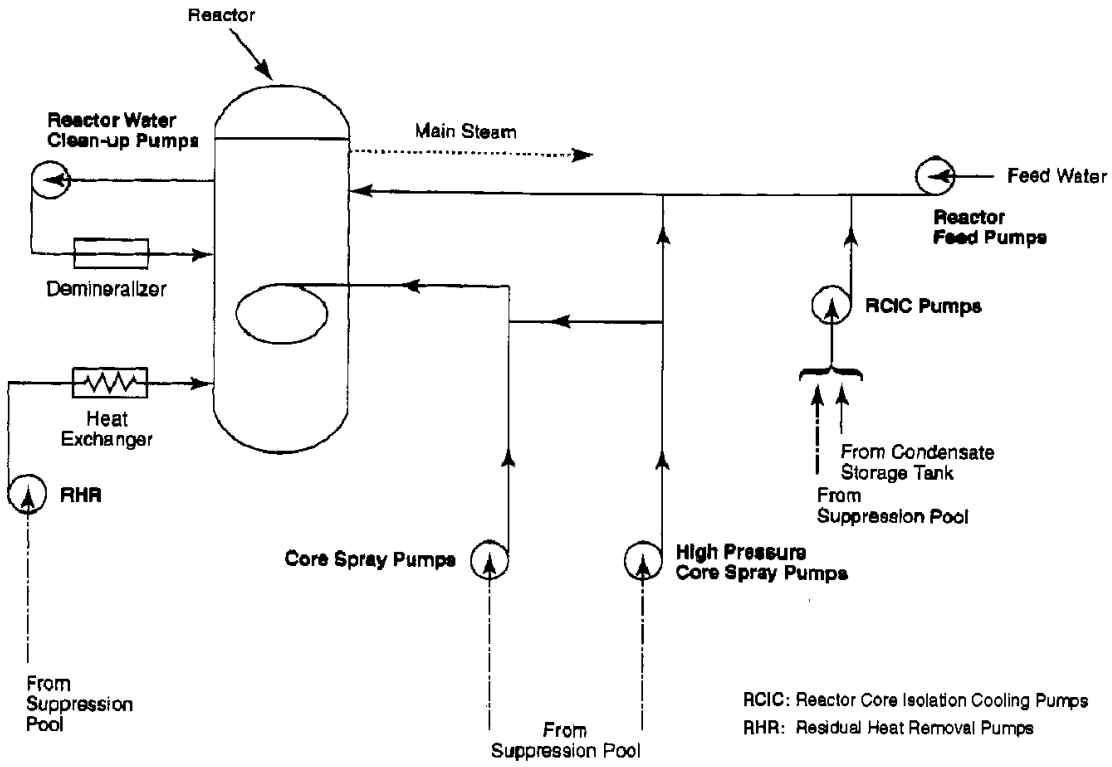
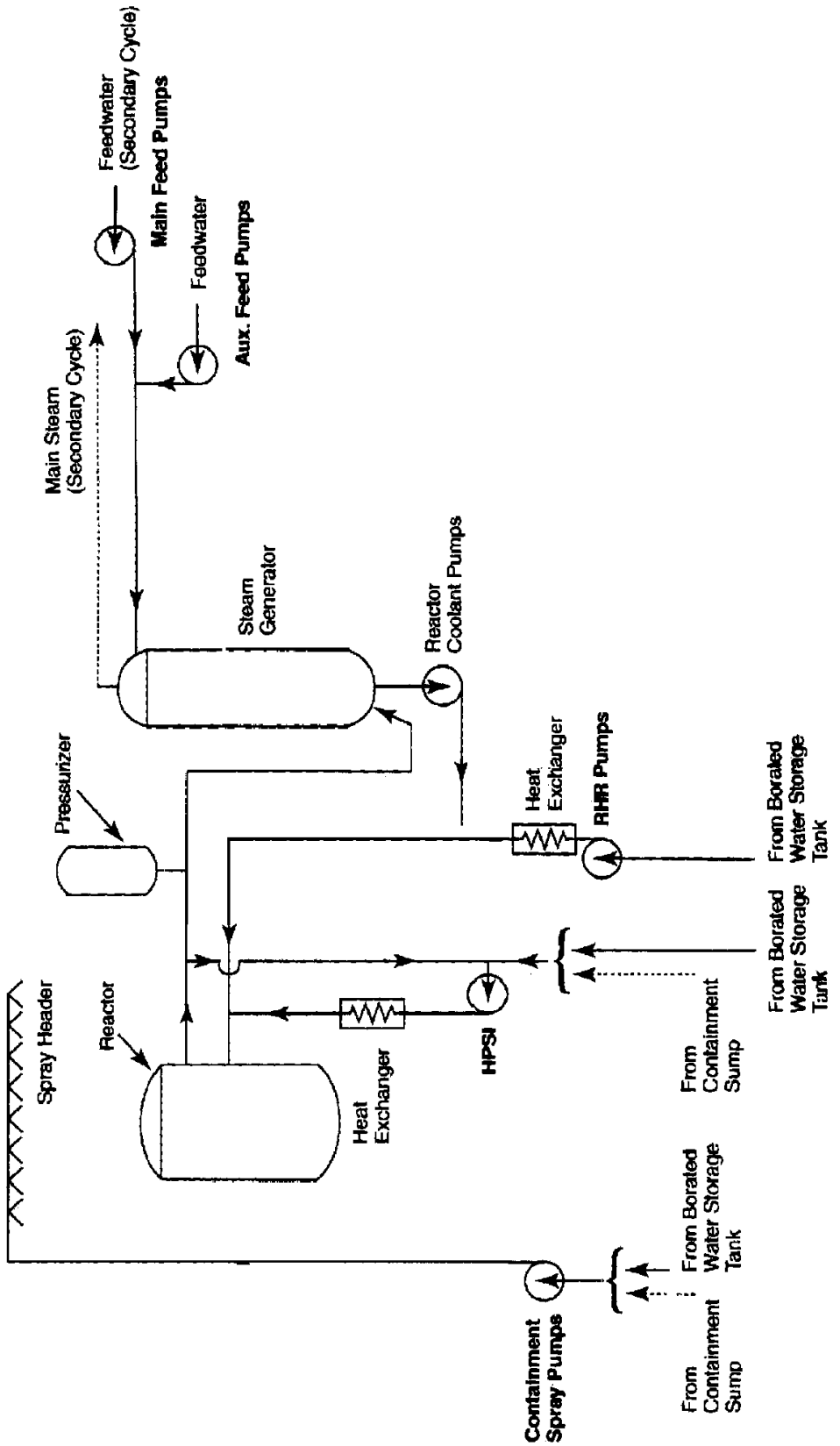


FIGURE 3 Nuclear power steam cycle, boiling water reactor

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RHR: Residual Heat Removal Pumps
HPSI: High Pressure Safety Injection Pumps

FIGURE 4 Nuclear power steam cycle, pressurized water reactor

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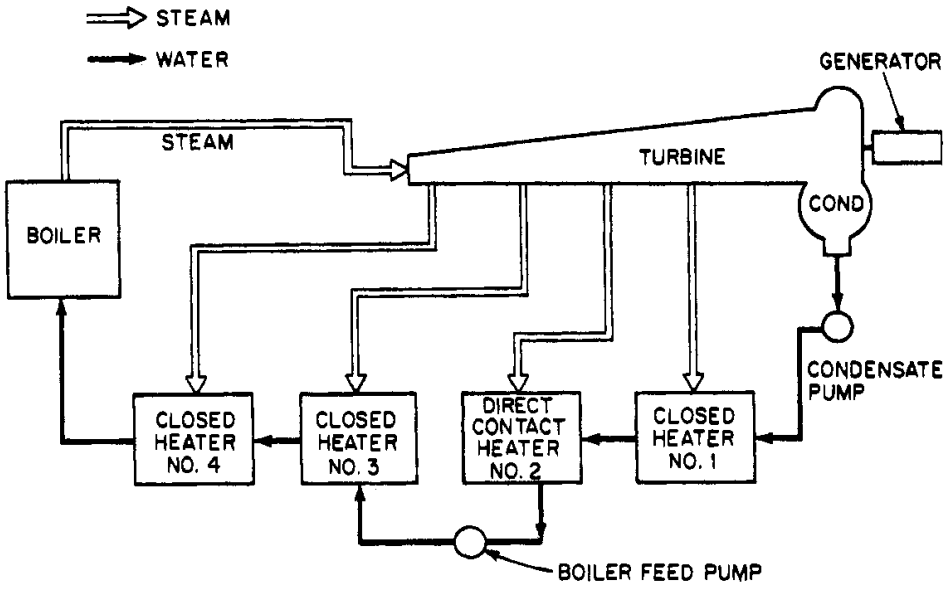


FIGURE 5 Open feedwater cycle with one deaerator and several closed heaters

STEAM POWER PLANT PUMPING SERVICES

Pumps are very important components of a steam electric power plant. The major applications are the condensate, boiler-feed, heater drain, and condenser circulating pumps. The all-inclusive category of "miscellaneous pumps" includes such a variety of services that it merits being broken down into its components and included in a representative listing. Table 1 provides such a listing for conventional (fossil fuel) steam power plants. The list is not necessarily complete but is reasonably representative.

BOILER-FEED PUMPS

Under the term *conditions of service* are included not only the pump capacity, discharge pressure, suction conditions, and feedwater temperature but also the chemical analysis of

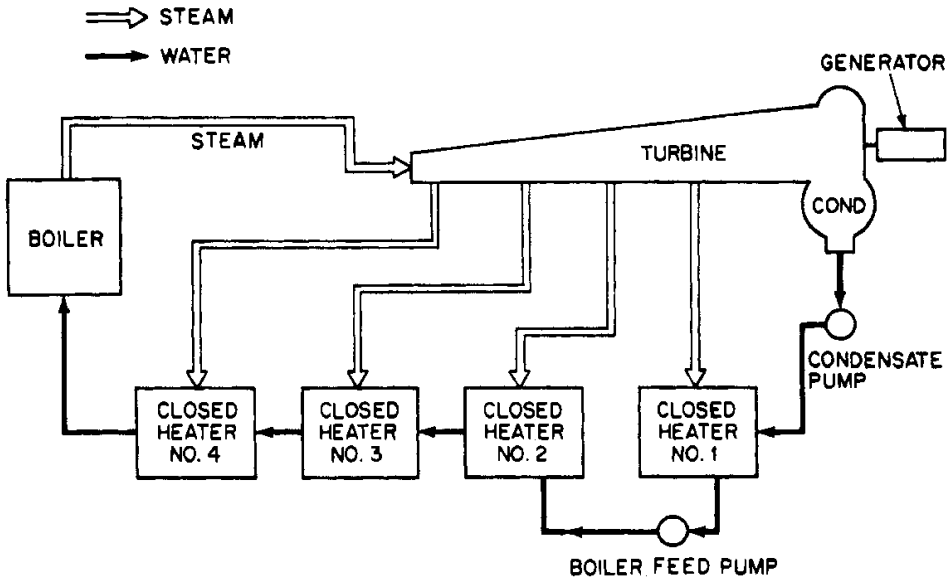


FIGURE 6 Closed feedwater cycle

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TABLE 1 Pump services in conventional steam power plants

- Turbogenerator and auxiliaries
  - Condenser circulating pumps
  - Screen wash-water pumps
  - Cooling tower make-up pumps
- Steam generator equipment
  - Condensate pumps
  - Condensate booster pumps
  - Boiler-feed pumps
  - Boiler-feed booster pumps
  - Deaerator make-up pumps
  - Heater drain pumps (low and high pressure)
- Chemical feed system
  - Amine pumps
  - Hydrazine pumps
  - Phosphate pumps
  - Caustic feed pumps
  - Acid feed pumps
  - Ammonia pumps
  - Regeneration waste pumps
  - Demineralizer pumps
  - Neutralizing metering pumps
  - Neutralizing tank sump pumps
  - Acid bulk-transfer pumps
  - Caustic bulk-transfer pumps
  - Inlet and effluent demineralizer waste tank pumps
- Fuel oil system
  - Fuel oil transfer pumps
  - Secondary fuel oil pumps
  - Secondary fuel oil heater drip pumps
  - Ignitor oil pumps
  - Auxiliary boiler fuel pumps
  - Warm-up oil pumps
  - High-temperature oil-circulating pumps
- Fuel oil system (continued)
  - Low-temperature oil-circulating pumps
  - Distillate oil unloading pumps
  - Fuel oil additive unloading pumps
  - Fuel oil additive transfer pumps
  - Fuel oil additive metering pumps
  - Fuel oil hose drain pumps
- Lubricating oil system
  - Lubricating transfer pumps
  - Starting oil pumps
  - Main oil pumps
  - Emergency oil pumps
  - Centrifuge feed pumps
- Fire protection system
  - Fire pumps
  - Jockey pumps
  - Foam proportioning pumps
- Heating, ventilating, and air conditioning system
  - Hot water circulating pumps
  - Chilled water pumps
- Service water system
  - Service water pumps
  - Air preheater wash pumps
  - Cooling water booster pumps
  - Primary air heating coil condensate return pumps
  - Heating drain tank return pumps
  - Sump pumps
  - Closed cooling water system pumps
- Miscellaneous
  - Ash sluice pumps
  - Slurry pumps
  - Acid cleaning pumps
  - Hydrostatic pressure test pumps

the feedwater, the pH at pumping temperature, and other pertinent data that may reflect upon the hydraulic and mechanical design of the boiler-feed pumps. Preferably, a complete layout of the feedwater system and of the heat balance diagram should be supplied to the boiler-feed pump manufacturer. The study of this layout will often permit the manufacturer to suggest an alternate arrangement of the equipment that would result in a more economical operation, in a lower installation cost, or even in longer equipment life to reduce the eventual maintenance expense.

Boiler-Feed Pump Capacity Total boiler-feed pump capacity is determined by adding margins for system surge and pump wear to the maximum required flow of the boiler. Typical flow margins range from 5 to 10% depending on the size of the boiler. The total required boiler flow may be served by a single pump or by some combination of identical pumps operating in parallel.

A common boiler-feed pump arrangement for large central power plants includes two auxiliary stream turbine-driven main boiler-feed pumps each with 50% boiler capacity and one electric motor-driven start-up boiler-feed pump with 20 to 30% boiler capacity. However, arrangements vary widely depending on preferences of different owners and engineers. Some owners prefer the use of auxiliary boilers in lieu of start-up pumps. Some have used a single 100% boiler feedwater pump for units as large as 1300 MW (Figure 7). Other owners require installed spare pumps in cases where pump element change-out cycle times are too long for a given plant availability requirement.

An HRSG in a combined cycle power plant typically has multiple conditions of service with differing pressures and flow requirements. These different conditions can be served

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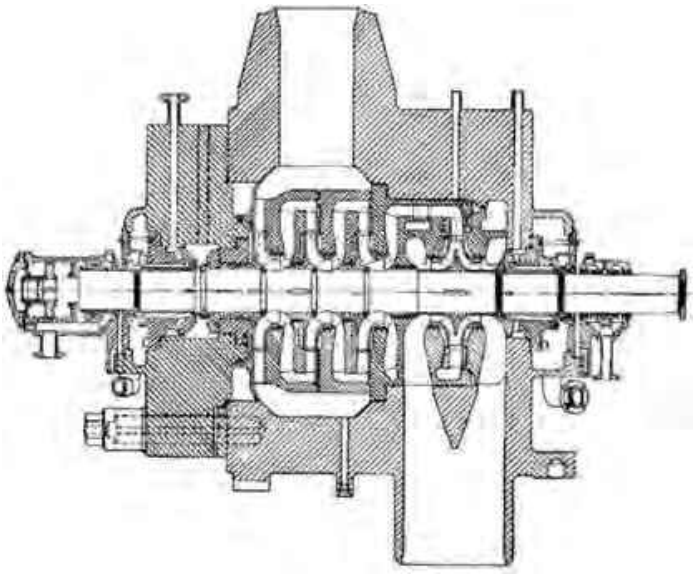


FIGURE 7 Cross-section, single 65,000 hp boiler feed pump, 1300 MW fossil power plant (Flowserve Corporation)

by unique high-pressure, medium-pressure, and/or low-pressure pumps. Or, they can be served by one high-pressure pump with medium- and/or low-pressure takeoff connections.

Suction Conditions The net positive suction head (NPSH) represents the net suction head at the pump suction, referred to the pump centerline, *over and above* the vapor pressure of the feedwater. If the pump takes its suction from a deaerating heater, as in Figure 5, the feedwater in the storage space is under a pressure equivalent to the vapor pressure corresponding to its temperature. Therefore the NPSH is equal to the static submergence between the water level in the storage space and the pump centerline less the frictional losses in the intervening piping. Theoretically, the required NPSH is independent of operating temperature. Practically, this temperature must be taken into account when establishing the recommended submergence from the deaerator to the boiler-feed pump. A margin of safety must be added to the theoretical required NPSH to protect the boiler-feed pumps against the transient conditions that follow a sudden reduction in load for the main turbogenerator.

Although the previous discussion applies primarily to the majority of installations, where the boiler-feed pump takes its suction from a deaerating heater, it holds as well in the closed feed cycle (Figure 6). The discharge pressure of the condensate pump or the booster pump must be carefully established so the suction pressure of the boiler-feed pump cannot fall below the sum of the vapor pressure at pumping temperature and the required NPSH.

Careful attention must be given to any strainer that might be installed in the pump suction piping. The pressure drop increase across the strainer is indicative of foreign material and it reduces the net positive suction head available (NPSHA) to the pump. Strainers in the pump suction pipe are most often removed following plant start-up qualification testing.

Transient Conditions Following Load Reduction Following a sudden load reduction, the turbine governor reduces the steam flow in order to maintain the proper relation between turbine and generator power and to hold the unit at synchronous speed. The consequence of this reduction is a proportionate pressure reduction at all successive turbine stages, including the bleed stage that supplies steam to the deaerator. The check valve in the extraction line closes and isolates the heater from the turbine. As hot feedwater continues to be withdrawn from the heater and cold condensate to be admitted to the heater, the pressure in the direct-contact heater starts to drop rapidly. The check valve reopens when the heater pressure has been reduced to the prevailing extraction pressure and stable conditions are reestablished.

It should be noted that, even though the feedwater system in a drum boiler may be provided with a three-element feedwater regulator, the feedwater flow will not instantaneously follow the steam flow as soon as the steam demand is reduced by a reduction in

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unit load. Because of the time lag between the reduction in steam demand and that of the fuel-burning rate and because of the heat retention in the steam generator, there is a momentary rise in the boiler pressure, with the resultant collapse of some of the steam and water bubbles in the boiler drum. This lowers the apparent boiler drum level, causing the level control to override to some degree the impulse from the change in steam flow. Therefore, there will generally be a definite lack of correlation between feedwater and steam flow following a sudden drop in load. The exact degree of the difference between these two flows will depend upon the particular type and setting of the feedwater controls. In some extreme cases, the feedwater flow after a sudden drop in load can actually exceed the feedwater flow at maximum design conditions. Thus, it is a safer practice to assume that the feedwater flow will not be reduced and to assume that the NPSH required will in turn correspond to at least its value under flow conditions preceding the drop in load.

In the interval, however, the pressure at the boiler-feed pump suction is reduced correspondingly. Unfortunately, until the suction piping has been completely voided of the feedwater it contained prior to the load reduction, its temperature and vapor pressure will not be reduced. As a consequence, the available NPSH will diminish and may become insufficient to provide adequate pump operation. In such a case, the pump will flash and serious damage may be incurred.

The factor that establishes the adequacy of an installation from the point of view of suction conditions after a load drop is the ratio between the direct-contact heater storage capacity and the suction piping volume. Based on a number of simplifying assumptions, a formula has been developed for the minimum value of this ratio:

$$\text{Minimum } \frac{Q_h}{Q_s} = \frac{h_{x0} - h_{c2}}{K_h H_x} \tag{1}$$

where Q_h = volume of feedwater in heater storage, gal (m^3)

Q_s = volume of feedwater in suction piping, gal (m^3)

h_{x0} = enthalpy of feedwater under initial conditions, Btu/lb (J/kg)

h_{c2} = enthalpy of condensate to heater under final conditions, Btu/lb (J/kg)

K_h = change in enthalpy with pressure at steam conditions prior to load reduction, Btu/lb · ft absolute pressure (J/kg · m) (Figure 8)

H_x = available excess $NPSH$ = $NPSH$ available − $NPSH$ required, ft (m)

This relationship is somewhat conservative and does not take into account the residence time of the condensate in the piping and the closed heaters between the condenser hot well and the direct-contact heater. A slightly less conservative formula that takes some account of this residence time is

$$\text{Minimum } \frac{Q_h}{Q_s} = \frac{h_{x0} - [(h_{c0} + h_{c2})/2]}{K_h H_x} \tag{2}$$

where h_{c0} = enthalpy of condensate to heater under initial conditions, Btu/lb (J/kg)

For example, let

Initial heater pressure = 153 lb/in² (10.5 bar)

Initial feedwater temperature = 360°F (182°C)

Initial feedwater enthalpy = 331.4 Btu/lb (770.8 kJ/kg)

Final condensate enthalpy = 82.95 Btu/lb (192.9 kJ/kg)

K_h (from Figure 8) = 0.22 Btu/lb/ft (1679 J/kg/m)

H_x (available excess $NPSH$) = 15 ft (4.57 m)

Then

in USCS units $$\text{Minimum } \frac{Q_h}{Q_s} = \frac{331.4 - 82.95}{0.22 \times 15} = 75.3$$

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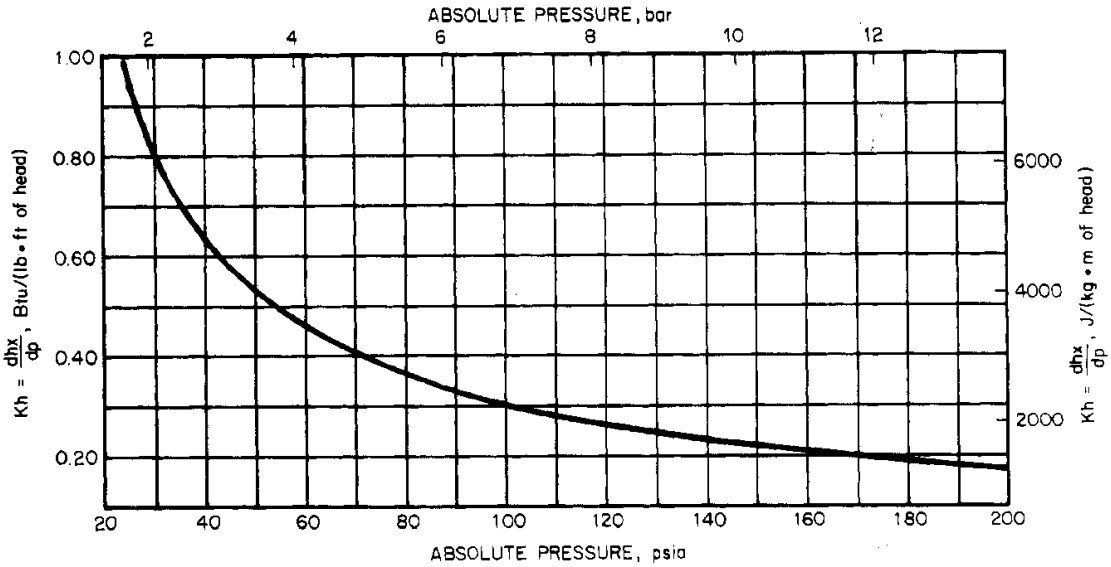


FIGURE 8 Enthalpy change with change of vapor pressure, for water

in SI units $$\text{Minimum } \frac{Q_h}{Q_s} = \frac{770{,}800 - 192{,}900}{1679 \times 4.57} = 75.3$$

This means that for safe operation after a sudden load drop in this particular ease, the heater storage volume must be at least 75.3 times the volume of the suction piping.

More complex and more rigorous calculations of the minimum ratio of heater storage volume to suction piping volume are provided in Reference 1.

Even where analysis indicates that the boiler-feed pumps are assured of their required NPSH during a reduction in turbine load, there is no guarantee that their operation will not be interrupted by flashing at some point in the suction piping. The criterion in determining the probability of flashing in the suction piping is to consider that the water that left the heater outlet at a saturated condition must pick up static pressure, by means of the vertical drop, at a rate at least equal to the pressure decay rate of the heater, or it will flash.

The most adverse conditions are those introduced by locating a horizontal run of piping too close to the heater outlet. A typical case is illustrated in Figure 9. (Because this example is used merely to illustrate the unfavorable effect of such a piping layout, the unit system used is immaterial and the example has been expressed in USCS units.) In the comparison of the two installations, we will stipulate that the total length and the sizes of the piping are the same for both arrangements and that the volumes of the suction piping between the heater outlet and points C and E of the two arrangements are the same. To simplify the comparison, the vertical distances between A and B, B and D, and A and E have been expressed in pounds per square inch instead of feet.

If a time interval x is selected such that water having left the heater outlet at the start of the transient conditions will have reached points C and E, respectively, at the end of the time interval, it becomes apparent that in the case illustrated on the left side of Figure 9, the pressure gain at point C is only 3 lb/in^2 by virtue of the vertical drop. Thus, x seconds after a pressure drop in the direct-contact heater from 52 to 48 lb/in^2 gage, the pressure at point C will be 51 lb/in^2, which is below the vapor pressure at the new temperature (296°F), and so flashing will occur. On the other hand, in the case of a straight vertical drop (right side of Figure 9), after the same time interval x the pressure will be 60 lb/in^2, which exceeds the vapor pressure, and so no flashing will occur. Formulas 1 and 2 can be used to determine the adequacy of the piping layout by selecting the critical point in the piping (in this case, point C) and substituting in the formulas so Q_s = volume in suction piping to point C and H_x = static head to point C less frictional losses to point C.

In the event that circumstances do not permit the provision of sufficient NPSH margin to provide adequate protection to the boiler-feed pumps during a sudden

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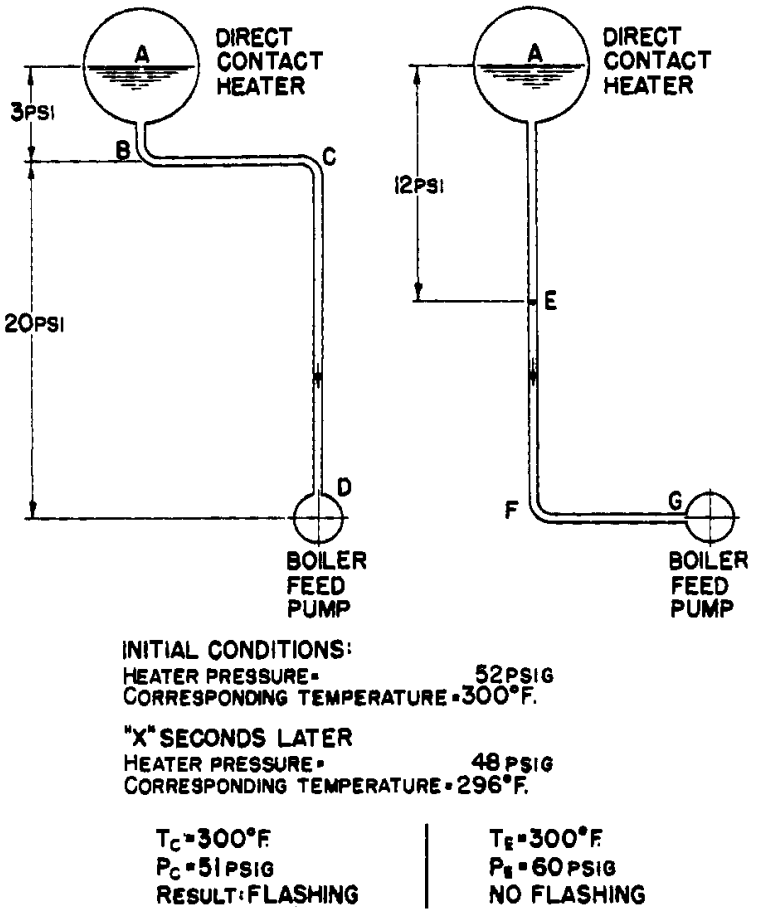


FIGURE 9 Comparison of suction piping arrangements

turbine load reduction, two alternate means are available to compensate for these circumstances:

1. A small amount of steam from the boiler can be admitted to the direct-contact heater through a pressure-reducing valve, to reduce the rate of pressure decay in the heater.
2. A small amount of cold condensate from the discharge of the condensate pumps can be made to bypass all or some of the closed heaters and be injected at the boiler-feed pump suction to subcool the feedwater, thus providing additional NPSH margin during load reduction.

Figure 10 illustrates the effect of subcooling (or temperature depression) on the available NPSH at various initial feedwater temperatures. Figure 11 shows, for varying ratios of injection flows, the temperature depression resulting from cold water injection plotted against the difference in temperature between the feedwater and the injection stream. For instance, if it were desired to provide 20 ft (6.1 m) additional NPSH to a boiler-feed pump that handles 325°F (163°C) water, the required temperature depression is 6°F (3.3°C). If the injection water temperature is 190°F (87.8°C), the difference between feedwater and injection water temperature is 135°F (75.2°C). From Figure 11, we can see that the injection flow must be 4.5% of the total feedwater flow.

An analysis of the relative merits of the two methods of protecting boiler-feed pumps against the unfavorable effects of transient conditions is presented in Reference 2. Either corrective action can be initiated automatically. This involves constant monitoring of the suction pressure and of the vapor pressure of the feedwater at the pump suction. The difference between the two is then constantly compared with a pre-established minimum NPSH. Any transient condition that causes the available NPSH to fall below this desired

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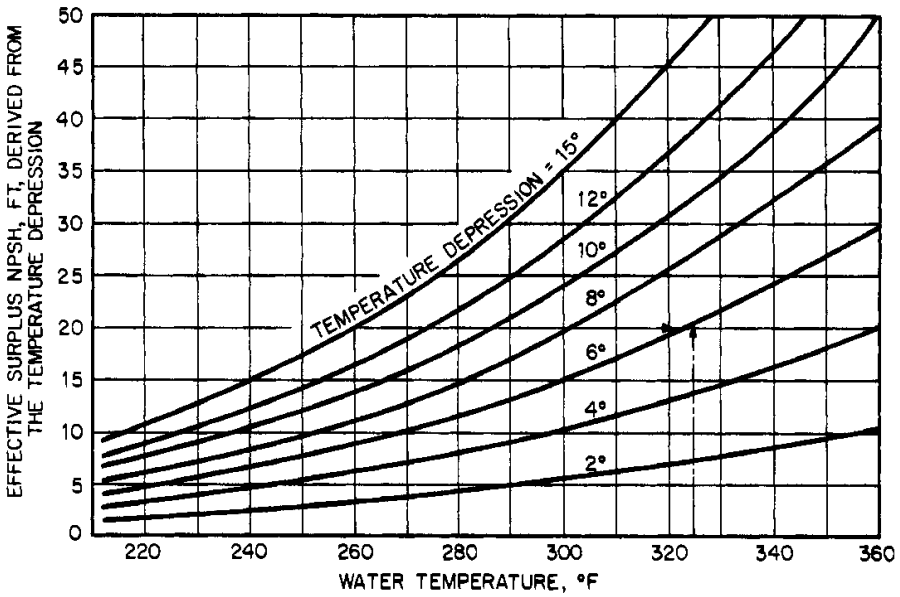


FIGURE 10 Effect of subcooling on available NPSH at various initial water temperatures [°C = (°F [min] 32)5/9; 1 ft = 0.3048 m]

minimum initiates corrective action, be it admission of cold condensate at the pump suction or admission of auxiliary steam to the direct-contact heater.

Another transient condition that will create a two-phase (steam-water mixture) flow at the pump suction—and in the pump—may occur during a "hot restart." When the plant experiences a trip, the pump is secured and the pressure in the deaerator drops. The temperature in the deaerator consequently drops within a relatively short period of time. The feedwater temperature has dropped from perhaps 350°F (175°C) to 250°F (120°C). The pump and suction piping near the pump remain at a higher temperature due to the mass of the metal (Figure 12). In a short time, the idle pump is switched on, and a two-phase flow condition occurs at the pump suction. Potential failure mode effects include suction cavitation, rotor upset and contact with stationary wear rings, and water hammer.

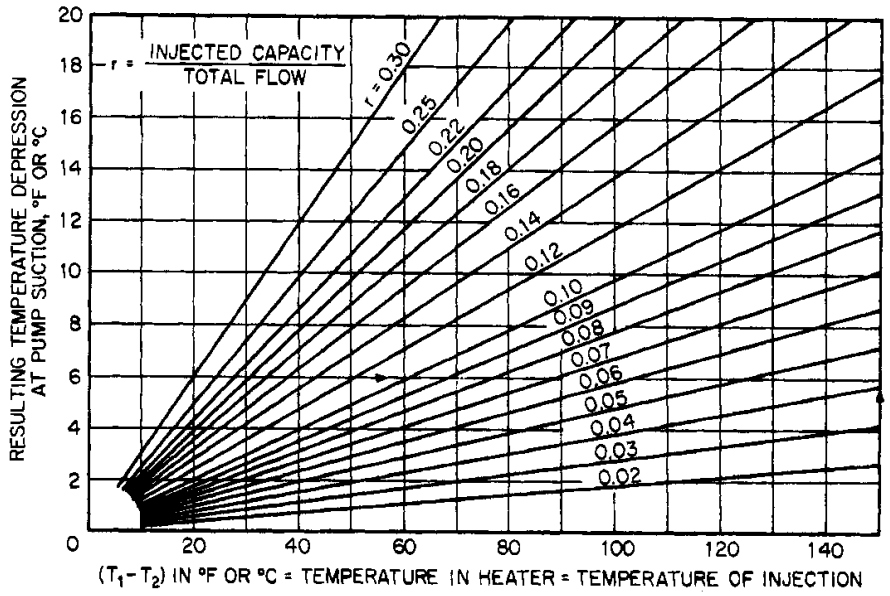


FIGURE 11 Required amount of cold water injection for a given temperature depression

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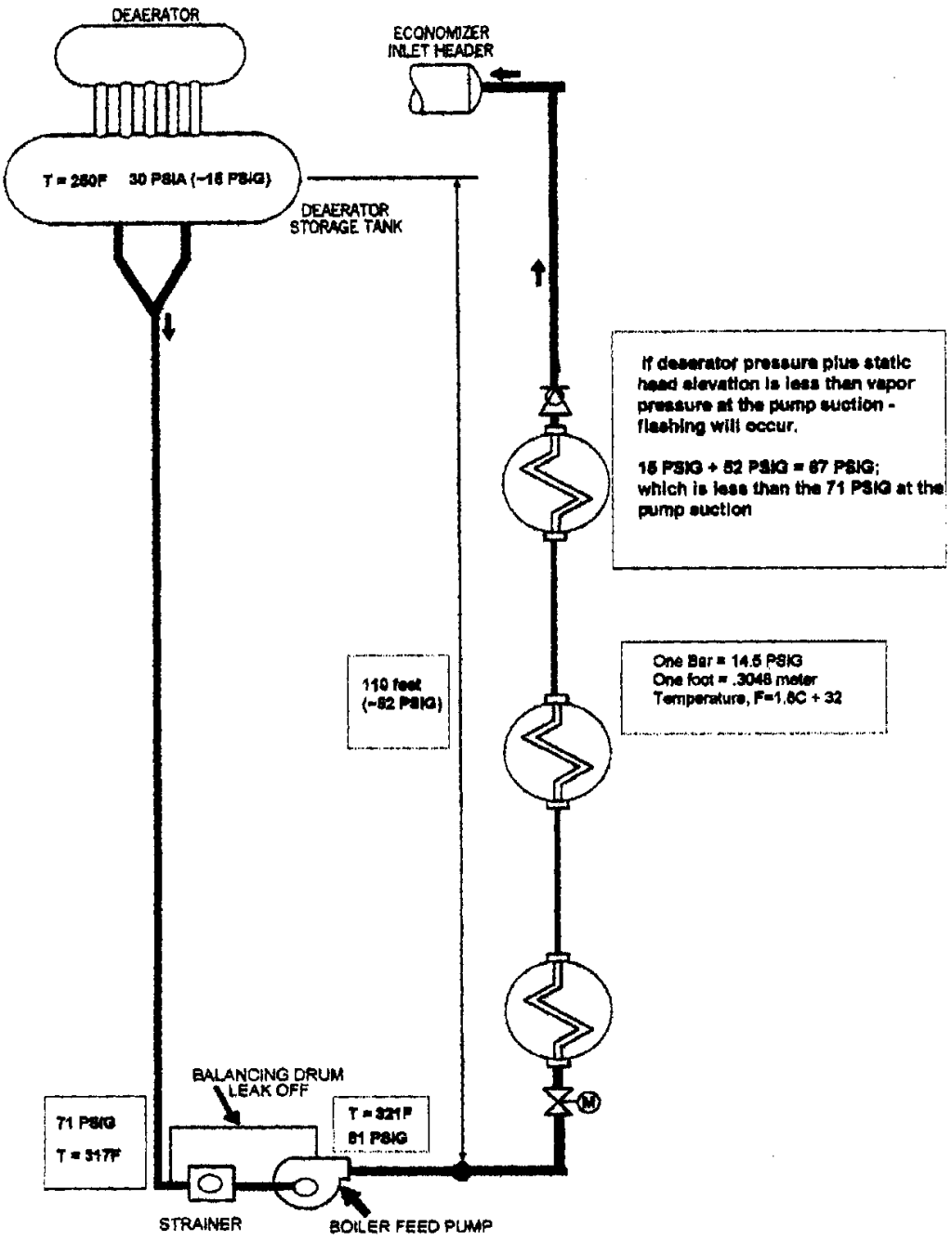


FIGURE 12 Feedwater system daigram, "hot restart" transient

BOOSTER PUMPS

Main boiler-feed pumps in central power plants are typically multistage pumps (three or more stages) operating at speeds between 5000 and 6000 rpm in order to delivery the required pressure to the boiler drum inlet. These large high-energy pumps need large values of NPSH required and benefit from large margins of NPSH available. In many cases, raising the elevation of direct-contact heaters high enough to provide sufficient NPSH available is costly and even impractical. Therefore, low-speed booster pumps are often utilized to maintain reasonable deaerator elevations and to provide sufficient NPSH available to the main boiler-feed pumps to avoid problems associated with cavitation.

Booster pumps for boiler-feed service are one- or two-stage pumps operating at lower speeds than the main pumps. They usually have double-suction first stage designs in order

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to minimize their NPSH required and thereby the height of the deaerator. Booster pumps can be driven through gearboxes from the same drivers as main boiler-feed pumps, or they can have separate electric motor drivers (Figure 42).

Discharge Pressure and Total Head The discharge pressure is the sum of the maximum boiler drum pressure and the frictional and control losses between the boiler-feed pump and the boiler drum inlet. The required discharge pressure will generally vary from 115 to 125% of the boiler drum pressure. The net pressure to be generated by the boiler-feed pump is the difference between the required discharge pressure and the available suction pressure. This must be converted to a total head, using the formula

$$\text{in USCS units} \qquad \text{Total head, ft} = \frac{\text{net pressure, lb/in}^2 \times 2.31}{\text{sp. gr.}}$$

$$\text{in SI units} \qquad \text{Total head, m} = \frac{\text{net pressure, bar} \times 10.2}{\text{sp. gr.}}$$

Slope of the Head-Capacity Curve The slope of a boiler-feed pump head-capacity curve is important, because the intersection of pump curve and the system curve affects system control and stability. Greater pump curve slope yields greater angles of intersection with the system curve, which are less sensitive to dynamic pressure variants and thereby inherently more stable. Greater curve slope and intersection angles also maintain better load-sharing between multiple pumps operating in parallel on the same system, as one pump wears ahead of another.

All boiler-feed pumps should have sloping curve shapes with continuously rising head with decreasing flow, within their expected flow range of operation. Rise-to-shutoff (RTSO) is a simple parameter used to evaluate the relative slope of various pump head-capacity curves. From a pump design standpoint, RTSO refers to the ratio of the pump head under closed-valve condition over the pump head at the best-efficiency-point (BEP). From a system standpoint, RTSO applies to the ratio of the pump head under closed-valve condition over the rated pump head at full unit load. Therefore, it's important that the rated flow point by close to the BEP of pump and that the pump have sufficient RTSO for stable operation.

RTSO values vary from pump design to pump design, but are somewhat dependent on pump specific speed (Ns). Designs with Ns ranging from 1000 to 1400 are often used in smaller multistage pumps with constant-speed motor drivers, because these pumps benefit from designs with high heads per stage at relatively low flow. Unfortunately, these designs can have low RTSO, which can cause system instability with boiler-feed pumps operating back to the left on their curve. Pumps with constant-speed drivers operate further back on their curves than pumps with variable-speed drivers. Therefore, attention to RTSO values is particularly important in these circumstances.

Designs with Ns ranging from 1600 to 2000 are often used for large boiler-feed pumps in large steam power plants with nuclear reactors, because these pumps have conditions of service with high flows and relatively low heads. Unfortunately, these designs can have discontinuities in their head-capacities curves in which a head depression occurs at partial flow conditions (Figures 13 and 14). Pumps with this characteristic can be unstable at lower flows and should not be operated near the head depression.

While generous RTSO benefits system stability, it is important to recognize that excessive RTSO increases the design pressures of pump casings and system components from the pump discharge and to the regulator valve. For a continuously rising head-capacity curve, the pump shutoff condition achieves the highest pump discharge pressure and therefore defines vessel and piping working pressures. A typical specified range for RTSO for boiler-feed pumps is not less than 120% and not more than 135%.

Control of Boiler-Feed Pumps Boiler-feed pumps operating at constant speed are controlled by feedwater regulator valves, which throttle the difference between pump head output and boiler pressure requirements. As unit loads decrease, the amount of throttling

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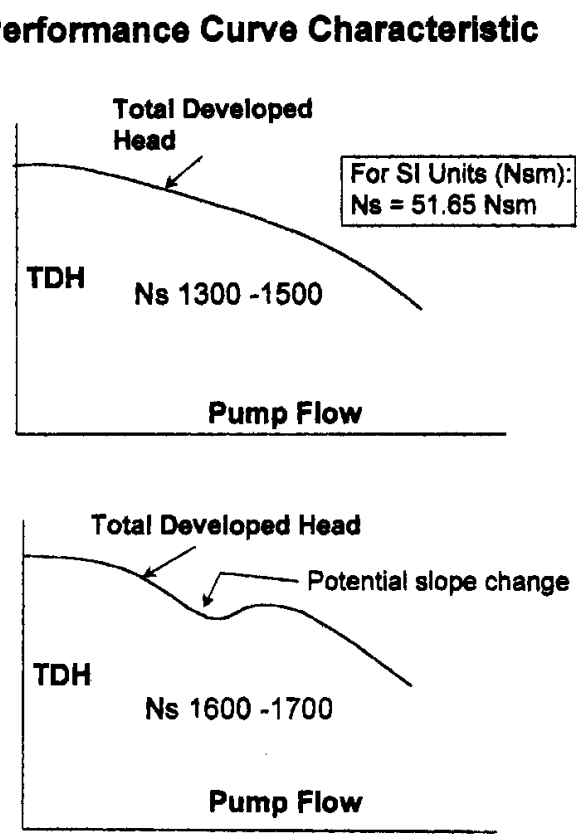


FIGURE 13 Pump performance curve characteristic—specific speed versus stability (Universal specific speed $\Omega_s = N_s/2733 \cdot N_q$ (in rpm, m^3/s, m) = $N_s/51.65$)

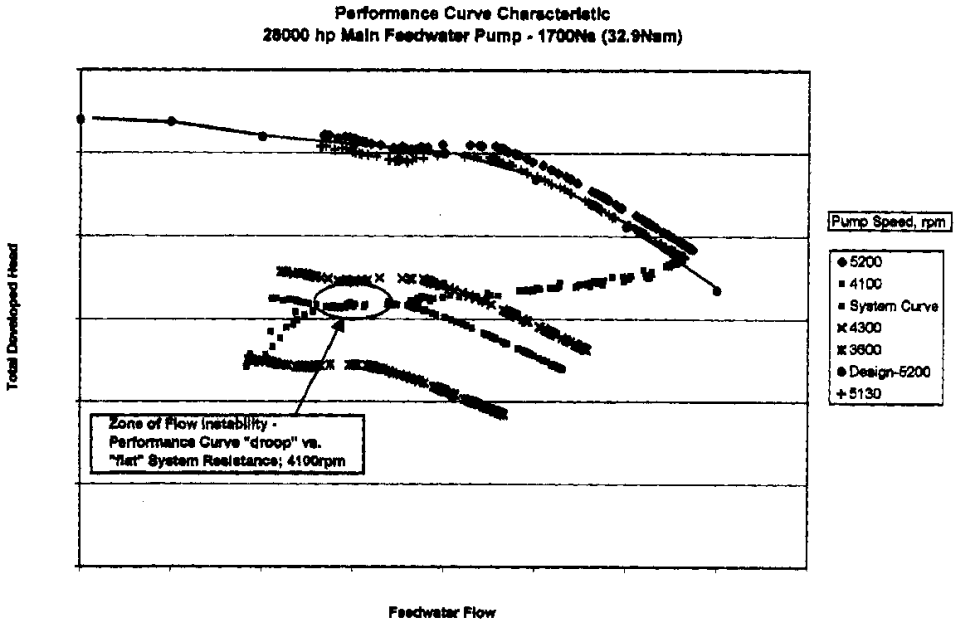


FIGURE 14 Performance characteristic—28,000 hp boilder feed pump (N_s – 1700) (Universal specific speed $\Omega_s = N_s/2733 \cdot N_q$ (in rpm, m^3/s, m) = $N_s/51.65$)

must increase because of the pump RTSO characteristic. Thus, throttle control of constant speed boiler-feed pumps wastes a portion of the energy of that pump. Also in large steam power plants, continuous throttling of large high-energy boiler-feed pumps can be damaging to the pumps and to the regulator valves. For these reasons, boiler-feed pumps with constant-speed driver are typically limited to 9000 BHP (6700 kW).

Large, high-energy pumps are best controlled by varying their speed. In some of these cases, feedwater regulator values are provided to throttle the pumps during low speed operation during plant starting. However, once the plant is at partial load, the feedwater delivery to the boiler is regulating by varying the pump driver speed. Typically, the boiler pressure curve for a large plant follows the affinity laws of the pump quite closely. Therefore, variable speed control keeps the pump running near its BEP, which maximizes machinery life and conserves energy.

Driver Power A single boiler-feed pump sized to deliver 100% capacity to a boiler or an HRGS generally will not operate at a capacity beyond the specified conditions of service. This is because conditions of service, which include safety margins, correspond to the largest operating capacity for the boiler or HRGS. In other words, a single 100% pump operates on a system head-capacity curve made up of the boiler drum pressure plus frictional losses in the piping and design margins. No further capacity is possible, unless the feedwater regulator valve goes full open, which is not a normal operating condition. Therefore, driver power ratings for single boiler-feed pumps may be selected with only 5% margin above the power at the rated conditions of service.

When two pumps are operated in parallel, feeding a single boiler, the situation is somewhat different. If one of the pumps is taken off the line at part load, the remaining pump could easily operate at capacities in excess of its design because its head-capacity curve would intersect the system-head curve at a head lower than the design head (Figure 15). In such a case, it is necessary to determine the pump capacity at the intersection point; the power corresponding to this capacity will be the maximum expected. It is not always necessary to select a driver that will not be overloaded at any point on the boiler-feed pump operating curve. Although electric motors used on boiler-feed service generally have an overload capacity of 15%, it is usually the practice to reserve this overload capacity as a safety margin and to select a motor that will not be overloaded at the design capacity. Exceptions occur in the case of very large motors. For instance, if the pump brake horsepower is 3100, it is logical to apply a 3000-hp motor, which will be overloaded by about 3% rather than a considerably more expensive 3500-hp motor. Because steam turbines are not built in definite standard sizes but can be designed for any intermediate rating, they are generally selected with about 5% excess power over the maximum expected pump power.

General Structural Features Multistage boiler-feed pumps are designed in three basic pressure casing types: Axial-split case, radial-split segmental ring, and radial-split barrel. Each casing type has its advantages and its limitations.

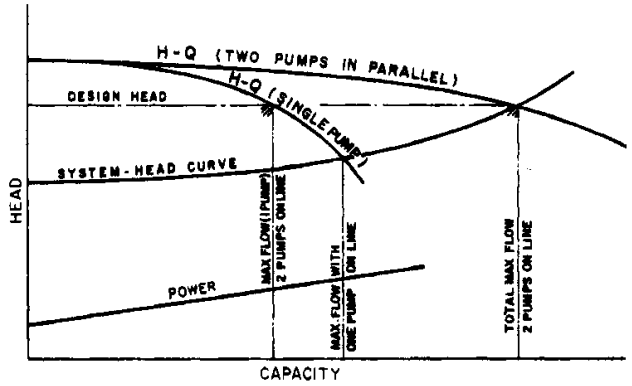


FIGURE 15 Method of determining maximum pump power for two boiler-feed pumps operating in parallel

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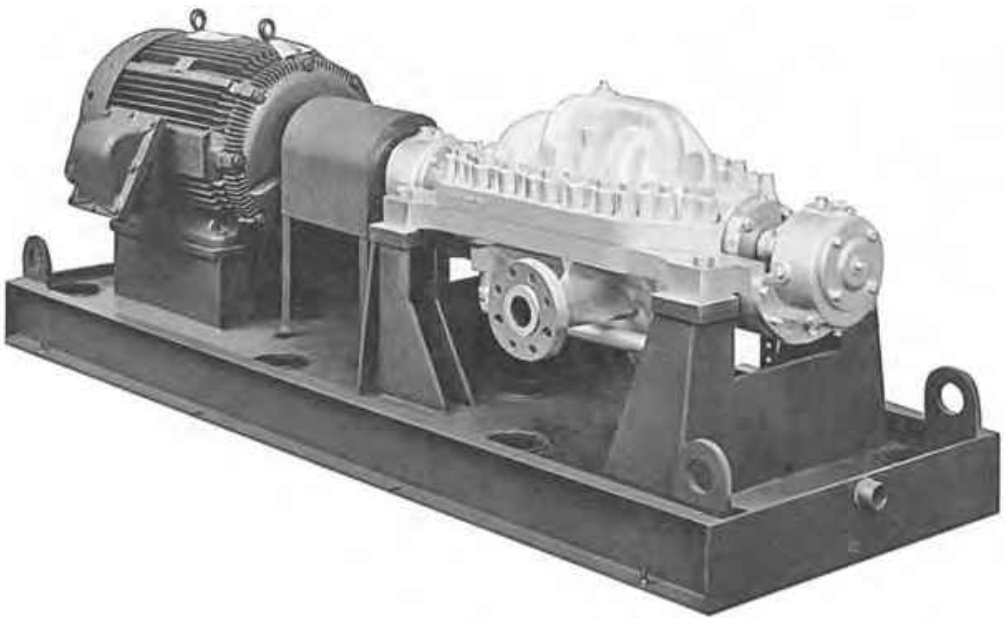


FIGURE 16 Axially split case multistage boiler feed pump, up to 3500 lb/in² (241 bar) (Flowserve Corporation)

Pumps with axial-split casings have two cast casing halves bolted together along a horizontal parting flange (Figure 16). Pumps with this casing type are easily inspected and serviced by removing the upper-half casing for access to the rotor without disturbing the piping. Pumps with axial-split cases require controlled prewarming and cooling for boiler-feed pump service, because of the complexity and wall thickness variations of the casings. Axial-split case designs are typically limited to design pressures of 2300 lb/in² (170 bar) in boiler-feed pumps.

Pumps with radial-split segmental ring casings have multiple stage casings clamped together between two head with tie-rods (Figure 17). Pumps with this casing type utilize minimal material with uniform thicknesses and are therefore capable of thermal transients hot or cold without warming provisions. They are troublesome to service, because any substantial work involve removal of the entire pump from the suction and discharge piping. Radial-split segmental ring casings nominally use O-rings at each joint for sealing. Therefore, boiler-feed pumps of this type are limited to approximately 350°F (177°C).

Pumps with radial-split barrel casings have double casings: an inner casing assembly made up of stage casings and an outer assembly made up of a barrel and a cover (Figure 18). The inner stage casing assembly contains the pumping elements (inner casings, diffusers, impellers, and so on). The outer barrel and cover assembly serves as the pump pressure vessel. Pumps with this casing type can be readily serviced without disturbing the piping, however once extracted, the inner assembly must be dismantled stage-by-stage for inspection and repair. Typically, the outer barrel and cover of a boiler-feed pump are steel forgings with uniform wall thicknesses, with confined gaskets at their radial joint. Therefore, radial-split barrel casings are capable of very high pressures and temperatures and are nominal used for high-energy boiler-feed pumps.

High-Speed, High-Pressure Boiler Feed Pumps Historically as fossil fuel-fired boiler pressures increased from 1800 lb/in² (120 bar) to 3000 lb/in² (200 bar) and eventually to 4500 lb/in² (310 bar), the total head required from main boiler-feed pumps increased from approximately 4000 ft (1220 m) to 7000 ft (2140 m) and 12,000 ft (3660 m). Initially during this transition, head increases were achieved by adding stages to pumps operating at two-pole motor speeds. Pumps became longer to accommodate the increased stages, which increased rotor deflections and lowered rotor critical speeds. These mechanical compromises interrupted the long life cycles of boiler-feed pumps, to which owners had became accustomed. The logical solution was to reduce the numbers of stages by increasing the head produced by each stage.

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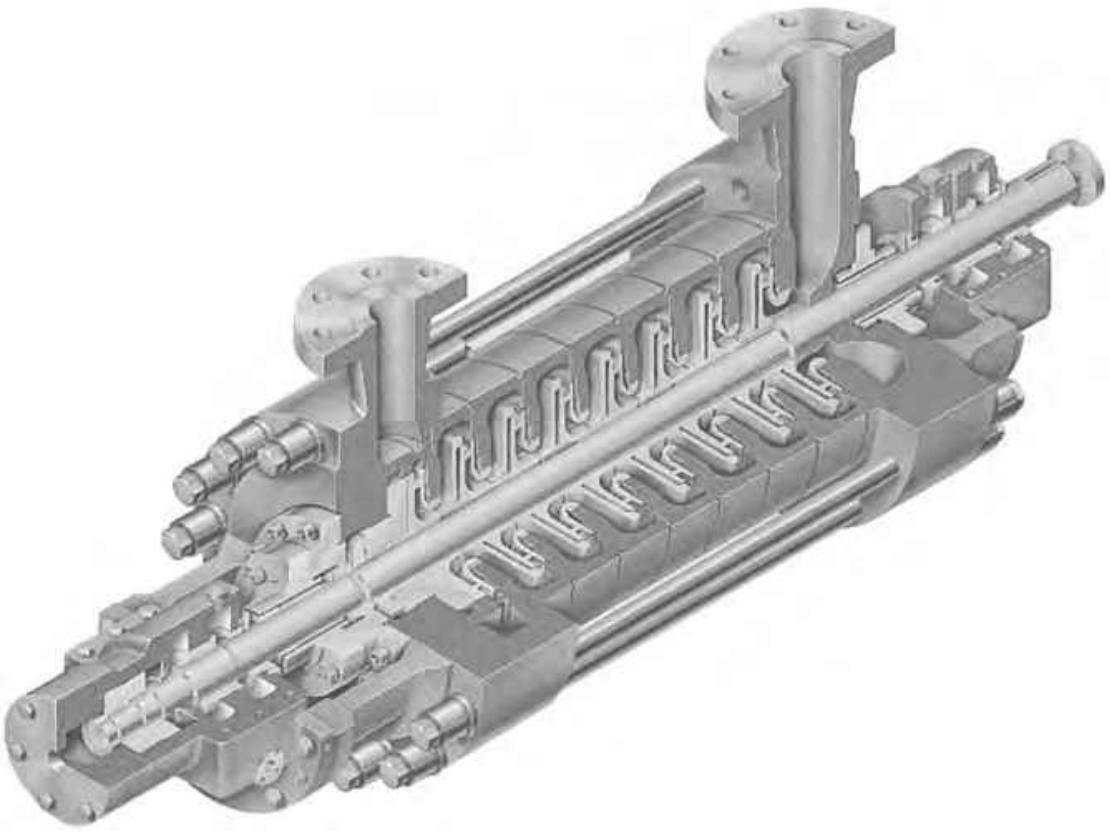


FIGURE 17 Radially split, segmental ring boiler feed pump (Flowserve Corporation)

In the 1970s, typical maximum head per stage values of main boiler-feed pumps increased from 800 ft (240 m) to 3000 ft (900 m). These increases were achieved by increasing rotating speeds from 3000 or 3600 rpm to 5000 to 6000 rpm, and even as high as 9000 rpm. As a result, high-speed high-energy pumps for boiler-feed service were established.

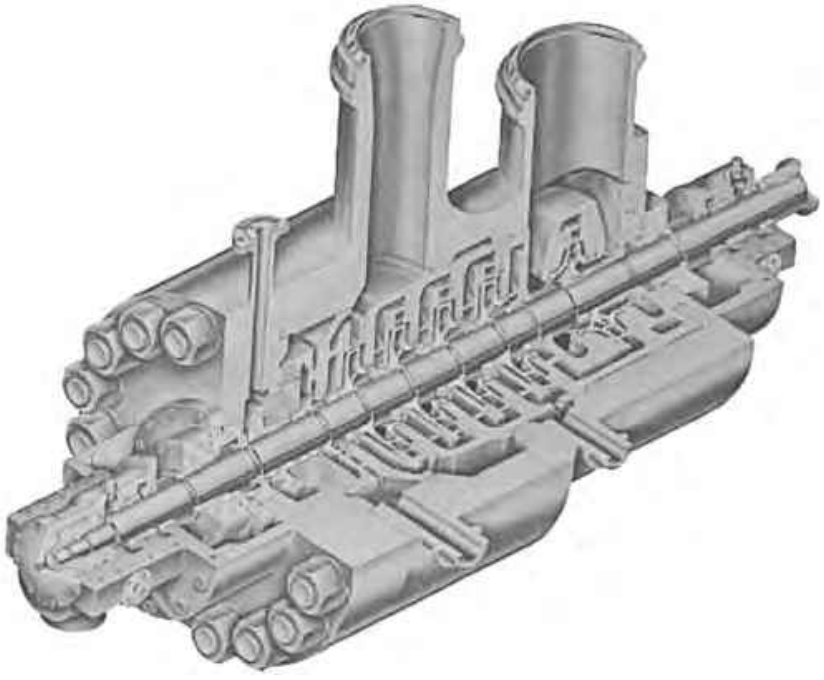


FIGURE 18 Radially split, double-case, barrel boiler feed pump (Flowserve Corporation)

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Subsequently during the 1980s, pump suppliers learned about the limitations of high-speed pumping particularly as it relates to high suction energy levels associate with high-speed pumps and the potential for cavitation erosion at first-stage impellers. Significant technological advances were made in new impeller vane designs and new impeller materials, which improve impeller life under these circumstances. Nevertheless, main boiler-feed pump installed in today's large fossil fuel-fired stream power plants typically have speeds between 5000 and 6000 rpm and heads produced per stage between 1800 ft (550 m) and 2400 ft (730 m). These design ranges with the latest pump technologies appear to provide the best balance of pump value, efficiency, and reliability.

Boiler-Feed Pump Drives The majority of boiler-feed pumps in small-size steam power plants, including most pumps in the steam cycles of combined-cycle power plants, are driven by electric motors at constant speeds. Furthermore, when the total head required in these plants is less than approximately than 5000 ft (1525 m), the pumps can reasonably operate at the motor speeds, either 3000 or 3600 rpm. Pumps required to produce greater total heads are driven at higher speeds using speed-increasing gearboxes with the electric motors. In some small plants where continuous process steam is critical (e.g., oil refinery-based steam power plants), multiple boiler-feed pumps are driven by both motors and steam turbines as protection against interruptions in electric power.

Boiler-feed pumps in large-size steam power plants are variable-speed driven with steam turbines, kinetic-type fluid couplings and electric motors, or with electric motors equipped with variable frequency drive controls. Variable-speed drives are typically used for boiler-feed pumps with power levels greater than 8000 hp (6000 kW) in order to match pump output with boiler demand and thereby achieve desired plant efficiencies and equipment life. Auxiliary steam turbines are by far the most common type of boiler-feed pump drives in these applications because:

1. Steam turbines provide variable-speed operation for matching pump output to boiler requirements as plant load varies.
2. Steam turbines provide ideal rotating speeds for multistage pumps with reasonable numbers of stages.
3. Steam turbines improve plant efficiency by utilizing exhaust steam from feedwater heaters.
4. Steam turbines reduce plant auxiliary electric power by eliminate large electric motors.
5. Steam turbines produce the high power levels needed for large high-energy boiler-feed pumps (Figure 19).

Operation of Boiler-Feed Pumps at Reduced Flows Operation of centrifugal pumps at shutoff or even at certain reduced flows can lead to very undesirable results. This subject is covered in detail in Sections 2.1.3 to 2.1.5, Sections 8.1 and 10, and Chap. 15, where methods for calculating minimum permissible flows and means for providing the necessary protection against operation below these flows are discussed.

Recent experiences have clearly defined the need to understand hydraulic instability, cavitation, and separation as they relate to off-design flow operation.

As deregulation and economic constraints dictate plant load cycling to match electricity demands and operating costs, the large central station boiler feed pumps experience significantly low operating flow. The low flow operation, high impeller suction specific speed, and high inlet tip speeds result in mismatched flow angles, backflow recirculation, and severe suction impeller inlet cavitation damage. This low flow hydraulic instability will also result in damage to pump volute cutwaters and diffuser vanes (Figure 20). Suction impellers where the suction specific speed exceeds 10,000 and eyebore inlet tip speeds exceed 200 ft/sec. are highly susceptible to this low flow instability and component damage. The series of photos in Figure 21 show a condition typical for many high-energy pumps operating at flows below design levels. They demonstrate how serious low-flow instability can be when it is coupled with two-phase flow activity.

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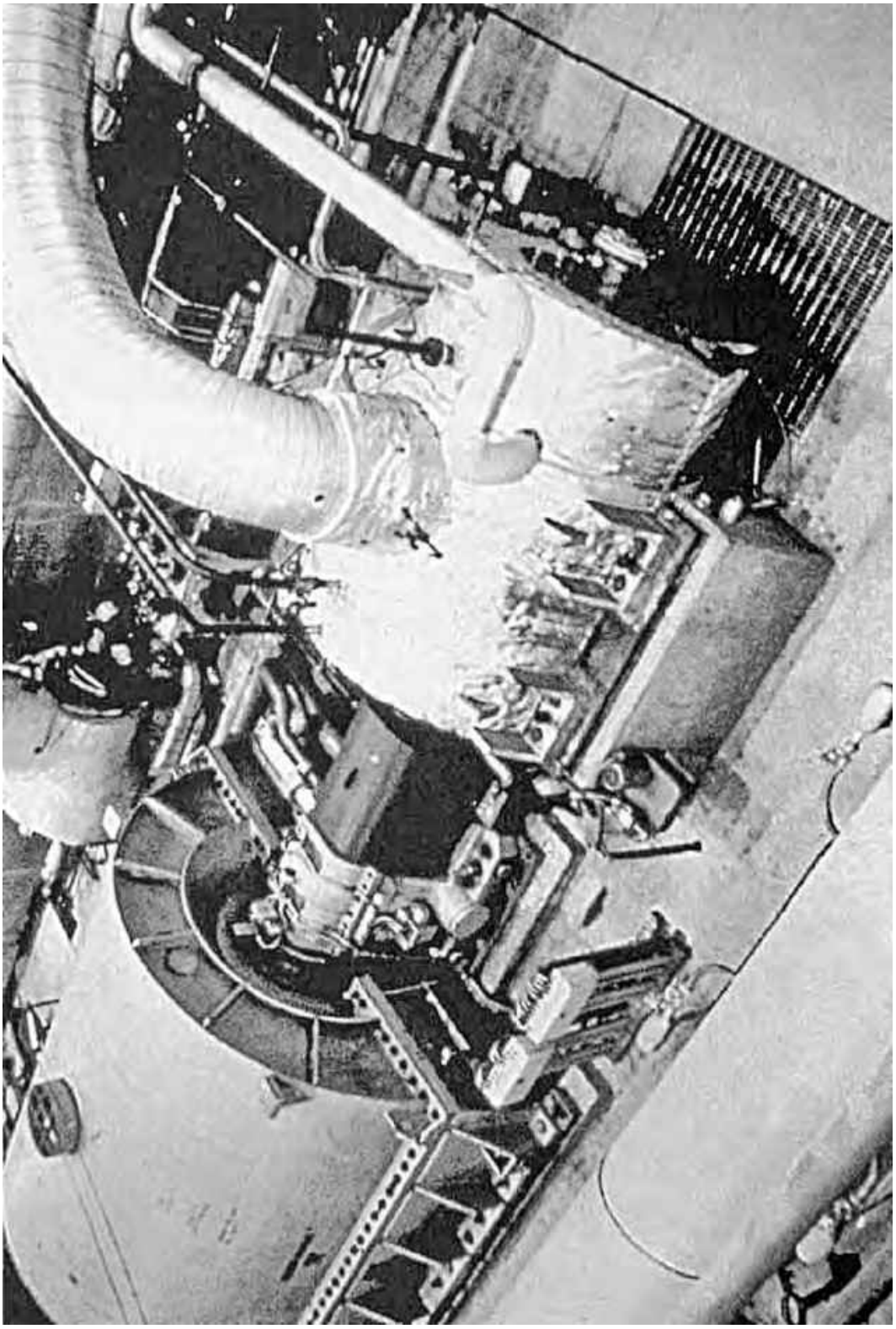


FIGURE 19 Installation of a single 65,000 horsepower (48,500 kW) boiler pump feed (Flowserve Corporation)

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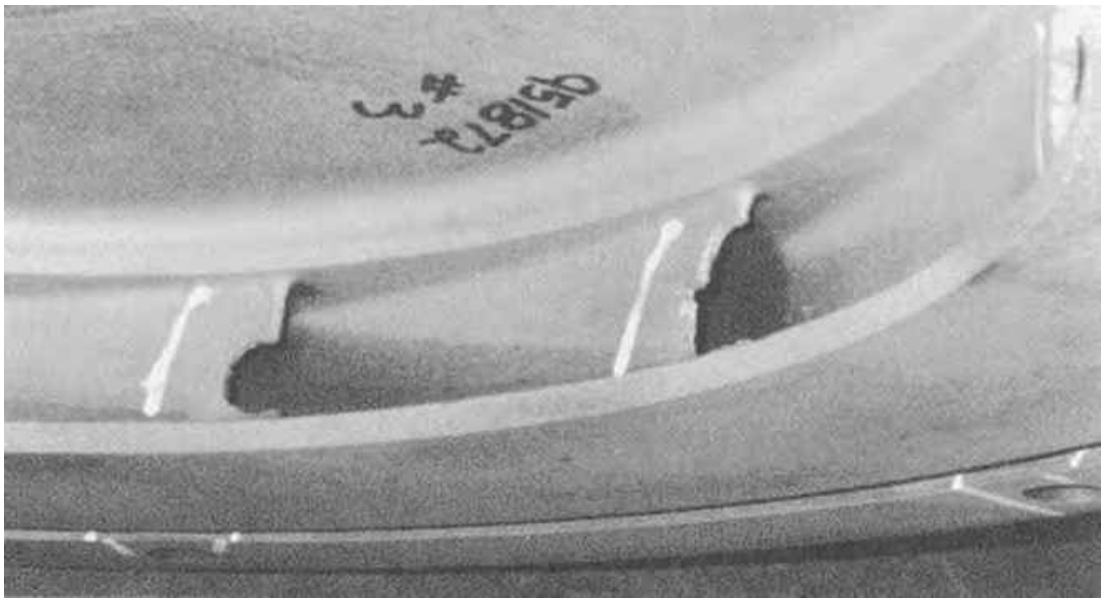


FIGURE 20 Diffuser inlet vane erosion damage (Flowserve Corporation)

The severity of cavitation erosion is highly dependent on the inlet tip speed of the suction stage impeller, the NPSHA, and the thermodynamic properties of the fluid being pumped. The erosion seen in Figure 22 was caused by the collapse of discrete cavitation vapor bubbles. Cavitation forms around an impeller blade because of local static pressure falling below the vapor pressure of the liquid being pumped. The vapor cavity shown in Figure 23 is an example of this phenomenon.

With decreasing flow rates (due to operating at off-design conditions), the fluid approaches the impeller blade with larger and larger angles of incidence altering the velocity and pressure fields inside the impeller.

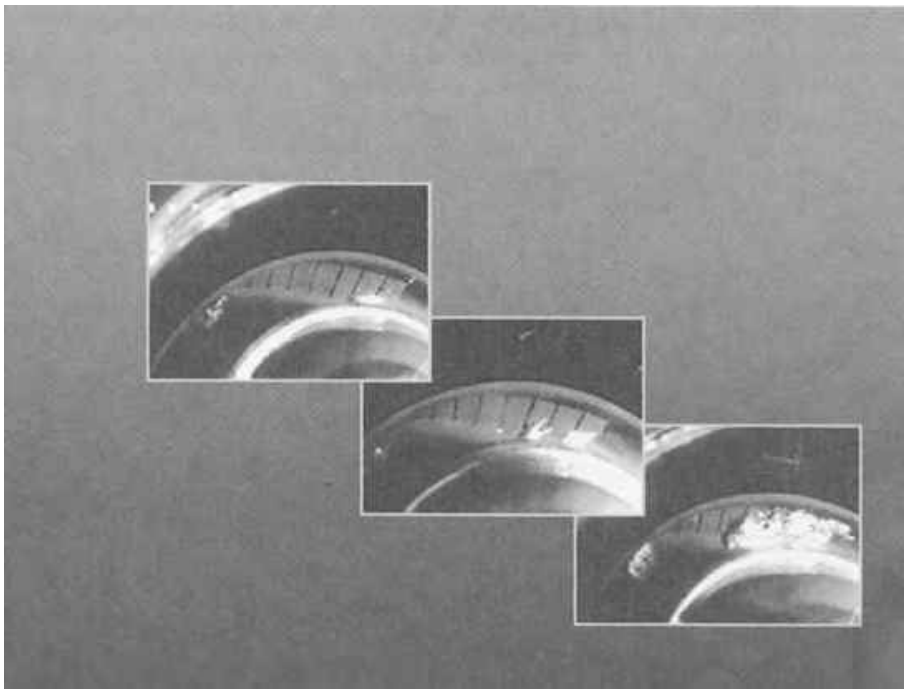


FIGURE 21 Unsteady vapor cavity behavior, feed pump at low flow (Flowserve Corporation)

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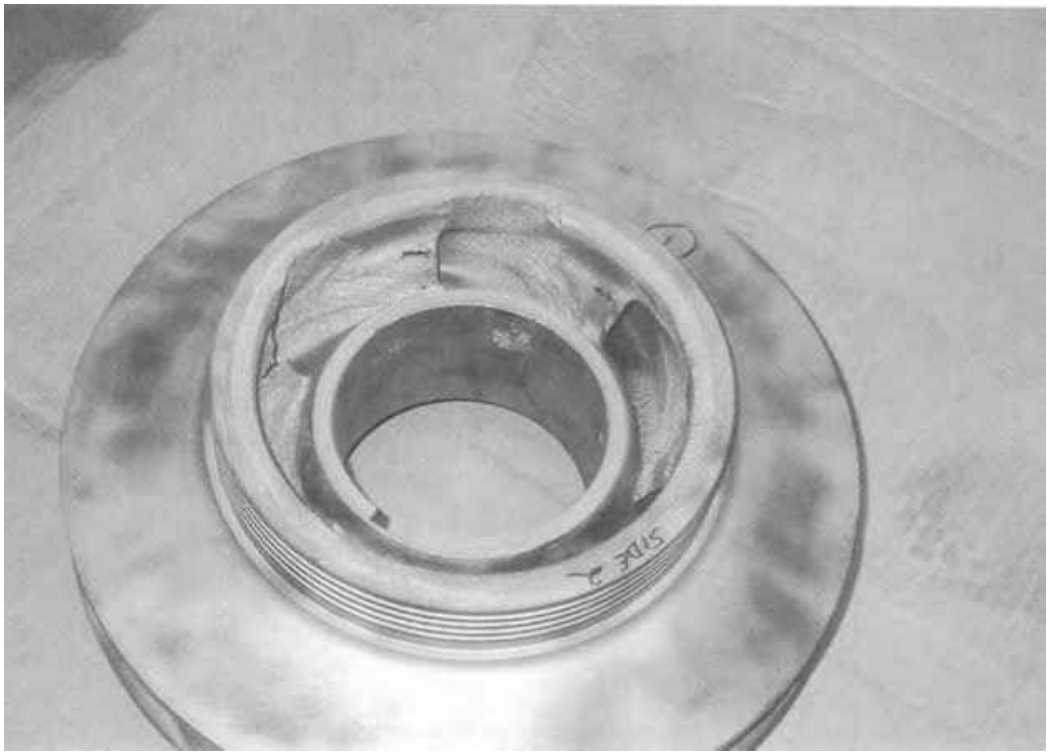


FIGURE 22 Impeller inlet vane cavitation erosion damage (Flowserve Corporation)

Pumps that are cycled between minimum-flow and flows in excess of the best efficiency point (BEP) create conditions at the impeller in excess of what "fixed geometry" machines can effectively tolerate. Impeller geometry has been shown to influence the degree and severity of cavitation problems experienced with high-energy pumps.

Through the 1980s, attempts were made to pursue "non-traditional" designs of impeller blading. These efforts took the form of profiling the inlet blade in a way that rapidly increased and then decreased the blade thickness.

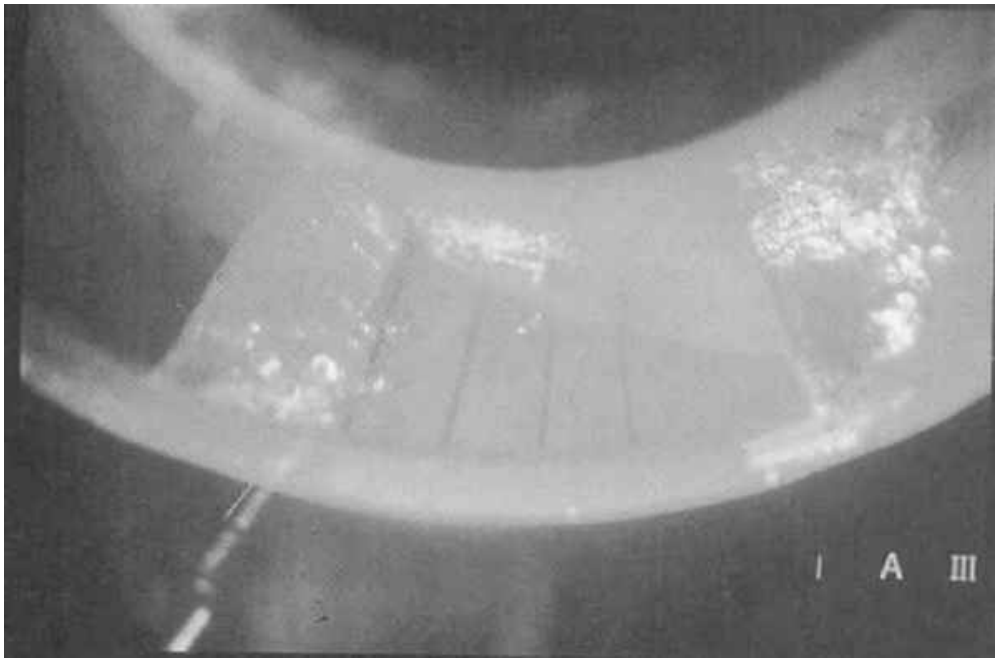


FIGURE 23 Cavitation vapor bubbles on suction surface of the impeller inlet vane (Flowserve Corporation)

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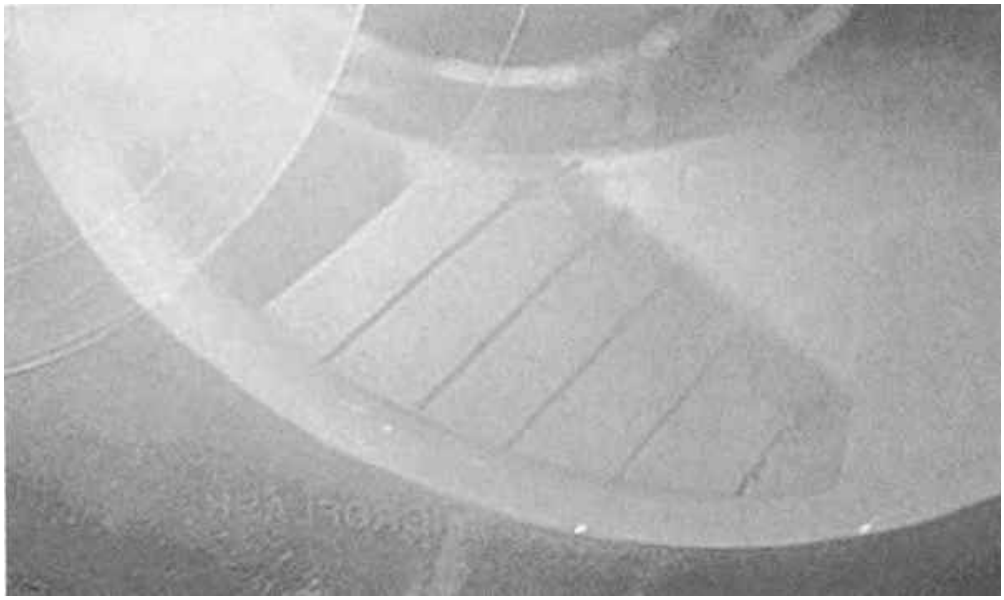


FIGURE 24 Operation at same flow as Figure 23 improved inlet vane shape—dramatic vapor bubble reduction (Flowserve Corporation)

A new impeller blade design approach, referred to as a "biased-wedge" design, has been found to provide a manufacturable configuration that enables cavitation bubble-free operation over a wide fluid flow range. This design approach is a result of extensive flow visualization test work and computational fluid flow analysis of many impeller geometries. It successfully advances the performance of high-energy pump suction stages to levels not achievable with conventional designs. Dramatic reduction in cavitation activity on the impeller was recorded as seen in the photo (Figure 24) of the suction surface of the final impeller taken at identical positions in the suction inlet and at the same operating conditions (baseload and minimum flow) as Figure 23. The inlet vane air foil shape has proven successful in facilitating feed pump flow rangeability.

Fundamentals for Successful Operating Life—Efficiency/Reliability Best practices for extended successful operating life of pumps are outlined in Chap. 15. Essential fundamentals to emphasize for boiler feed pumps are proper pump warm-up, standby warming, and shaft (fixed bushing) seal drain temperature control. These characteristics have become more critical as central station plants are cycled and large feed pumps are operated with varying loads and in standby modes. Current designs of multistage pumps (Figure 17) installed in combined cycle plants are less sensitive to thermal transients and wide swings in load (pump flow).

Pre-warming of the pump and maintaining warm-up flow to an idle pump to assure dimensional thermal uniformity is essential to maintenance of internal clearances, pump efficiency, and long life. This process is critical for multistage pumps to minimize thermal stratification within the pump. The distortion, including shaft bowing (Figure 25), will cause the following potential failure modes:

1. Flashing
2. Internal rubbing
3. Increased wear ring clearances
4. Pump seizure
5. Worn seal bushing clearance and excessive leakage
6. Loss of pump performance and efficiency
7. High pump vibration
8. Worn bearings/bearing clearances

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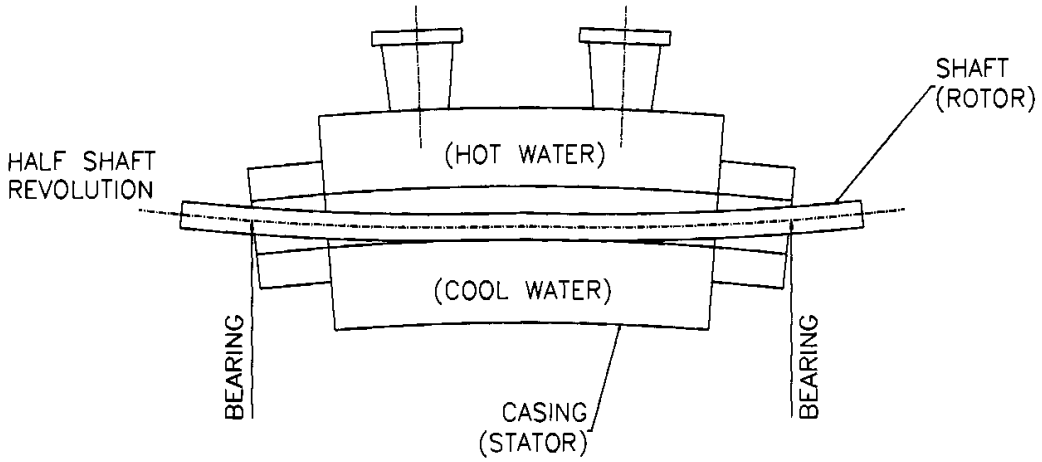


FIGURE 25 Thermal distortion of feed pump casing and shaft, due to improper warm-up and thermal stratification

Installation features and operating practices that extend pump life, efficiency, and reliability are

1. Proper pump insulation (Figure 26) at the casing and discharge head
2. Warm-up orifice, piped around the discharge check valve. Preference is to inject warm-up flow to the bottom of the pump casing to minimize short-circuiting of the hot feedwater and potential thermal stratification within the casing.
3. Maintaining shaft seal leakage drain temperature (Figure 27 between 150 and 170°F (65 and 77°C); utilize an electro-pneumatic temperature control system.

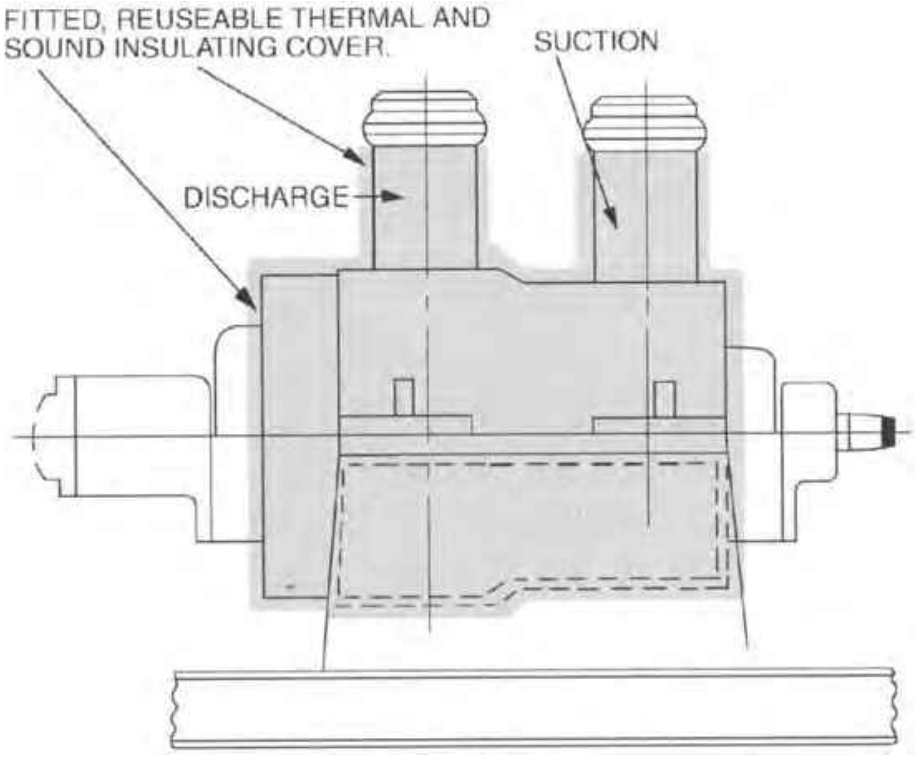


FIGURE 26 Recommended thermal insulation of a boiler feed pump (Flowserve Corporation)

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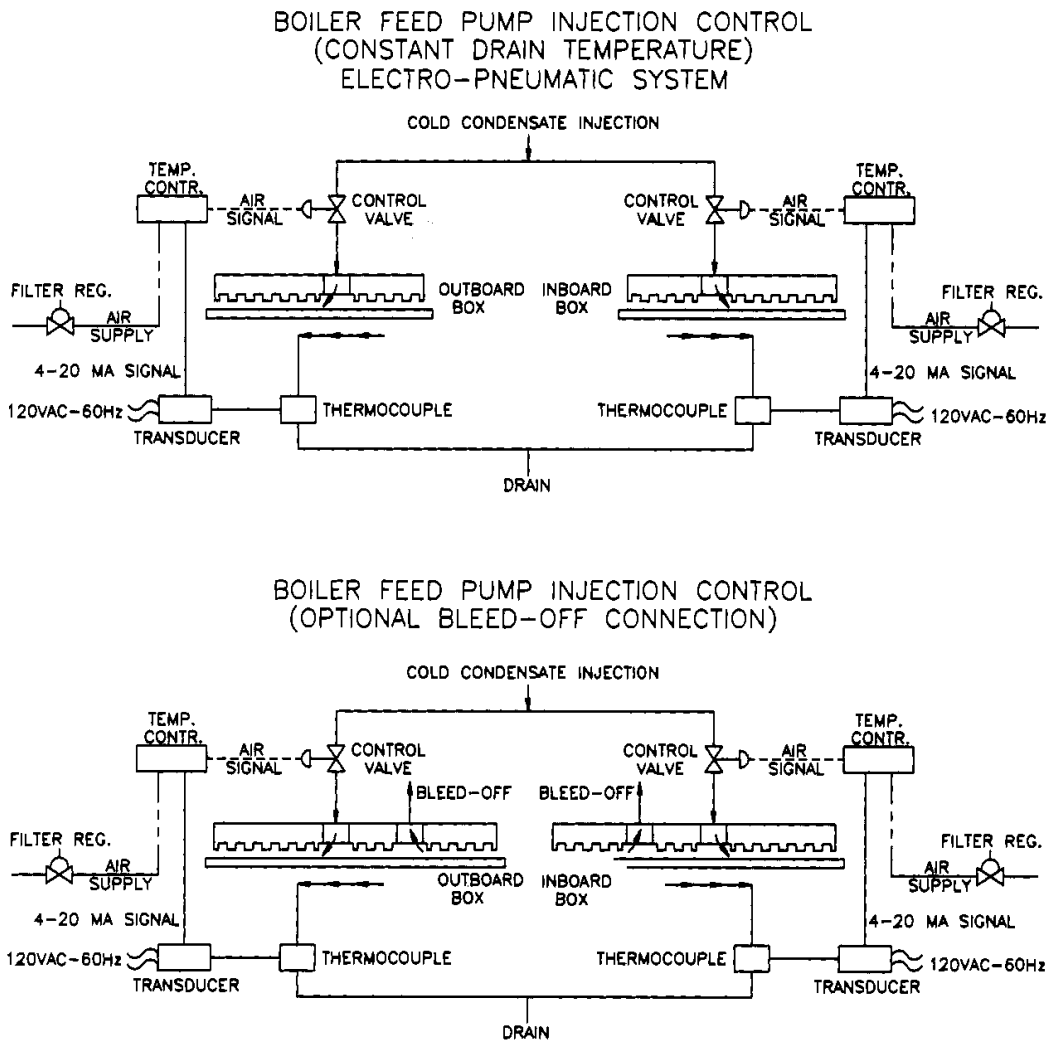


FIGURE 27 Boiler feed pump shaft seal injection/leakage control system; electro-pneumatic control of constant drain temperature

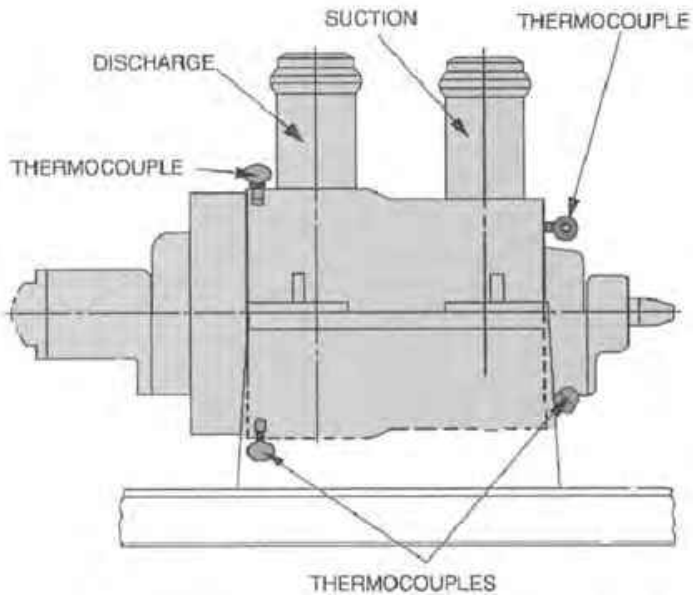


FIGURE 28 Feed pump casing and discharge head thermocouple installation, to monitor and control uniform temperature distribution (Flowserve Corporation)

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4. Installation of thermocouples or other temperature-detecting instruments (Figure 28) in the pump casing and discharge head to confirm temperature differences within 50°F (28°C) across the pump and relative to the feedwater temperature.
5. Assurance of proper functioning of the pump casing "pin" and "key" block to allow uniform thermal growth. Confirm that the hold-down bolts for the outboard casing feet are not over-torqued, preventing uniform axial thermal growth as the pump is heated.
6. Assurance of proper location and functioning of critical pipe hangers to minimize pipe strain on the pump suction and discharge nozzles.

Nuclear Power Plants In oversimplified form, the nuclear energy steam power plant differs from the conventional power plant only in that it uses a different fuel. Thus what is called the secondary cycle (consisting of turbogenerator, condenser and auxiliaries, and boiler-feed pumps) is not very different from its counterpart in the conventional steam power plant. The main differences are a desire for even greater equipment reliability and a preference for an absence or minimum of leakage to avoid any possibility of contamination with radioactive material. One other difference distinguishes most nuclear power plants today from their fossil fuel counterparts: their operating steam pressures and temperatures are much lower. Consequently, in most cases, reactor feed pumps are single-stage pumps; a typical section is shown in Figure 29. The lower operating conditions result in higher heat rates, and the flows—both of the feedwater and of the condenser circulation—are about one-third higher than for fossil fuel power plants of equal megawatt rating.

More detailed information on other nuclear power plant pumping services is given in Section 12.6.1.

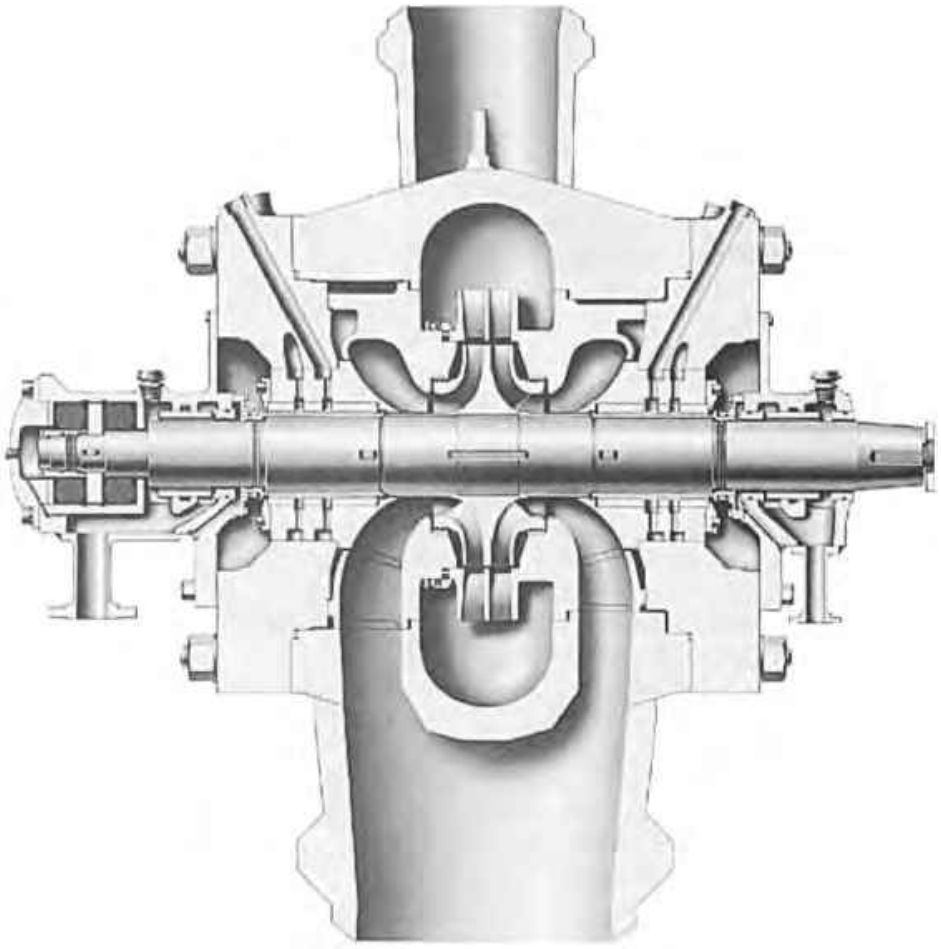


FIGURE 29 Single stage double suction reactor feed pump, 12,000 horsepower (8950 KW) (Flowserve Corporation)

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CONDENSATE PUMPS

Condensate pumps take their suction from the condenser hot well and discharge either to the deaerating heater in open feedwater systems (refer to Figure 5) or to the suction of the boiler-feed pumps in closed systems (refer to Figure 6). These pumps, therefore, operate with a very low pressure at their suction. The available NPSH is obtained by the submergence between the water level in the condenser hot well and the centerline of the condensate pump first-stage impeller. Because it is desirable to locate the condenser hot well as low as possible and avoid the use of a condensate pump pit, the available NPSH is generally extremely low, on the order of 2 to 4 ft (0.6 to 1.2 m). The exception to this occurs when vertical-can condensate pumps are used because these can be installed below ground and higher values of submergence can be obtained. Frictional losses on the suction side must be kept to an absolute minimum. The piping connection from the hot well to the pump should therefore be as direct as possible and of ample size and should have a minimum of fittings.

Because of the low available NPSH, condensate pumps operate at relatively low speeds, ranging from 1750 rpm in the low range of capacities to 880 rpm.

It is customary to provide a liberal excess capacity margin above the full-load steam condensing flow to take care of the heater drains that may be dumped into the condenser hot well if the heater drain pumps are taken out of service for any reason.

Types of Condensate Pumps Historically, horizontal multistage pumps with complex split-case casing configurations and double-suction-type impellers were used for condensate pump service (Figure 32). These were large low-speed pumps with limited head capability, because of the limited NSPH available of condensate extraction applications.

Modern steam power plants use vertical can-type multistage pumps (Figure 30). The length of these pumps provides adequate NPSH available at the first-stage impeller and their multistage configuration produce the required heads and flows required of any condensate pump service. Vertical can-type condensate pumps are available with either single-suction or double-suction impellers at the first stage position (Figure 31). For a given flow and speed, double-suction impellers need less NPSH required than suction-suction impellers. Therefore, double-section impellers reduce the length of the pump assembly and suction can, because the NPSH available at the first-stage position is strictly a function of that static elevation. However, high-flow condensate pumps benefit more from the advantages of double-suction impellers than low-flow pumps, because their NPSH required and costs of installation are greater.

Condensate pumps are located very close to the condenser hot well, and the suction piping is generally so short that the frictional losses in this piping are not significant. However, strainers are occasionally installed in this piping, and a great deal of attention must be paid to the frictional losses through them and to their frequent cleaning. Cases have been reported on occasion where the pressure drop across these strainers was sufficient to cause flashing at the suction nozzle of the condensate pumps.

Vertical can-type condensate pumps are driven by electric motors. Operating speeds are typically fixed at nominally 1500 or 1800 rpm. Larger pumps with flows greater than approximately 5000 gpm have motors with nominal speeds of 1000 or 1200 rpm.

To prevent air leakage at the stuffing boxes, condensate pumps equipped with packing are always provided with seal cages. The water used for gland sealing must be taken from the condensate pump discharge manifold beyond all the check valves. Pumps fitted with mechanical seals also use injection of condensate at a pressure higher than atmospheric pressure to prevent air from being drawn into the pump and feedwater system across the seal faces.

Condensate pumps are typically supplied with cast iron casings or bowls. Impellers are typically stainless steel or chrome steel materials. Carbon or other nonmetallic bearings are now used in place of bronze materials, because of potential steam turbine blade sensitivity to any copper in the flow stream.

Condensate Pump Regulation When condensate pumps operate in a closed cycle ahead of boiler-feed pumps, the two sets of pumps in series combine insofar of the system curve is concerned. With this arrangement, variation in flow through both sets of pumps

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FIGURE 30 Vertical condensate or heater drain pump (Flowserve Corporation)



FIGURE 31 Vertical turbine condensate pump (Flowserve Corporation)

is regulated by throttling the feedwater regulating valve or by varying the speed of the boiler-feed pump.

In an open cycle with a deaerator heater, the condensate pumps must be regulated separately from the boiler-feed pumps. This can be accomplished several methods:

1. Limit the NPSH available to the condensate pump such that it breaks at higher flows (Figure 32).
2. Throttle the discharge of the condensate pumps using deaerator level control.
3. Bypass excess flow from the condensate back to the hot well.
4. Varying the speed of the condensate pumps.

Method 1 requires the condensate pumps operate with significant amounts cavitation at all flow rates. Historically, specialized pumps worked without severe damage under

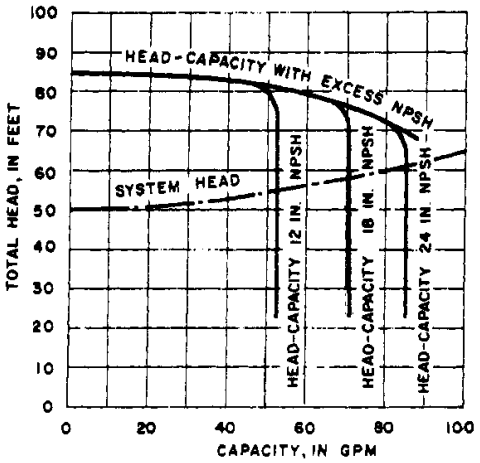


FIGURE 32 Characteristic of a condensate pump operating on a submergence-controlled system

these circumstances, because their suction energy and the damage potential of saturated water were not substantial. However, today this method is rarely used, because it is not appropriate for vertical can-type pumps.

The condensate pump discharge can be throttled by a float control arranged to position a valve that increases the system-head curve as the level in the hot well is drawn down. This eliminates the cavitation in the condensate pump, but at the cost of a slight power increase. Furthermore, the float necessarily operates over a narrow range, and the mechanism tends to be somewhat sluggish in following rapid load changes, often resulting in capacity and pressure surges. This transient condition is often the root cause of failed thrust bearings and axial rotor shifting. Another critical piping arrangement feature is the discharge piping check valve. The check valve in every condensate pump must be below the condenser hotwell level to ensure prevention of air entrapment and start-up waterhammer. [Refer to Section 11.3.]

When condensate delivery is controlled through bypassing, the hot well float controls a valve in a bypass line connecting the pump discharge back to the hot well. At maximum condensate flow, the float is at its upper limit with the bypass closed and all the condensate is delivered to the system. As the condensate flow to the hot well decreases, the hot well level falls, carrying the float down and opening the bypass. Sluggish float action can create the same problems of system instability in bypass control as in throttling control, however, and the power consumption is excessive because the pump always operates at full capacity.

A combination of throttling and bypassing control eliminates the shortcoming of excessive power consumption. The minimum flow at which bypassing begins is selected to provide sufficient flow through the ejector condenser.

A modification of the bypassing control for minimum flow is illustrated in Figure 33, which shows a thermostatic control for condensate recirculation. With practically constant steam flow through the ejector, the rise in condensate temperature between the inlet and outlet of the ejector condenser is a close indication of condensate flow rate through the ejector condenser tubes. Therefore an automatic device to regulate the condensate flow rate can be controlled by this temperature differential. A small pipe is connected from the condensate outlet on the ejector condenser back into the main condenser shell. An automatic valve is installed in this line and is actuated and controlled by the temperature rise of the condensate. Whenever the temperature rises to a certain predetermined figure, indicating a low flow of condensate, the automatic valve begins to open, allowing some of the condensate to return to the condenser and then to the condensate pump, which supplies it to the ejector at the increased rate. When the temperature rise through the ejector condenser is less than the limiting amount, indicating that ample condensate is flowing through the ejector condenser, the automatic valve remains closed.

As condensate flow design demand increased, the vertical multistage pumps were installed as two half-capacity pumps. Flow variation is accomplished by operating one or two pumps and by utilizing the regulating discharge valve.

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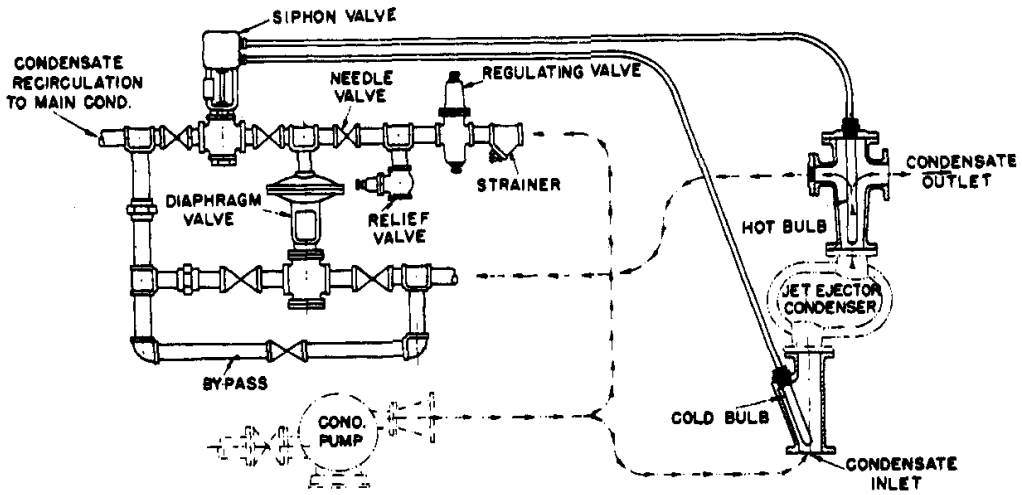


FIGURE 33 Thermostatic control for condensate recirculation

Today, most modern steam power plants use some combination of flow throttling and flow bypassing (methods 2 and 3 above) to control condensate pumps. When two or more pumps in parallel service are controlled in this manner with common control elements, emphasis must be placed on preventing zero-flow and low-flow operation of any pumps sharing the controls. Under these circumstances, a weaker pump can be "dead-headed" by a stronger pump and severe pump damage can be expected. In these cases, individual pumps should be monitored with instrumentation to confirm their operation in the allowable operating range.

Just as with boiler-feed pumps—or any high-energy pump for that matter—large high-energy condensate pumps should not be operated at reduced flows for prolonged periods of time. Such operation wastes significant pumping energy and risks pump damage. In these cases, variable speed control is a viable alternative. Application of variable speed control also reduces the magnitude of flow bypassed back to the hot well during reduced unit load operations.

HEATER DRAIN PUMPS

Service Conditions Condensate drains from closed heaters can be flashed to the steam space of a lower-pressure heater or pumped into the feedwater cycle at some higher-pressure point. Piping each heater drain to the heater having the next lower pressure is the simpler mechanical arrangement and requires no power-driven equipment. This "cascading" is accomplished by an appropriate trap in each heater drain. A series of heaters can thus be drained by cascading from heater to heater in the order of descending pressure, the lowest being drained directly to the condenser.

This arrangement, however, introduces a loss of heat because the heat content of the drains from the lowest-pressure heater is dissipated in the condenser by transfer to the circulating water. It is generally the practice, therefore, to cascade only down to the lowest-pressure heater and pump the drains from that heater back into the feedwater cycle, as shown in Figure 34. Because the pressure in that heater hot well is low (frequently below atmospheric even at full load), heater drain pumps on that service are commonly described as on "low-pressure heater drain service."

In an open cycle, drains from heaters located beyond the deaerator are cascaded to the deaerator. Although the deaerator is generally located above the closed heaters, the difference in pressure is sufficient to overcome both the static and the frictional losses. This difference in pressure decreases with a reduction in load, however, and at some partial main turbine load it becomes insufficient to evacuate the heater drains. They must be switched to a lower-pressure heater or even to the condenser, with a subsequent loss of

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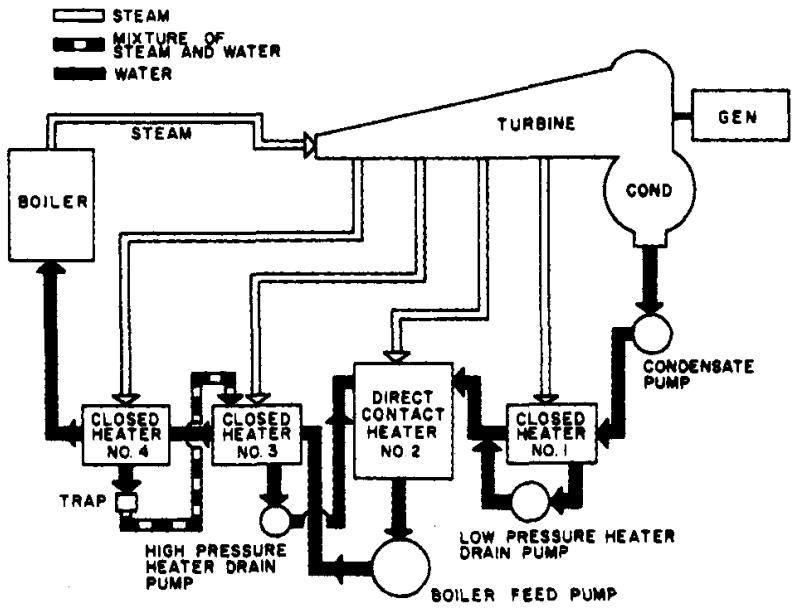


FIGURE 34 Typical arrangement for heater drain pumps

heat. To avoid these complications, a "high-pressure heater drain pump" is generally used to transfer these drains to the deaerator. Actually, this pump has a "reverse" system head to work against; at full load, the required total head may be negative, whereas at light loads, the required head is at its maximum.

Heater drain pumps are subjected to more severe conditions than condensate pumps:

1. Suction pressures are higher and can fluctuate from vacuum to positive.
2. Suction temperatures are higher.
3. Suction transients are more frequent, because of smaller capacitance in the pump suction.
4. Discharge transients are more frequent, because of regulating valve controls to prevent excessive pump runout.

Types of Heater Drain Pumps Heater drain pumps should be capable of continuous operation during sudden load fluctuations. They should be designed to run-dry momentary and once more pump forward as suction pressure is reestablished. They should be designed for wide ranges of flow from minimum to runout.

In the past, heater drain pumps were horizontal pumps with either single- or multistage construction depending on head requirements of a particular system. Horizontal pumps offered advantages such as accessibility and ease of maintenance. However, horizontal pumps with multistage construction were often unreliable, because of rotor contact problems during suction transient conditions.

For low-head services, single-stage horizontal pumps are appropriate. These pumps should be robust designs such as API process pumps (Figure 39). For high-head services, specialized vertical can-type pumps are most appropriate (Figure 30). As previous described for condensate pumps, vertical can-type pumps offer the advantage of increased NPSH available to the first-stage impeller below the floor elevation, which reduces the impact of transients. Also, the rotors of vertical pumps are not subject to static deflections which cause automatic contact at the impeller wear rings. Nevertheless, vertical can-type heater drain pumps should be fitted with carbon bearings and hard-surfaced journals to improve their dry-run capability. In addition, the bearing in the suction bell is often pressurized with water from the first-stage casing to ensure continuous lubrication during suction transients.

Like condensate pumps, all heater drain pumps should be continuously vented back to the steam space in the feedwater heater. Also like condensate pumps, shaft seals should

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be fitted to prevent air ingress during vacuum conditions. However, materials of construction for heater drain pumps should be like those of boiler-feed pumps: steel casings with stainless or chrome steel impellers.

CONDENSER CIRCULATING PUMPS

Service Conditions Condenser circulating pumps pump cooling water through the condenser (refer to Figure 1). The source of the cooling water can be a natural body of water, such as the sea, a river, or lake, or a cooling tower. The arrangement with natural source is called an open cooling water system. The arrangement with the cooling tower is called a closed cooling water system, because the same water is cycled through the system. Closed cooling water systems can use forced draft or natural draft cooling towers with makeup water from some natural source. Alternately, some closed cooling water systems with direct contact-type condensers and an air fin-type cooling towers use cool condensate as the cooling water.

Condenser circulating pump services are mostly the same for open and closed cooling water systems. Generally, closed cooling systems require more head from circulating pumps because of static height of cooling tower. Pump materials of construction must take into account increased concentrations of corrosive agents as water evaporates in closed cooling water systems. In open cooling water systems, concentrations of suspended solids from the water source must be considered.

Types of Pumps Condenser circulating pumps may be of either horizontal or vertical construction. For many years, the low-speed, horizontal, double-suction volute centifugal pump (Figure 35) was the preferred type. This pump has a simple but rugged design that allows ready access to the interior for examination and rapid dismantling if repairs are required.

Today, condenser circulating pumps in most new steam power plants are wet-pit vertical pumps with flows ranging from 60,000 to 300,000 gpm. Some large plants use concrete volute-type pumps with flows up to 500,000 gpm.

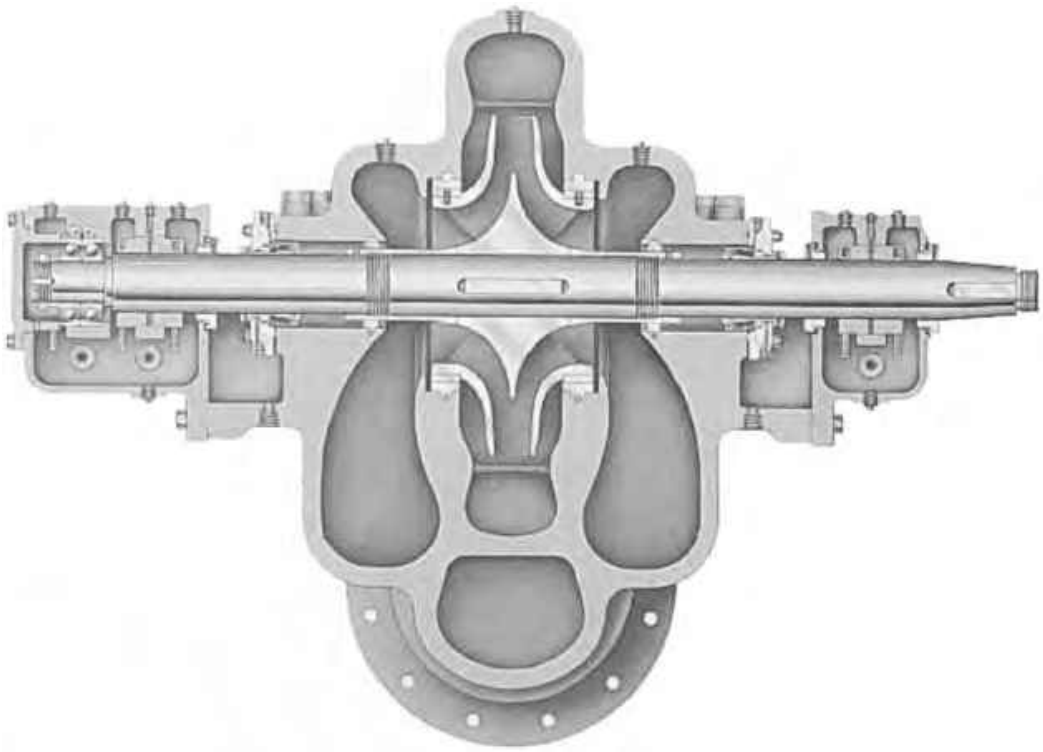


FIGURE 35 Horizontal double suction single stage pump, IDP model LN (Flowserve Corporation)

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Mechanical Considerations The dry-pit installation was a single-suction, medium-specific-speed, mixed-flow pump (Section 2.2.1, Figure 79). This design combined the high efficiency and low maintenance of the horizontal double-suction radial-flow centrifugal pump with lower cost and slightly higher rotative speeds.

Because of their suction and discharge nozzle arrangements, these pumps are ideally suited for vertical mounting in a dry pit, preferably at the lowest water level, so they are self-priming on starting. They are directly connected to solid-shaft induction or synchronous motors, either close-coupled or with intermediate shafting between the pump and the motor, which is then mounted well above the pump pit floor.

Like the horizontal double-suction pump, the vertical dry-pit mixed-flow pump is a compact and sturdy piece of equipment. Its rotor is supported by external oil-lubricated bearings of optimum design. This construction requires the least attention, for the oil level can be easily inspected by means of an oil sight glass mounted at the side of the bearing or oil reservoir.

Because the rotor is readily removed through the top of the casing, facilitating maintenance and replacement, the pump does not have to be removed from its mounting and the suction and discharge connections do not have to be broken to make periodic inspections or repairs.

In recent years, power plant designers have shown a preference for the wet-pit column-type condensate circulating pump. The term *wet-pit* normally implies a casing diffuser-type pump, employing a single open vane impeller. The wet-pit pump (Figure 36) employs a long column pipe that supports the submerged pumping element. It is available with open main shaft bearings lubricated by the water handled, or with enclosed shafting and bearings, lubricated by clean, fresh, filtered water from an external source. There is some danger of contamination of the lubricating water from seepage into the shaft enclosure tube during shut-downs.

Pulling up the column in a long pump requires special facilities and, in addition, the discharge flange must be disconnected when withdrawing the pump and column from the pit. This design has been designated a "non-pull-out" design (Figure 36). To avoid the necessity of lifting the entire pump when the internal parts require maintenance, some units are built so the impeller, impeller shroud, casing, and shaft assembly can be removed from the top without disturbing the column pipe assembly. (The driving motor must be removed.) These designs are commonly designated "pull-out" designs (Figure 37).

Condenser cooling water is often corrosive. Power plants are often located near salt or brackish bodies of water. Plants near rivers often encounter water contaminated with high silt levels. With such waters, selection of materials can be critical to long service life. Material selection for sea water applications must also consider the potential for electrolytic (galvanic) corrosion.

Performance Characteristics Condenser circulating pumps are normally required to work against low or moderate heads. Extreme care should be exercised in calculating the system frictional losses, which include losses from friction in the condenser. If more total head is specified than is required, the resulting driver size may be unnecessarily increased. For instance, an excess of 1 or 2 ft (0.3 or 0.6 m) in an installation requiring only 20 ft (6 m) of head represents an increase of 5 to 10% in excess power costs.

The range of suction lift for dry-pit pumps must be determined very accurately and checked with the manufacturer to ensure that cavitation will be avoided in the installation. Priming facilities must be provided, or the pump must be installed in a dry pit at such an elevation that the water in the suction channel leading to the pump will be maintained at the level recommended by the manufacturer. This presents no problem in a wet-pit installation because the pump column can be made long enough to provide adequate submergence, even with minimum water levels in the suction well or pit. The dry-pit pump will generally have 3 to 4% higher efficiency than the wet-pit type and therefore 3 to 4% lower power consumption. The two types are available for the same specific speed range. When pumping total head is 25 ft (7.6 m) or less, an axial-flow propeller (approximately 10,000 specific speed in USCS units) can be used in either type of pump.

The low-specific-speed, double-suction pump has a very moderate rise in head with reducing capacities and a nonoverloading power curve with a reduction in head. The mixed-flow impeller with a higher specific speed has a steeper head-capacity curve and a

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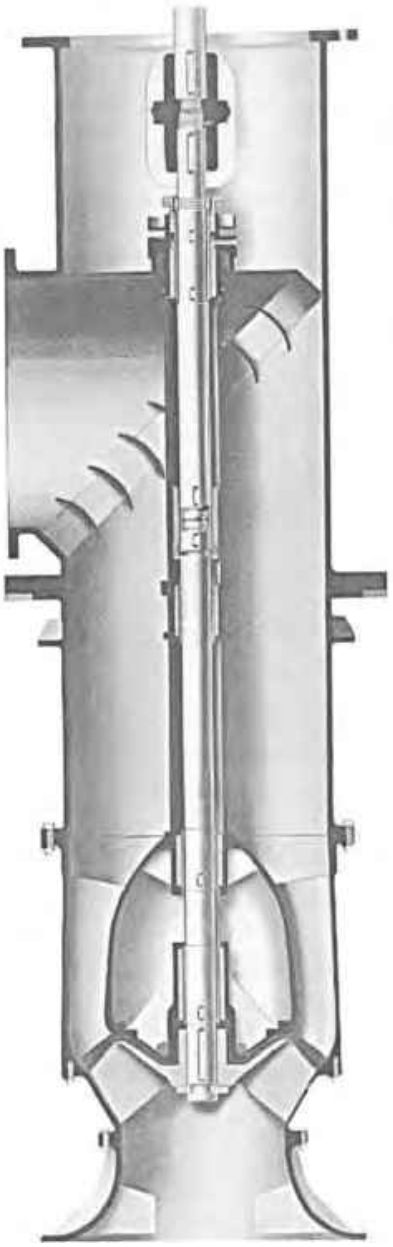


FIGURE 36 Vertical wet pit circulating water pump (non-pull-out) (Flowserve Corporation)



FIGURE 37 Vertical wet pit circulating water pump (pull-out) (Flowserve Corporation)

reasonably flat power curve that is also nonoverloading. As the specific speed increases, the steepness of the head-capacity curve increases and the curvature of the power curve reverses itself, hitting a maximum at the lowest flow. Finally, the curve of a high-specific-speed propeller pump has the highest rise in both head-capacity and power-capacity curves toward zero flow. The head range developed by the mixed-flow pump is ideal for condenser service; this pump is usually furnished with an enclosed impeller, which produces a relatively flat head-capacity curve and a flat power characteristic.

Higher head circulating water pumps were developed in the 1970s as cooling towers were introduced to improve plant efficiency and environmental contamination. The cooling tower arrangement effectively increased the total system resistance head requirements.

System Hydraulics The dry-pit pump is not too sensitive to the suction well design because the inlet piping and the formed design of the suction passages into the pump

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normally ensure a uniform flow into the eye of the impeller. On the other hand, the higher-speed wet-pit pumps are more sensitive to departures from ideal inlet conditions than the low-speed centrifugal volute pump or the medium-speed mixed-flow pump. A discussion of the arrangements recommended for wet- and dry-pit pump installations is presented in Section 13.1.

Condenser circulating pumps are typically provided in sets of two, three, or four identical pumps operating in parallel on the same cooling water system. Generally, discharge throttle valves are not provided. Flow is simply controlled by reducing the number of pump in service. Therefore, multiple condenser circulating pumps operate with flows from rated to runout conditions. Careful consideration must be given to pump runout capability, including NPSH required at maximum flow conditions.

Large concrete volute pumps are often configured as a single 100% pump in a condenser circulating cooling water system. If system flow reductions are needed, variable speed control of the pumps is best for single pump configurations.

Drivers Dry-pit horizontal pumps are typically coupled with flexible-element couplings to horizontal electric motors. These pumps operate at fixed speeds.

Wet-pit vertical pumps for condenser circulating service are usually coupled with rigid couplings to vertical electric motors. In this case, the motor thrust bearing is designed for the pump hydraulic thrust loads plus the rotor static weights. Alternative designs are available with thrust bearings mounted in the pumps and flexible element coupling. However, these designs are more difficult to service and increase the height of the pump-motor assembly.

Concrete volute-type pumps are typically driven by electric motors with speed-decreasing gearboxes, because these large pumps have low rotating speeds. The pumps have oil or grease lubrication bearings in the pump assembly and therefore flexible-element couplings are used.

BOILER CIRCULATING PUMPS

The forced circulation, or controlled circulation, boiler requires the use of circulating pumps that take their suction from a header connected to several downcomers, which originate from the bottom of the boiler drum and discharge through the various tube circuits operating in parallel (Figure 38). The circulating pumps therefore must develop a pressure

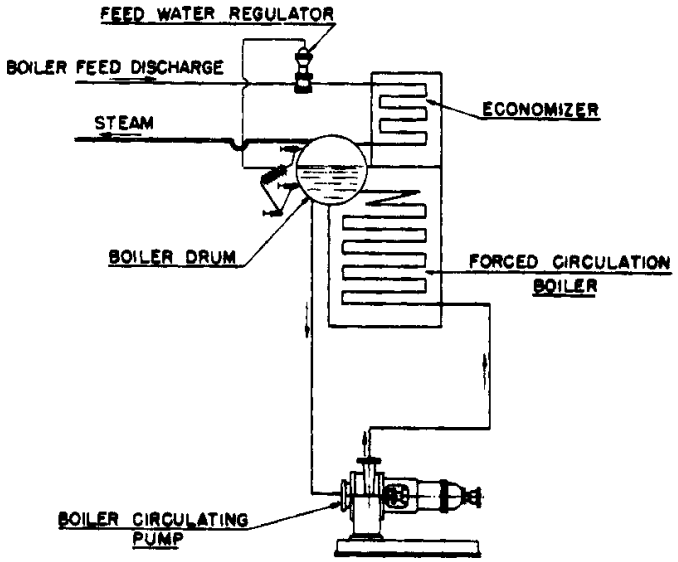


FIGURE 38 Forced, or controlled, circulation system

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equivalent to the frictional losses through these tube circuits. Thus, in the case of different boilers operating in pressure ranges from 1800 to 3000 lb/in^2 (124 to 207 bar), the boiler circulating pump must handle feedwater from 620 to 690°F (326 to 365°C) under a suction pressure of 1800 to 2900 lb/in^2 (124 to 200 bar). Such a combination of high suction pressure and high water temperature at saturation imposes very severe conditions on the circulating pump stuffing boxes, making it necessary to develop special designs for this part of the pump.

The net pressure to be developed by these pumps is relatively low, ranging from 50 to 150 lb/in^2 (3.4 to 10.3 bar). Hence these are single-stage pumps with single-suction impellers and a single stuffing box. The high boiler pressure imposes an extremely severe axial thrust on the pump, placing a load of several tons on the thrust bearing. In many cases, a thrust-relieving device must be incorporated to permit pump start-up. Two general types of construction are used for this service: (1) the conventional centrifugal pump with various stuffing box modifications (Figure 39) and (2) the submersible motor pump of either the wet- or dry-stator type (Figure 40). In the lower boiler pressure range-up to 500 or 600 lb/in^2 gauge (34 or 41 bar), the construction shown in Figure 41 may be used. The pump is of the same general type as is used on high-pressure heater drain service. The packed stuffing box or mechanical seal is preceded by a pressure-reducing bushing. Feedwater from the boiler-feed pump discharge, at a temperature lower than in the boiler drum and at a pressure somewhat higher than pump internal pressure, is injected into the middle of this bushing. Part of this injected feedwater proceeds toward the pump interior, making a barrier against the outflow of high-temperature water. The rest proceeds outward to a bleed portion of the bushing, from where it is bled to a lower pressure, often the deaerator. The packing or mechanical seal needs to withstand only the lower boiler-feed pump temperature and a much lower pressure than boiler pressure.

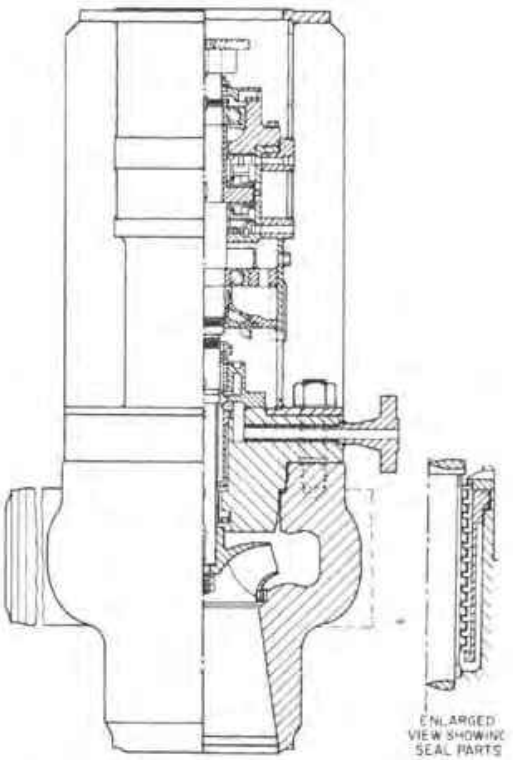


FIGURE 39 Vertical injection-type boiler circulating pump for high pressures (Flowserve Corporation)

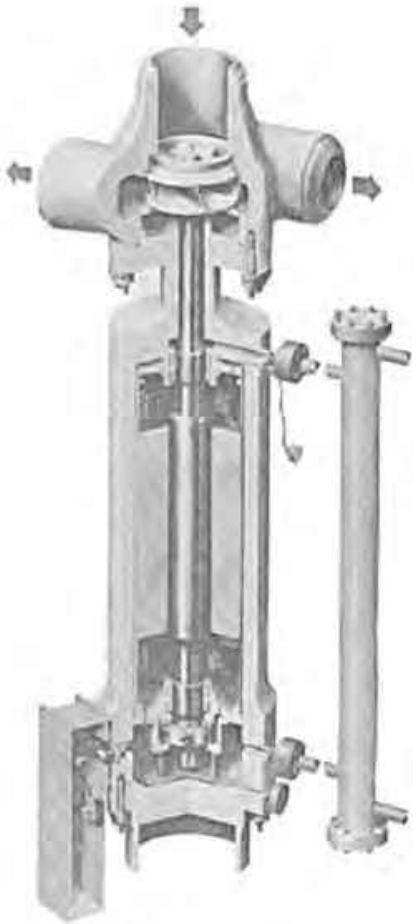


FIGURE 40 Boiler circulating pumps driven by a wet stator motor (Hayward Tyler)

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FIGURE 41 End-suction boiler circulating pump for low-pressure range (Flowserve Corporation)

More sophisticated designs are required for pressures from 1800 to 3000 lb/in^2 gauge (124 to 207 bar) (Figure 39). The shaft is sealed by two floating ring pressure breakdowns and a water-jacketed stuffing box. Boiler feedwater is injected at a point between the lower and upper stacks of floating ring seals at a pressure about 50 lb/in^2 (3.5 bar) above the pump internal pressure. Here again, part of this injection leaks into the pump interior and the rest leaks past the upper stack of seals to a region of low pressure in the feed cycle. Leakage to atmosphere is controlled by the conventional stuffing box located above the upper stack. The seal injection and leakoff control system is very sensitive to boiler and feedwater pump transients. Loss of injection results in flashing in the sealing chamber and failure of the sealing rings. Current technology utilizes a two-stage, high-pressure mechanical seal. This eliminates the need for a separate seal injection system and seal-injection booster pump.

The available NPSH may not be sufficient at start-up, when the water in the boiler is cold and the pressure is low. Therefore certain installations include two-speed motors so a lower NPSH is required at start-up. There is an added advantage to this arrangement: under normal operating conditions, the feedwater will heat boiler saturation temperature and therefore will have a specific gravity of as low as 0.60; at start-up, however, the specific gravity will be 1.0. The power consumption on cold water would therefore be some 65% higher than in normal operation if the pump operates at the same speed, and a much larger motor would be required. If a two-speed motor is used, however, the pump is operated at lower speed when the water is cold, and the motor need be only large enough to supply the maximum power required under normal operating conditions.

ASH-HANDLING PUMPS

Boilers that use solid fossil fuels produce refuse that is generally classified as ash and includes bottom ash or slag, fly ash, and mill rejects such as pyrites and so on. This refuse must be removed and disposed of, either by hydraulic or pneumatic conveying. The latter, of course, does not involve the use of pumps and need not be discussed here.

Fly ash and bottom ash is removed from the boiler system through a series of water ejectors. Centrifugal pumps provide the water, taken from either the circulating water pump header or ash-settling pond. These pumps (Figure 42) are subjected to severe duty as pump flows vary significantly and the pumpage is often contaminated with suspended silt or fly ash.

Hydraulic conveying is generally restricted to bottom ash and mill rejects. There are a great number of different systems in use today, and the reader should consult boiler design and operation literature to become acquainted with this subject. What is common to most hydraulic conveying systems is that they use centrifugal pumps to handle concentrated

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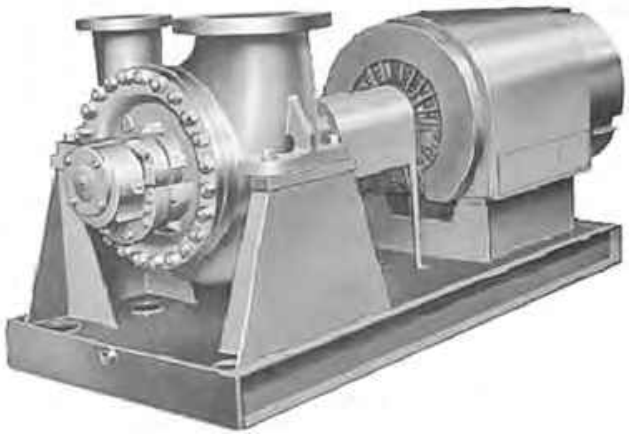


FIGURE 42 Horizontal single stage double suction, high head pump (Flowserve Corporation)

ash slurries that may contain considerable amounts of coarse, heavy pieces of stone, slate, and iron pyrites. Pumps suitable for this service are described in detail in Section 4.2.

SLURRY PUMPS

Modern coal-fired steam power plants use wet limestone scrubber systems to reduce in high rates the SO_2 emissions from flue gases. These systems utilize a variety of conventional and slurry pumps to prepare and feed limestone slurries and to dewater and transport by-product slurries including:

- Mill slurry pumps
- Limestone slurry feed pumps
- Filter feed pumps
- Filtrate return pumps
- Gypsum slurry pumps

The pumps in these services are single-stage, end-suction centrifugal designs. The slurry pumps have hard-metal or rubber-lined construction. For additional information about slurry pumps, refer to Section 4.2.

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FURTHER READING

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SECTION 12.6
NUCLEAR

12.6.1
NUCLEAR ELECTRIC GENERATION

W. M. WEPFER
W. W. PARRY, JR.

APPLICATION

The primary difference between the pumps for nuclear service and conventional steam generation is the stringent requirements for reliability and safety. The continuous functioning of nuclear pumps is critical to the safe operation of the plant. In addition, many of these units maintain the responsibility of ensuring safe shutdown of the facility during emergency conditions. Because the release of radioactive fluids could result in potential personnel and environmental hazards along with associated costly cleanup, nuclear pumps must be self-contained. The rules and regulatory standards for nuclear pumps are dedicated to the promotion of public safety through reliable operation. Pump manufacturers must be fully knowledgeable of these standards and certified by the American Society of Mechanical Engineers (ASME) (or an equivalent authority outside the United States) before they can offer pumps for the nuclear market.

All nuclear power plants built for public utility service involve the generation of steam. The method of steam generation differentiates the two major types of plants. The pressurized water reactor (PWR) employs two distinct systems, a primary and a secondary loop. The primary loop circulates pressurized radioactive fluid from the reactor to the steam generator where it heats the secondary fluid. The secondary loop directs steam from the steam generator to the turbines and condensed fluid back to the generators. A boiling water reactor (BWR), on the other hand, only employs one loop. The steam is generated in the reactor and then circulated through the turbines, condensers, and heaters and back to the reactor.

It is usual for the plant to contain one major system, or loop, and a number of supporting systems. Because virtually every system requires some form of pumping, it is conventional to designate each pump by the name of the system with which it is associated. For each of the systems previously discussed, the main and auxiliary systems are reviewed, including brief descriptions of the pumping requirements.

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PWR Plants PWR plants employ two separate main systems to generate steam. In the primary system, the water is circulated by *reactor coolant pumps* through the nuclear reactor and large steam generators. Overpressure is provided to prevent vapor formation. The secondary side of the steam generator provides nonradioactive steam to the turbogenerator. Typical primary water conditions are 2250 lb/in^2 gauge (15.51 MPa) and 550°F (288°C). Table 1 lists the parameters of more important pumps used in PWR plants.

Figure 1 is a simplified flow diagram of a PWR reactor coolant system. Only one primary loop is shown, but in practice, plants use two, three, or four loops in parallel, each loop consisting of its own reactor coolant pump, steam generator, and optional stop valves. A single reactor and pressurizer supply all loops. The pressurizer provides the overpressure referred to by maintaining a body of water at an elevated temperature such that its vapor pressure satisfies the primary loop pressure requirements.

In the PWR plant, high-pressure water circulates through the reactor core to remove the heat generated by the nuclear chain reaction. Figure 2 is an illustration of a typical PWR reactor vessel. The heated water exits from the reactor vessel and passes via the coolant loop piping to the primary side of the steam generators. Here it gives up its heat to the feedwater to produce steam for the turbogenerator. The primary-side cycle is completed when the reactor water is pumped back to the reactor by the reactor coolant pumps. The secondary-side cycle, isolated from the primary loops, is completed when the *feedwater pumps* return water to the secondary side of the steam generator.

Because a reactor, after it has been critical, continues to generate heat even when shut down, *residual heat removal* (RHR) *pumps* are provided to circulate reactor water through coolers any time the reactor is inoperable and at low pressure, even during refueling. These pumps serve other functions also, as described next.

The chemical and volume control system (Figure 3) performs a number of functions. Through *charging pumps,* which may be centrifugal or positive displacement or may include some of each type, the primary system can be filled and pressurized when cold. When the system is hot, the pumps are used to maintain the water level in the pressurizer and to replenish any fluid drawn from the primary loops by other systems. Additionally, the pumps supply clean water to the reactor coolant pump seals and are used to adjust the boric acid concentration in the reactor coolant water, which provides an auxiliary means of reactor power regulation. If positive displacement pumps are included in the chemical and volume control system, they are also used to hydrostatically test the reactor primary coolant system. Where all the charging pumps are centrifugal, it is customary to provide a small positive displacement pump exclusively for this hydrostatic testing.

For cooling essential components and for supplying a variety of heat exchangers, a component cooling water system is provided. *Component cooling water pumps* circulate clean water at low system pressures for the purpose of cooling: (1) primary water, which is continuously bled for purification, (2) main and auxiliary pump bearings and seals, (3) primary pump thermal barriers, (4) large motors, (5) the containment vessel, and (6) the spent fuel pit water.

When the primary pressure boundary is breached, elements of the emergency core cooling system (ECCS) are immediately activated. The primary function of the ECCS following a loss-of-coolant accident is to remove the stored and fission product decay heat from the reactor core. The safety injection system (Figure 4) does most of this. Upon actuation of the safety injection signal, the charging pumps inject boric acid solution, which is stored in special tanks and continuously circulated by the *boron injection recirculation pumps,* into the reactor coolant system. At the same time, the *residual heat removal pumps* are started. These pumps take suction from a large refueling water storage tank and inject cold water into the reactor coolant circuit. To provide additional capacity, the *safety injection pumps* are started, taking suction from the cold water in the refueling water storage tank and pumping this water into the reactor coolant system. If the large storage tank should run dry, these pumps will take suction from the containment sump.

Operated by a pressure signal, *containment spray pumps* condense any steam in the containment in order to lower the temperature and pressure in that environment. By taking suction from the containment sump, these pumps continue to circulate water through spray nozzles located near the top of the containment until the pressure has been reduced to an acceptable level.

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TABLE 1 Typical nuclear pump parameters in PWR plants

Pump	Number per plant	Flow, gpm (m^3/h)	Head, ft (m)	Design pressure, lb/in^2 (MPa)	Design temp., °F (°C)	Driver, hp (kW)	Shaft	Length or height, including driver, in (mm)	Speed (nominal), rpm	Notes
Reactor coolant	4	100,000 (22,700)	290 (88)	2500 (17.2)	650 (343)	7000 (5220)	Vert.	305 (7750)	1200	
Component cooling water	3	4800 (1090)	250 (76)	200 (1.38)	200 (93)	450 (336)	Horiz.	110 (2790)	1800	
Residual heat removal	2	3800 (863)	350 (107)	600 (4.14)	400 (204)	500 (373)	Vert.	97 (2460)	1800	
Containment spray	2	2600 (590)	450 (137)	300 (2.07)	300 (148)	400 (298)	Horiz.	112 (2840)	1800	
Spent fuel pit cooling	3	4500 (1022)	150 (46)	150 (1.03)	200 (93)	250 (186)	Horiz.	87 (2210)	1800	
Charging (centrifugal)	2	120 (27)	5800 (1768)	2800 (19.3)	300 (148)	600 (448)	Horiz.	234 (5940)	4850	Gear drive
Charging (reciprocating)	1	98 (22)	5800 (1768)	2800 (19.3)	250 (121)	200 (149)	Horiz.	208 (5280)	205	Reciprocating
Safety injection	2	440 (100)	2680 (817)	1750 (12.07)	300 (148)	450 (336)	Horiz.	190 (4830)	3600	
Chilled water	2	400 (91)	150 (46)	150 (1.03)	200 (93)	40 (30)	Horiz.	52 (1320)	3600	
Spent resin sluicing	1	150 (34)	250 (76)	240 (1.66)	250 (121)	30 (22)	Horiz.	46 (1170)	3600	
Reactor coolant drain tank	2	150 (34)	250 (76)	240 (1.66)	250 (121)	30 (22)	Horiz.	46 (1170)	3600	

(Continued)

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TABLE 1 Typical nuclear pump parameters in PWR plants (*Continued*)

Pump	Number per plant	Flow, gpm (m^3/h)	Head, ft (m)	Design pressure, lb/in^2 (MPa)	Design temp., °F (°C)	Driver, hp (kW)	Shaft	Length or height, including driver, in (mm)	Speed (nominal), rpm	Notes
Boric acid transfer	2	100 (22.7)	200 (61)	150 (1.03)	200 (93)	15 (11)	Horiz.	45 (1140)	3600	
Boron recycle evaporator feed	2	100 (22.7)	200 (61)	150 (1.03)	200 (93)	15 (11)	Horiz.	45 (1140)	3600	
Boron injection recirculation	2	20 (4.5)	100 (30.5)	240 (1.66)	200 (93)	3 (2.2)	Horiz.	18 (460)	3600	Canned
Spent fuel pit skimmer	1	100 (22.7)	50 (15.2)	150 (1.03)	200 (93)	3 (2.2)	Horiz.	42 (1070)	1800	
Refueling water purification	1	200 (45)	200 (61)	150 (1.03)	200 (93)	30 (22)	Horiz.	52 (1320)	1800	
Waste processing system	5	100 (22.7)	200 (61)	150 (1.03)	200 (93)	15 (11)	Horiz.	45 (1140)	3600	
Gas decay tank drain	1	10 (2.27)	90 (27)	150 (1.03)	180 (82)	3 (2.2)	Horiz.	15 (380)	3600	
S. G. blowdown—spent resin sluice	1	110 (25)	165 (50)	150 (1.03)	100 (37)	15 (11)	Horiz.	39 (990)	3600	

Source: Westinghouse Electric.

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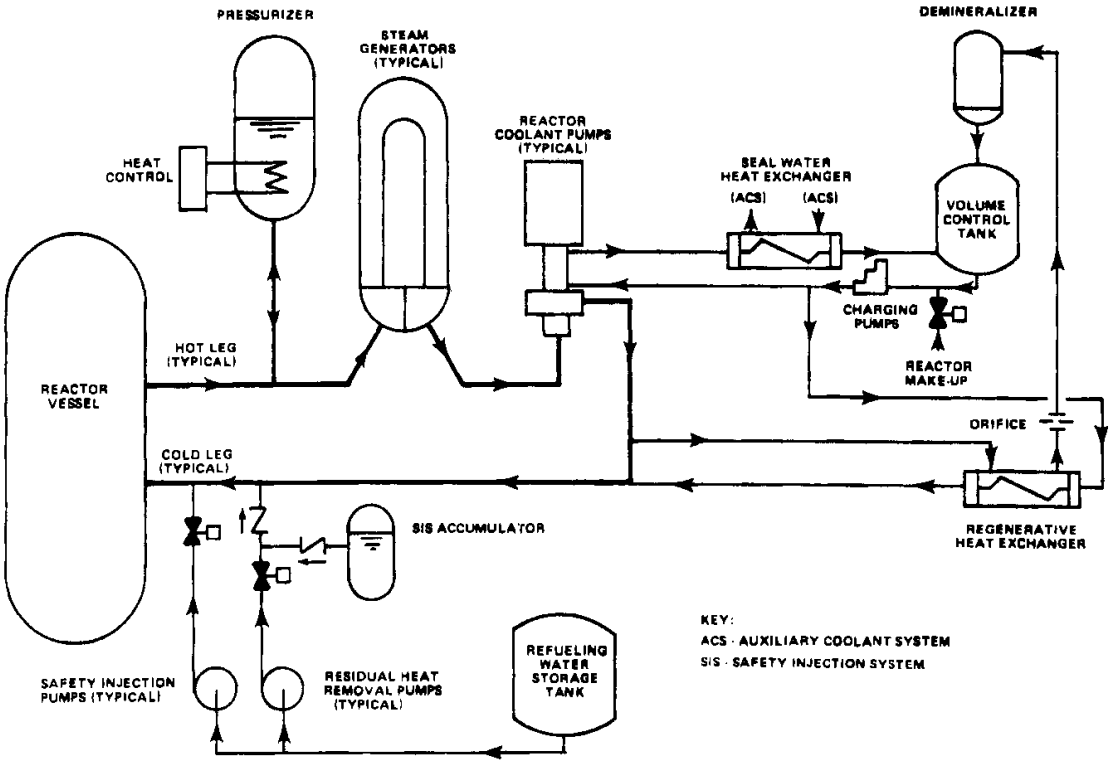


FIGURE 1 Flow diagram for pressurized water reactor coolant system (Westinghouse Electric)

All of the pumps and vital equipment associated with pipe rupture (that is, the emergency core cooling system) is supplied with diesel generator electrical power backup. If needed, the diesel generators will accept the various pump loads sequentially at intervals of a few seconds until all needed equipment is on line.

Other pumps serve other systems. *Spent fuel pit pumps* provide the necessary cooling of the fuel elements that have been removed from the reactor. Resin beds, which are a part of the water purification system, are flushed to a storage tank by *spent resin sluicing pumps.* An evaporator package, partly for removing boron from the primary water, is supplied by *recycle evaporator feed pumps. Chilled water pumps* supply the boron thermal regeneration system. Similarly, other pumps, some not listed in Table 1, support auxiliary systems.

BWR Plants In BWR plants, active boiling takes place in the nuclear core and steam is piped to the turbogenerator (Figure 5). Typical reactor water conditions are 1000 lb/in^2 gauge (6.895 MPa) and 550°F (288°C). In the United States, the plant is usually arranged with a low-leakage containment vessel completely surrounding a dry well and a pressure-suppression pool (Figure 6). The containment vessel is a cylindrical steel or concrete structure with an ellipsoidal dome and a flat bottom supported by a reinforced concrete mat. The containment forms a security barrier and prevents the escape of radioactive products to the atmosphere if an accident should occur.

Table 2 shows the principal pumps used in BWR plants together with significant characteristic data.

To assure a high reactor flow rate and to avoid local areas of core overheating, internal *jet pumps* have been used in all but the earliest U.S. BWR plants. These jet pumps are driven by large-volume, medium-head *recirculation pumps.* Variable flow rate is achieved either by flow-control valves or by variable-speed motors driven by motor generator sets. The latest designs employ a flow-control valve. The use of jet pumps decreases the size of the external loop piping and pumps and provides a core reflood capability in the event of pipe rupture. The main recirculation pumps are not required for emergency cooling. In normal operation, the recirculation pumps operate in conjunction with the jet pumps,

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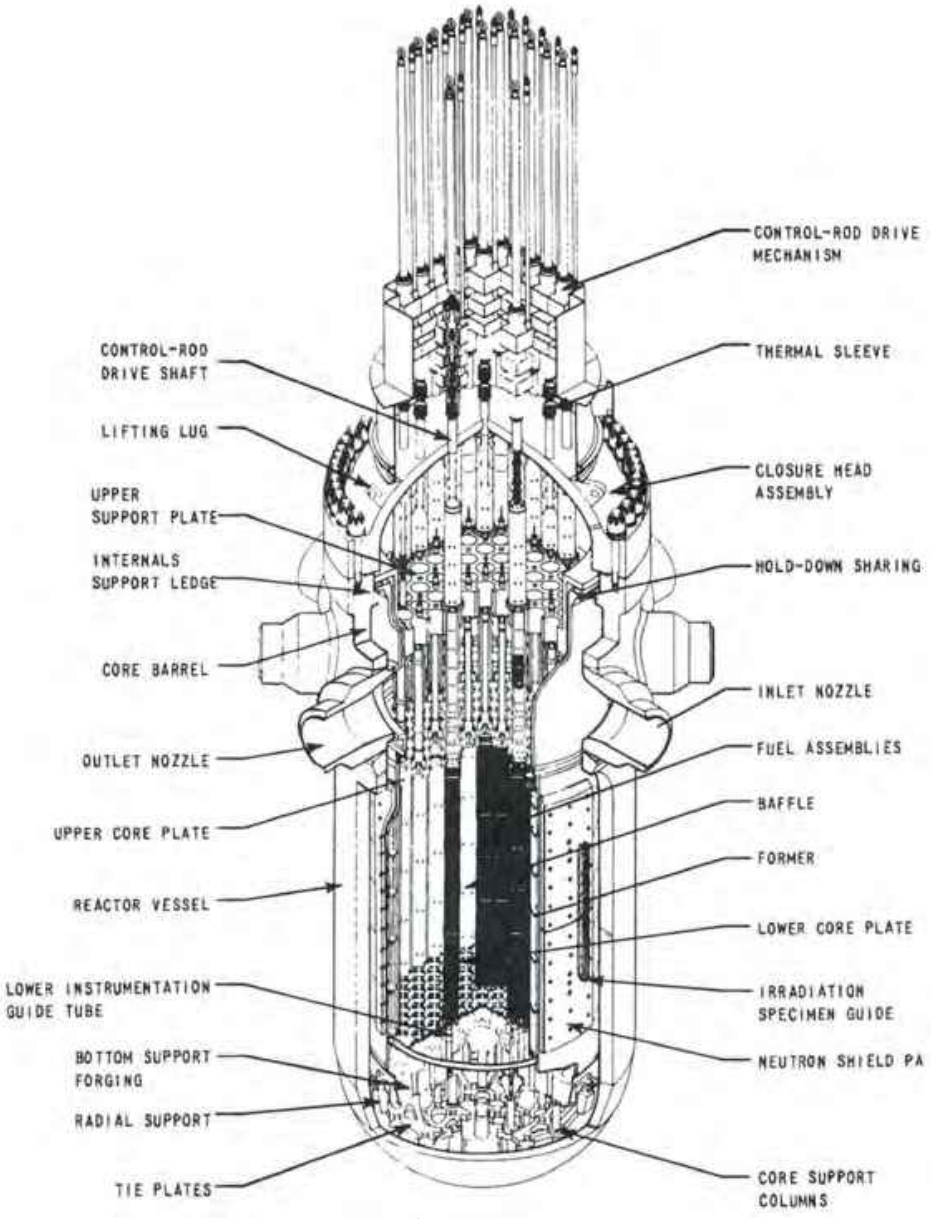


FIGURE 2 PWR reactor vessel (Westinghouse Electric)

which are wholly contained in the reactor vessel. The purpose of the jet pumps is to increase the flow from the recirculation pumps at reduced head for reactor cooling. The jet pumps have no moving parts. In addition to their normal service, they also play a role in the natural circulation of the reactor water during emergency cooling.

Several subsystems operate in support of the recirculation system, and each contains one or more pumps. *Reactor water cleanup pumps* are used in a filter-demineralizer system to remove particulate and dissolved impurities from the reactor coolant. This system also removes excess water from the reactor. The control rods are operated hydraulically with water pressure provided by the *control rod drive pumps.* The pumps are located in an auxiliary building, and the fluid is piped to the control rod drive units, which are positioned directly under the reactor vessel. For those components that require constant cooling, such as the recirculation pump motors and other equipment located in the containment, auxiliary, fuel, or radwaste buildings, *closed cooling water pumps* furnish the necessary flow. These pumps are located in an auxiliary building to permit ready access for servicing if needed. The system is closed so it can be isolated from an ultimate, usually raw water, heat sink, such as a river, lake, or ocean.

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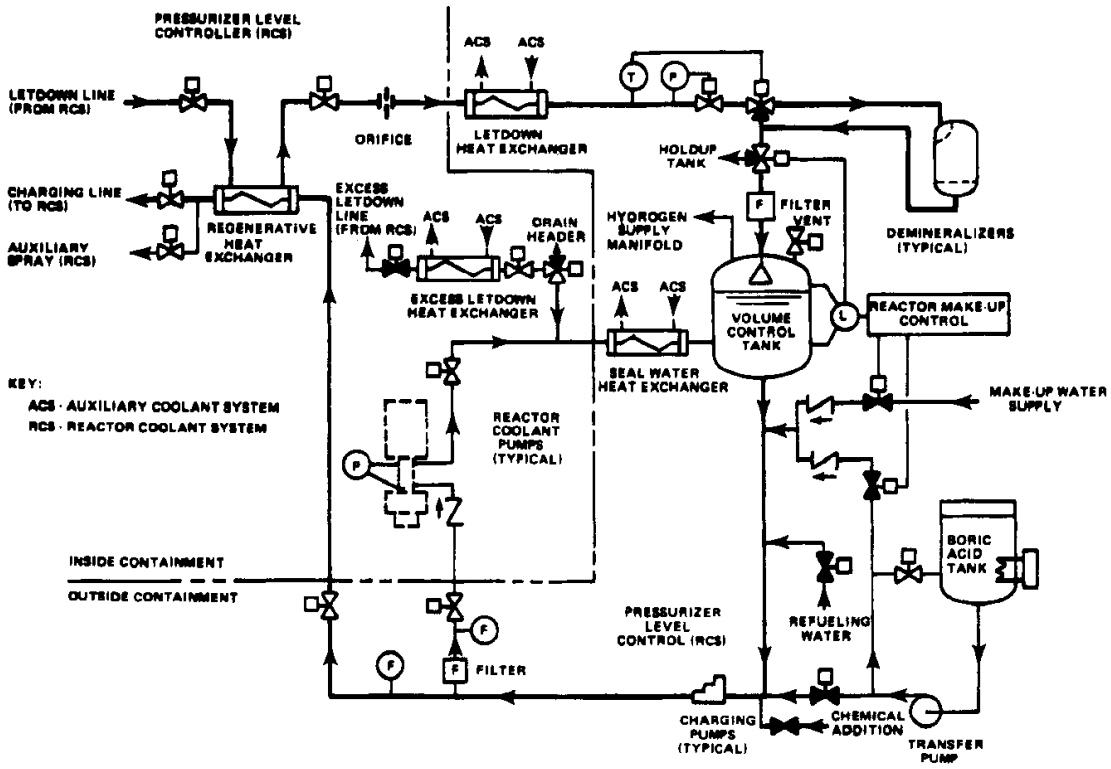


FIGURE 3 Flow diagram for chemical and volume control system (Westinghouse Electric)

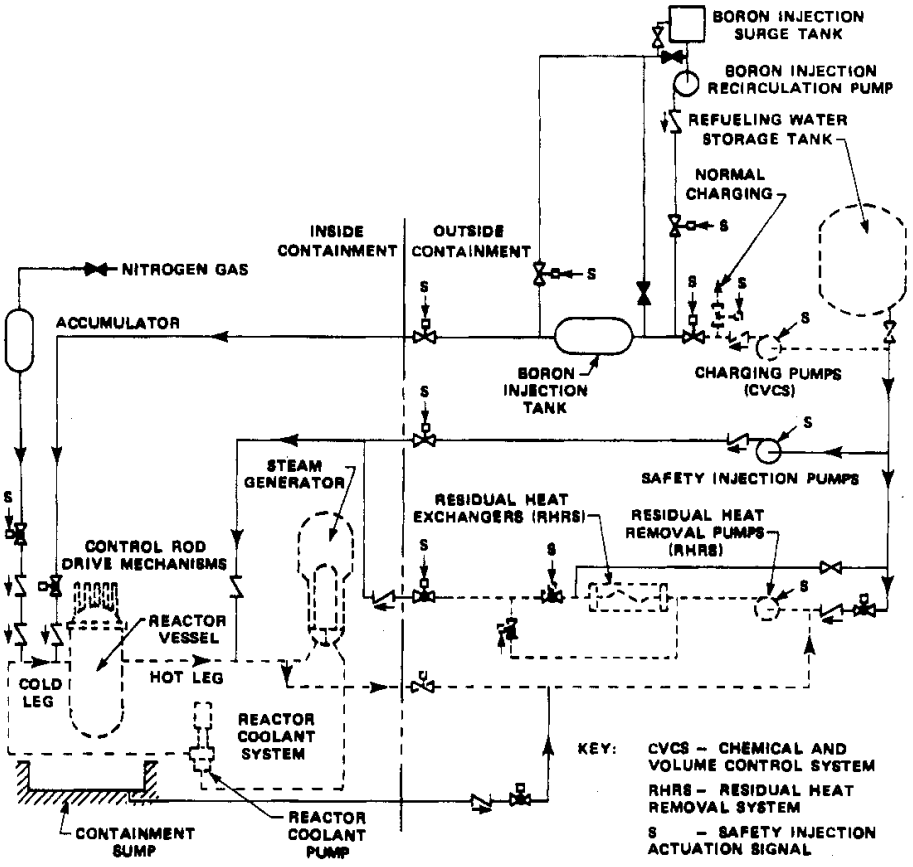


FIGURE 4 Flow diagram for safety injection system (Westinghouse Electric)

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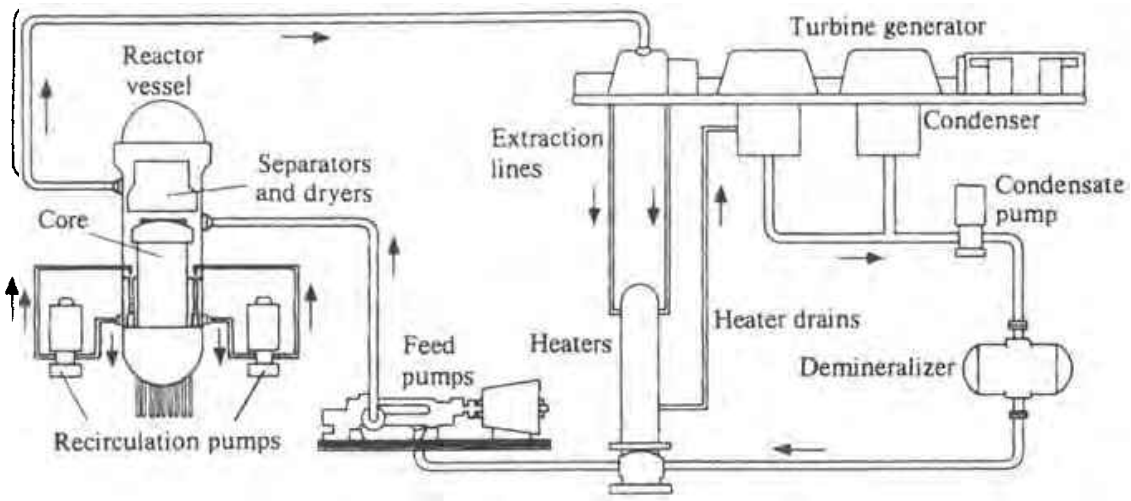


FIGURE 5 Boiling water direct cycle reactor system (General Electric)

To cool the fuel stored under water in the fuel building and the water in the upper containment, separate *fuel pool cooling pumps* are provided in an independent system.

The emergency core cooling system is in reality an array of subsystems providing the necessary features, including redundancy, to protect the core in case of a significant malfunction. The high-pressure core spray system uses a vertical *high-pressure core spray pump,* motor-driven but backed by a diesel generator in event of loss of electric power. This pump, a single unit, provides the initial response when a small pipe breaks or an equivalent malfunction occurs. Should this system be inadequate to maintain reactor water level, the reactor vessel is automatically depressurized and the *low-pressure core spray pumps* supply additional capacity. As an added safeguard, the *RHR pumps* are used in a

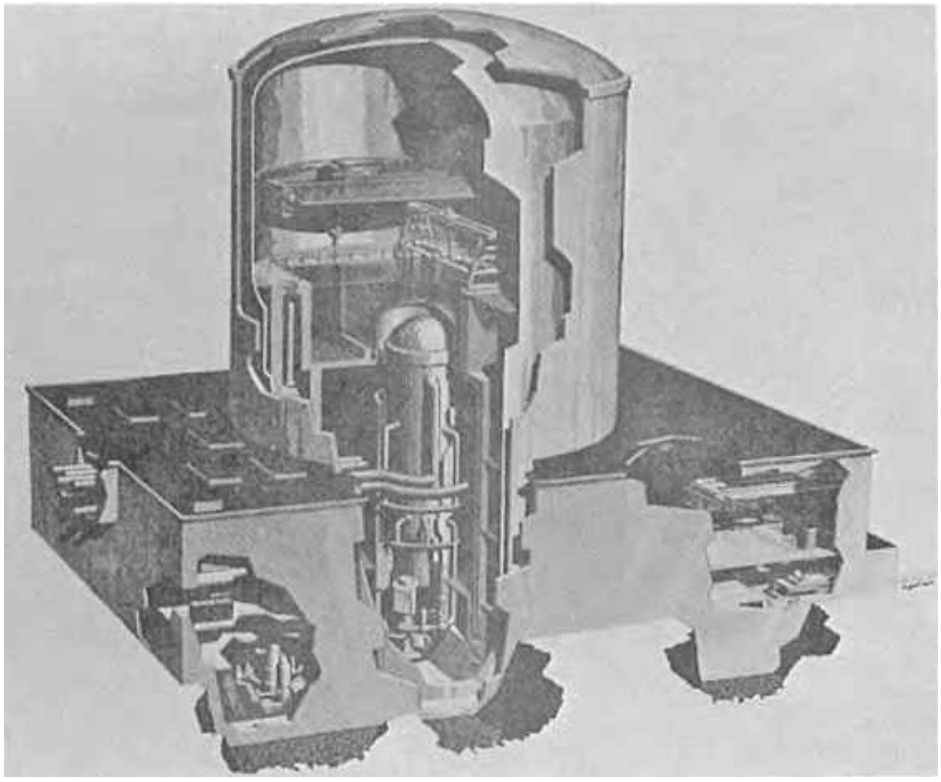


FIGURE 6 BWR containment and reactor vessel, showing one of two external recirculation pumps (General Electric)

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TABLE 2 Typical nuclear pump parameters in BWR plants

Pump	Number per plant	Flow, gpm (m^3/h)[a]	Head, ft (m)	Design pressure, lb/in^2 (MPa)	Design temp., °F (°C)	Driver, hp (kW)[a]	Shaft	Length or height, including driver, in (mm)	Speed (nominal), rpm	Notes
Recirculation coolant	2	44000 (9993)	760 (231)	1675 (11.55)	575 (302)	8000 (5970)	Vert.	240 (6096)	1800	
RHR service water	4	7300 (1658)	115 (35)	150 (1.03)	150 (65)	300 (224)	Vert.	200 (5080)	900	
RHR	3	8520 (1935)	275 (84)	450 (3.10)	212 (100)	900 (670)	Vert.	350 (8890)	1800	
High-pressure core spray	1	1465 (333)	2600 (792)	1600 (11.03)	212 (100)	3000 (2240)	Vert.	500 (12700)	1800	
Low-pressure core spray	1	6000 (1363)	280 (85)	550 (3.79)	212 (100)	1750 (1310)	Vert.	400 (10160)	1800	
Closed cooling water	2	2040 (463)	110 (34)	150 (1.03)	212 (100)	60 (45)	Horiz.	100 (2540)	1800	
Reactor core isolation cooling	1	700 (159)	2600 (792)	1500 (10.34)	212 (100)	800 (597)	Horiz.	123 (3124)	4000	Turbine-driven variable-speed
Fuel pool cooling and cleanup	2	600 (136)	300 (91)	150 (1.03)	150 (65)	75 (56)	Horiz.	95 (2413)	1800	
Reactor water cleanup	2	150 (34)	500 (152)	1400 (9.65)	560 (293)	50 (37)	Horiz.	78 (1981)	3600	

(*Continued*)

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TABLE 2 Typical nuclear pump parameters in BWR plants (*Continued*)

Pump	Number per plant	Flow, gpm (m^3/h)[a]	Head, ft (m)	Design pressure, lb/in² (MPa)	Design temp., °F (°C)	Driver, hp (kW)[a]	Shaft	Length or height, including driver, in (mm)	Speed (nominal), rpm	Notes
Control rod drive hydr. system	2	95 (22)	3500 (1067)	1750 (12.07)	150 (65)	300 (224)	Horiz.	168 (4267)	1800	
Standby liquid control	2	40 (9)	2800 (853)	1400 (9.65)	150 (65)	40 (30)	Horiz.	60 (1524)	1800	Reciprocating pumps
Jet	20	10000 (2271)	80 (24)	N.A.	575 (302)	N.A.	Vert.	250 (6350)	N.A.	
Waste evaporator	2	9000 (2044)	50 (15)	50 (0.35)	274 (134)	150 (112)	Horiz.	120 (3048)	720	
Resin tank precoat	1	255 (58)	85 (26)	150 (1.03)	150 (65)	10 (7.5)	Horiz.	65 (1651)	1800	

[a]Vary depending on reactor power rating.
Source: General Electric.

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secondary-mode operation to inject cooling water directly into the reactor vessel. If steam should enter the containment region, the *RHR pumps* operate in another mode—as containment spray pumps—and are manually operated to condense the steam and thus reduce any potential pressure build up in the containment. The *RHR pumps* function when needed to limit the temperature of the water in the suppression pool. The turbine-driven *reactor core isolation cooling pumps*, in a redundant and independent system, inject cool water into the reactor vessel. The *standby liquid control system pumps* inject boron solution into the reactor for alternative shutdown.

Additional Equipment In addition to these systems, a number of support systems exist, most of which require some form of pumping. Without attempting to describe their systems, there are, for example, pumps for feedwater, condensate, chilled water, booster service, condenser service, demineralized water transfer, condensate transfer, dry-well drain, containment drain, concentrated borated water tank, water leg, precoat, radwaste, and sample station.

MAIN COOLANT PUMPS

The term *main coolant pumps* as used here includes both the *recirculation pumps* in BWR plants and the *reactor coolant pumps* in the primary systems of PWR plants. As shown typically in Figures 7, 8, 9, and 10, main coolant pumps use a vertical shaft with the impeller at the bottom and a drive motor coupled to the top end of the shaft.

Bearings Main coolant pumps are usually single-bearing units, although one manufacturer provides a second guide-and-thrust oil-lubricated bearing just below the coupling. Within the pump, hydrodynamic water-lubricated bearings are conventionally used (one supplier using a hydrostatic type). The hydrodynamic bearing generally consists of a hardened sleeve journal shrunk on the shaft and a carbon-lined bearing with some type of self-aligning feature. Bearing diametral clearances are approximately 1.5 mi/in of bearing diameter (0.0015 mm/mm). This rather large clearance is desirable because of thermal transient conditions. Where carbon bearings are used, a cooling mechanism must be provided because the carbon is not suited to long exposure in a hot environment. Many methods are available to accomplish this.

With bearings adequately sized and operating in clean water, virtually trouble-free performance can be expected even with relatively frequent starts and stops.

The hydrostatic, or pressurized, bearing is used where it is not convenient or desirable to provide bearing cooling, for the hydrostatic bearing can be designed to operate in reactor temperature water. Usually it will be larger than its hydrodynamic equivalent and somewhat more sensitive to starting because of the metal-to-metal rubbing that occurs until rotative speed has built up a small pressure to provide a water film clearance. At normal operating speed, however, a hydrostatic bearing will have a significantly larger lubricating film than a hydrodynamic bearing and can tolerate larger particulate matter without wear.

An exception to the previous discussion occurs in both German and Swedish BWR designs, where the recirculation pumps are inverted and inserted directly into the bottom periphery of the reactor vessel and use wet winding motors. There are no jet pumps. Flow is varied by changing motor speed with solid-state power supplies, one for each recirculation pump.

Seals Pump seals are invariably of the pressure-balanced type because of the high pressures involved. Several suppliers use pressure breakdown techniques to distribute the pressure either equally or in some desired proportion between two or more seals in series (Figure 11). The technique is analogous to that of a potentiometer, where particular voltages may be obtained by selecting the proper point along an electrical resistance. Also, as with a potentiometer, the flow through the primary resistance path must exceed the tap off flow in order to maintain the system stability. For pressures in PWR systems, three series seals are frequently used (with each seal taking one-third of the overall pressure), and with lower-pressure BWRs, two series seals are usually sufficient. A margin of safety is built into the systems such that pump operation can be continued even if one of the series seals fails.

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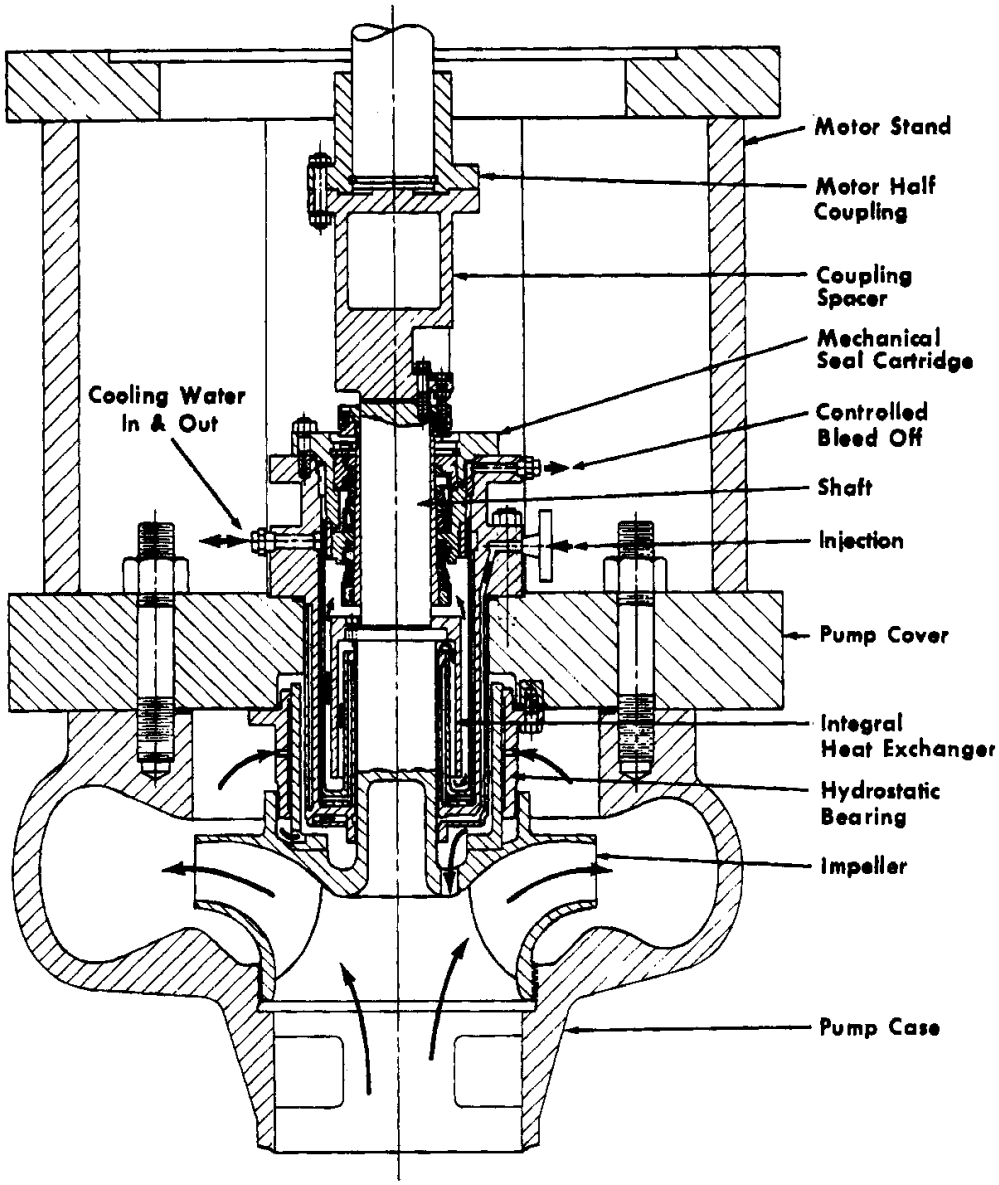


FIGURE 7 Main coolant pump (Flowserve Corporation)

In addition, a safety seal may also be installed as a backup. At least one manufacturer uses a high-pressure ceramic seal with a carbon seal backup followed by a vapor seal. Main coolant pump seals are usually supplied with reactor-grade water through a seal injection system. The advantage of using injection water is that it can be temperature controlled and filtered. Most pumps, however, can be operated without it for at least reasonable time periods.

PRINCIPAL TYPES OF PUMPS

Most of the pumps in nuclear service are one- or two-stage centrifugal motor-driven pumps. Both vertical and horizontal types are used. Charging, safety injection, feedwater, and other high-head pumps are usually multistage motor-driven centrifugal units. Some requiring high power are turbine driven. Double-suction designs are frequently used for RHR pumps, where service requires operation at low available NPSH. Reciprocating pumps find limited service in nuclear plants for makeup flow, seal injection flow, or

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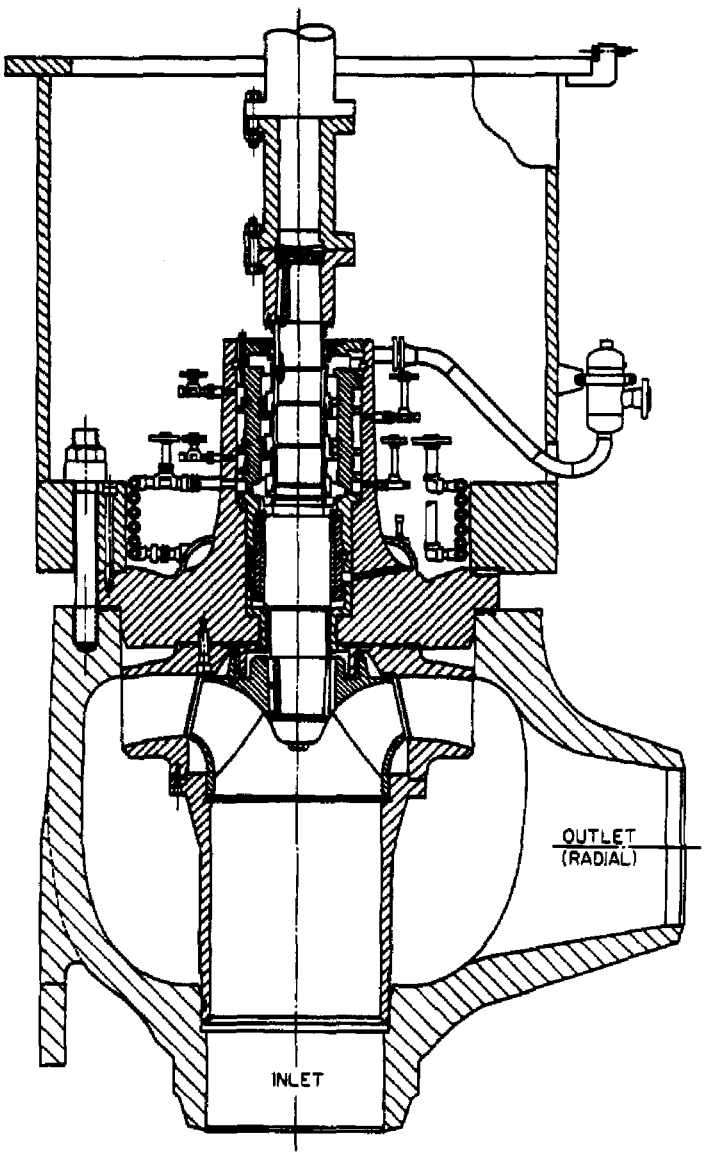


FIGURE 8 Main coolant pump (Sulzer Pumps)

chemical mixing service. Canned pumps are frequently used in subsystems where their zero-leakage capabilities can be exploited.

Figure 12 shows a multistage centrifugal pump used in charging and safety injection service.

Figure 13 illustrates a reciprocating pump often used for charging and hydrostatic test service. It is rated at 98 gpm (22 m^3/h) at 5800 ft (1768 m) head. In Figure 14, a vertical multistage unit is shown. This type of unit is common in heater drain service. Figure 15 shows a single-stage pump of a type used in many service functions in a nuclear plant. Typically, its flow is 75 gpm (17 m^3/h) with a developed head of 235 ft (71.6 m) at 3500 rpm.

SPECIAL REQUIREMENTS FOR NUCLEAR SERVICE PUMPS

It is in the area of special requirements that nuclear service pumps differ most widely from commercial products. These special requirements, described in greater detail later, far exceed the requirements of the general industrial field and illustrate the strong emphasis placed upon pressure integrity and pump operability.

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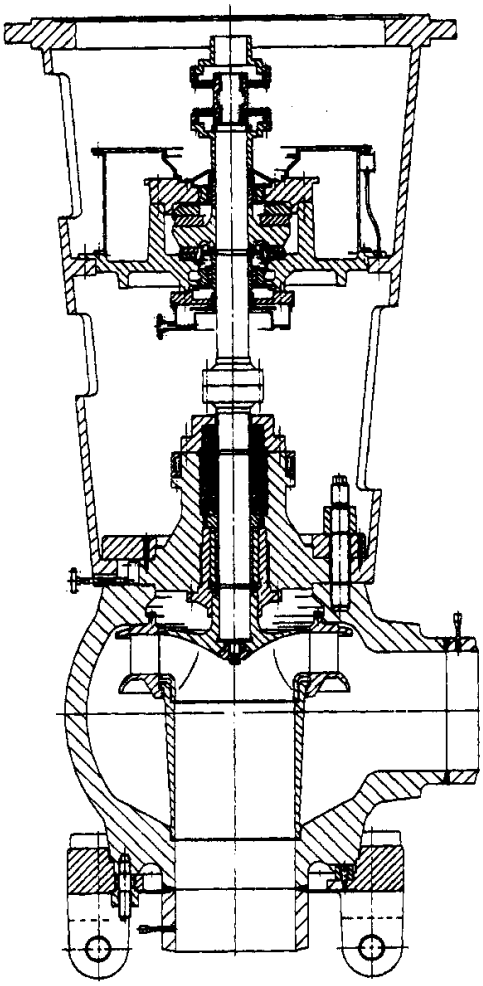


FIGURE 9 Main coolant pump (CE-KSB Pump)

Nuclear-grade pumps are designed, built, inspected, tested, and installed to rigid standards of the U.S. Nuclear Regulatory Commission, the American Society of Mechanical Engineers, the American National Standards Institute, and other regulatory agencies such as state and local jurisdictional bodies. Established rules can be divided into two distinct categories: (1) those controlling integrity of the pressure-retaining boundary and (2) functional considerations.

The hydraulic design of nuclear pumps is the same as that of pumps in conventional service. For recirculation and reactor coolant pumps, a radial discharge is preferred by some users because it tends to simplify certain aspects of the plant design.

Vibration characteristics of pumps in nuclear service are especially important because of the relative inaccessibility of the equipment for checking and servicing and because safety requirements permit only limited outage of pumps; otherwise, the plant must come to standby condition.

An analytical or experimental determination of lateral and torsional natural frequencies is routine for most pump-driver combinations, and occasionally a transient analysis may be required.

Design under ASME Code Rules Under the rules established by the ASME Boiler and Pressure Vessel Code, Section III (Nuclear Components), the owner, such as the plant designer or public utility, either directly or through an agent, prepares a design specification for a specific pump, which in code terms is a component. This specification must be approved by a licensed professional engineer and list the applicable ASME Code Year and Addenda.

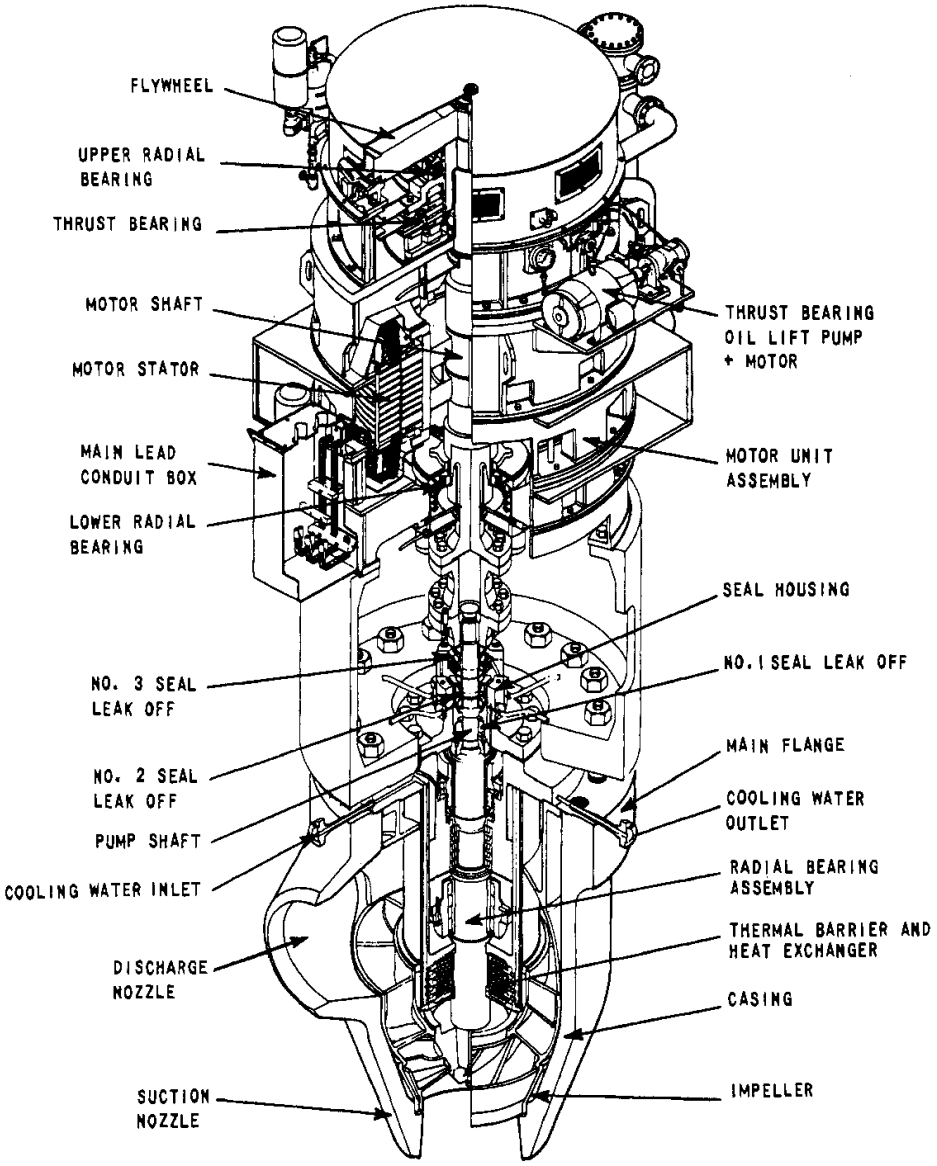


FIGURE 10 Main coolant pump (Westinghouse Electric)

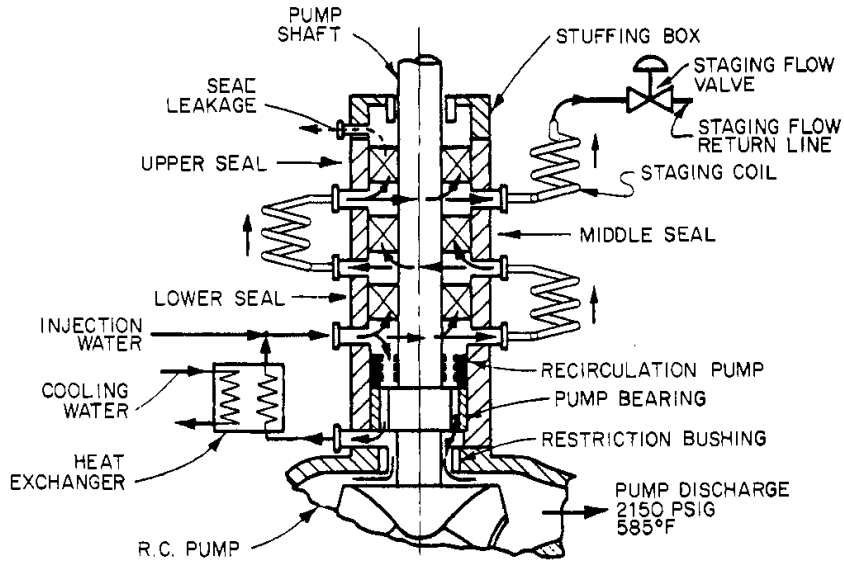


FIGURE 11 Flow schematic for a reactor coolant pump seal (Sulzer Pumps)

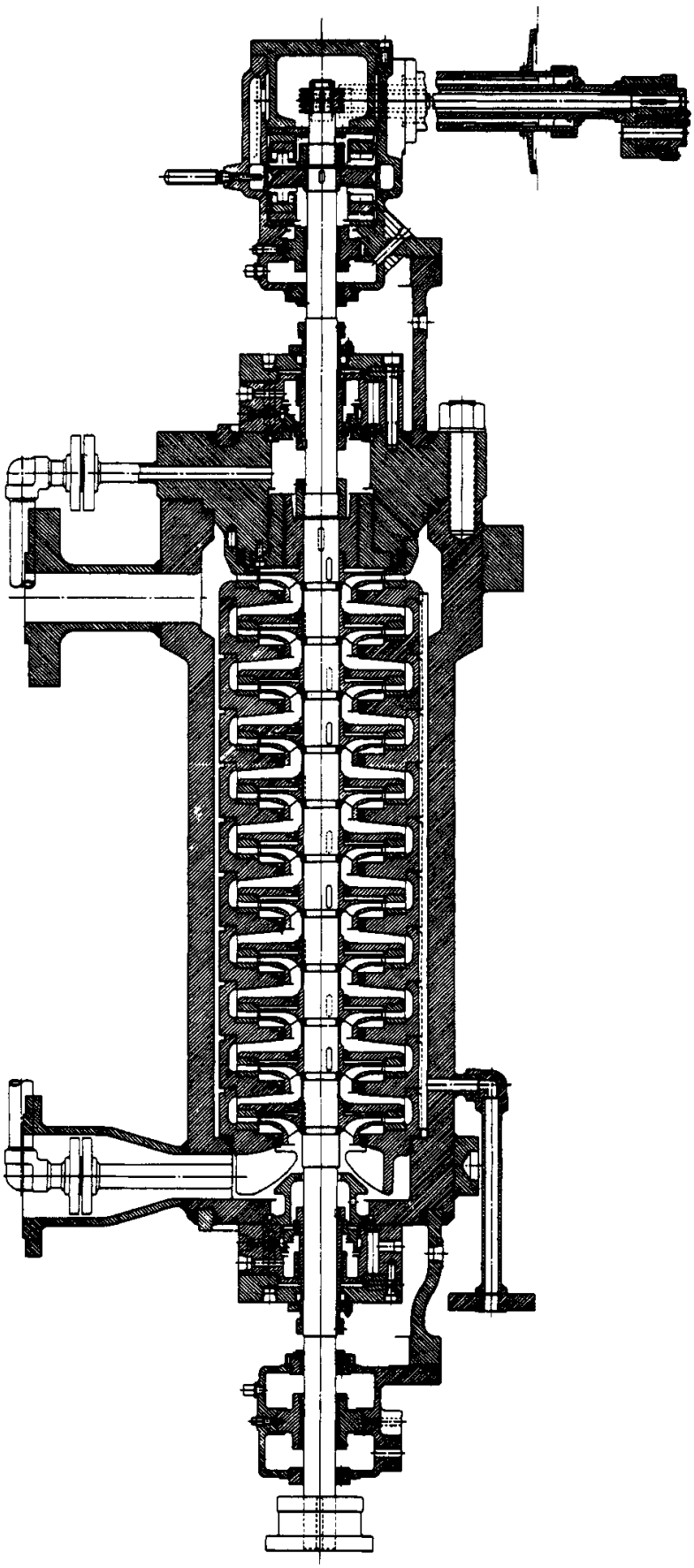


FIGURE 12 Pump for PWR charging and safety injection service (Flowserve Corporation)

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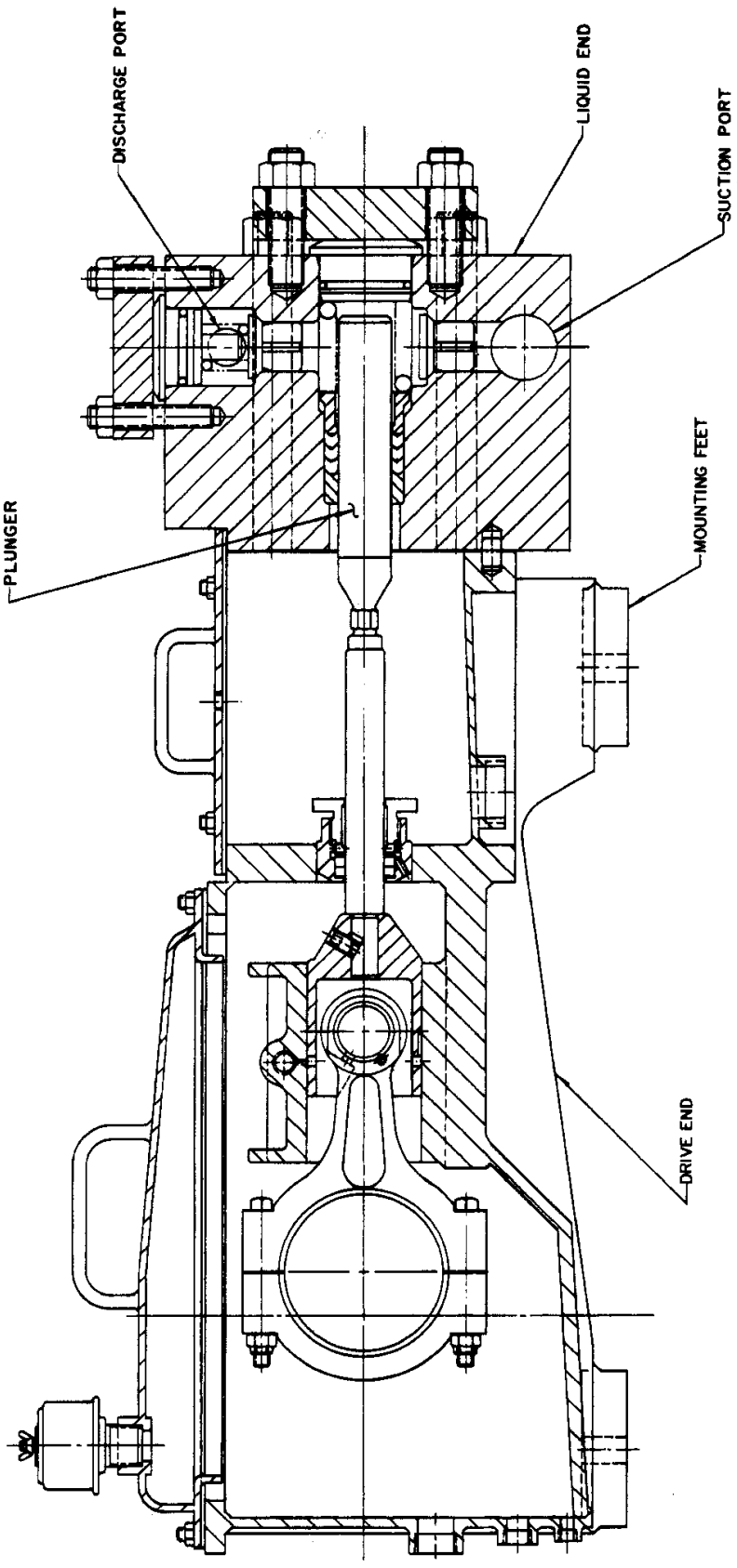


FIGURE 13 Reciprocating pump for nuclear service (Gaulin)

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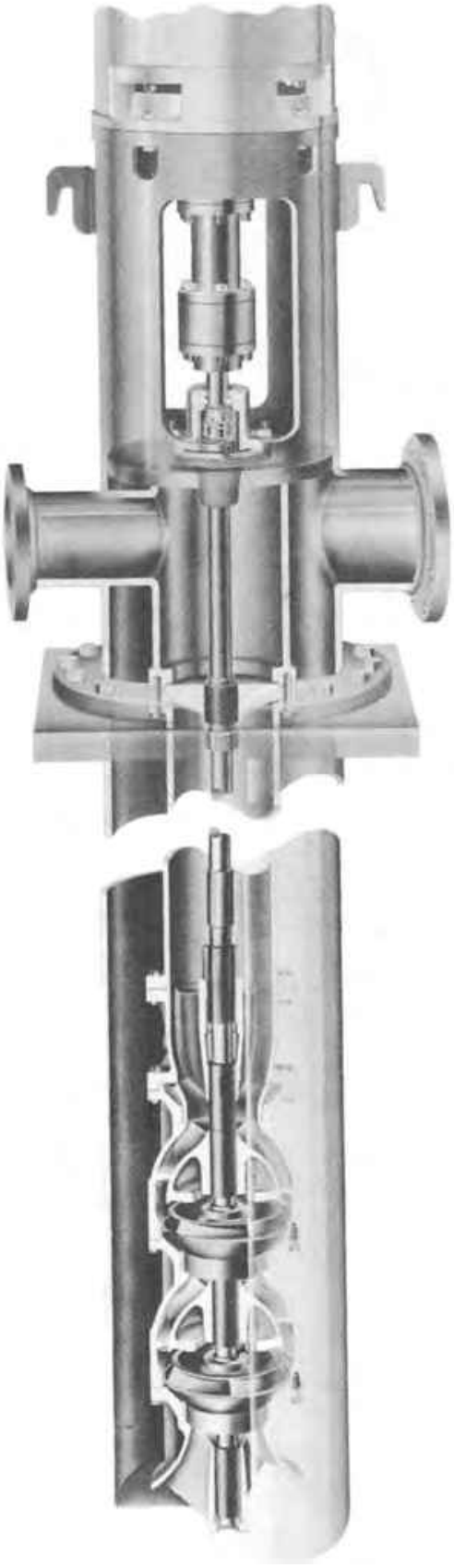


FIGURE 14 Vertical multiple-stage pump (Flowserve Corporation)

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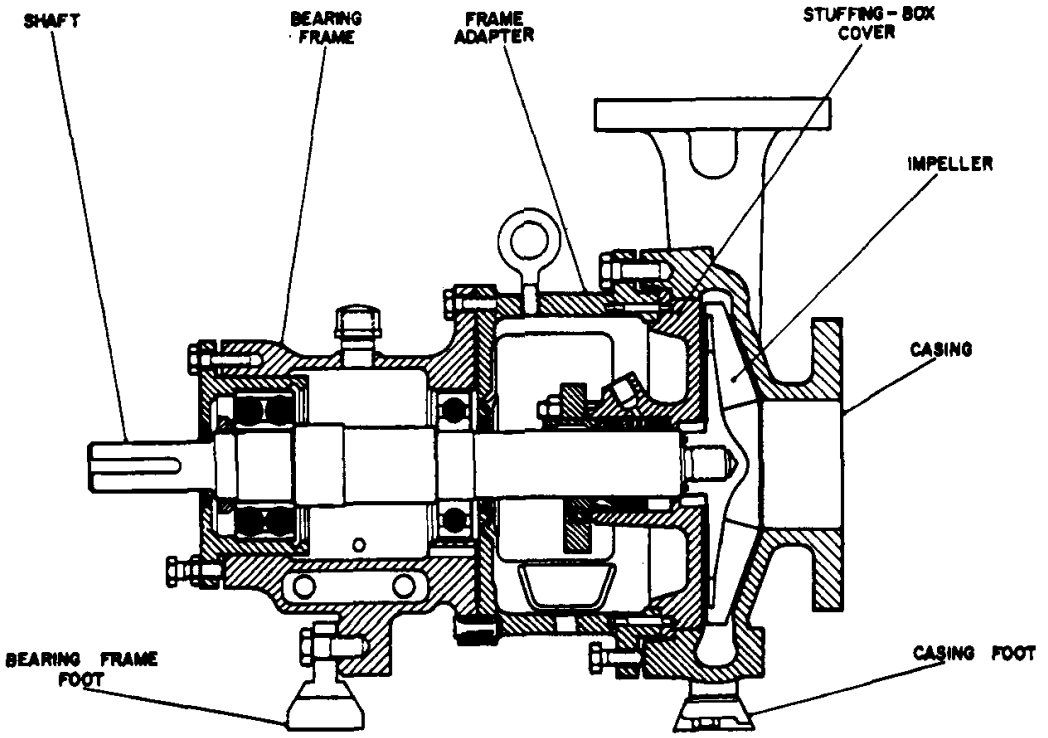


FIGURE 15 Single-stage pump used for many service functions in a nuclear plant (ITT/Gould Pumps, Inc.)

The manufacturer builds the pump to the design specification. Verification is provided through the combined efforts of the material supplier, manufacturer, insurance inspector, and state and local enforcement agencies.

The design specification requires compliance with the rules of the code with regard to the design of the pressure boundary and, in addition, includes supplementary requirements prescribed by the owner. Other standards are invoked as applicable to meet the safety and environmental requirements of the U.S. Nuclear Regulatory Commission (NRC). Functional needs may be included, and the code class to which the pump is to be built must be identified.

There are three ASME code classes for pumps. For the most critical service, a Class 1 pump is specified. Class 2 represents a pump serving a less critical system, and Class 3 is the lowest-class pump for nuclear service. It is the owner's responsibility to establish the pump class, with guidance provided by the Nuclear Regulatory Commission and the manufacturers.

For the pressure boundary evaluation, a Class 1 pump, by code rules, receives the most detailed analysis using verified and validated modern design techniques. These techniques must be supported, if necessary, by experimental stress analysis, and a certified Design Report must be submitted to the customer to document adherence to the code. Fatigue analysis of critical portions of the pump may be required, and behavior under all plant conditions, including accident, must be evaluated. Class 2 pumps require less analysis, but a certified design report is still required, documenting compliance with the customer's specification and the code. For Class 3 pumps, even wider latitude of design is permitted, but a certified design report must still be submitted to the customer. All three classes of pumps are to receive an ASME N-stamp by a qualified pump manufacturer upon successful completion of design, manufacture, inspection, test, and document review.

Additionally, nuclear service pumps are usually examined in design for thermal steady state and transient conditions, behavior under seismic disturbances, conditions of nozzle loadings imposed by system piping, means of support, and accessibility for service, in-service inspection, and replacement.

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Rules for quality control during material procurement, manufacture, and test are also contained in the code. Nondestructive examination and document control are detailed, and a quality assurance program requiring a formal procedure manual must be prepared and implemented by the manufacturer. Periodic surveys by the ASME verify adherence to code rules. Local jurisdictional authorities provide day-to-day inspection services as required by the manufacturer.

Design of Noncoded Parts Parts of the pump that are not classified as pressure boundary items or attachments to the pressure boundary are not covered by the ASME code. The owner's design specification may describe applicable requirements for these nonpressure parts, referencing other documents or excluding use of certain objectionable materials. If the pump is in critical service or is needed in emergency situations, certain ANSI standards may be invoked. Accompanying these rules will be extensive quality assurance and documentation to demonstrate compliance.

MATERIALS OF CONSTRUCTION

Material Limitations When pumps built to ASME standards are required, materials for the pressure-retaining parts must be selected from a list approved by the ASME. Section III (Nuclear Components) of the ASME Boiler and Pressure Vessel Code lists these materials, their allowable stresses, and the examination requirements that must be applied to ensure their suitability.

Typical Materials Examples of acceptable materials for pressure-retaining boundary parts are shown in Table 3. They are suitable for all three ASME code classes; however, other acceptable materials may be restricted to use for a particular class. Hazardous and porous materials are generally avoided, as are materials such as cobalt, which, though normally harmless, may become hazardous from radioactive considerations (see following text). Cobalt content is often limited in large stainless steel parts but is usually permitted in concentrated form of small areas, for example, where hard-facing is required.

Materials of construction should not be affected by the usual decontamination chemicals. Nonpressure boundary parts may be made of conventional materials but will usually require the buyer's approval. Certain elastomers, such as ethylene propylene, which has good radiation stability, are excellent for seal parts. Many fine grades of carbon-graphite are available for water-lubricated bearings and for mechanical seal facings.

TABLE 3 Materials for pressure-retaining boundary parts

Carbon steel	
Castings	SA-216, Gr WCA, WCB, WCC
Forgings	SA-105, Gr I, II
Plate	SA-515, Gr 55, 60, 65, 70
Bolting	SA-193, Gr B6, B7, B8, B16
Stainless steel	
Castings	SA-351, Gr CF8 (304), CF8M (316)
Forgings	SA-182, Gr 304, 316, 321, 347
Plate	SA-240, Gr 304, 316, 321, 347
Nonferrous	
A limited number of nonferrous materials are permitted	

CONSIDERATIONS OF RADIOACTIVITY

Radioactivity may become a serious consideration in the design of nuclear pumps because of the need for servicing the equipment. The water used in the primary system becomes contaminated with metallic elements through solubility, corrosion, and erosion. When circulated through the core region, the metallic elements become radioactive because of interaction with neutrons. These radioactive contaminants may, if soluble, remain in solution in the water or, if insoluble, plate out on metal surfaces or become lodged in "crud traps," such as fit interfaces, screw threads, porous base metals, extremely rough surfaces, cracks, and certain types of weld configurations, such as socket welds. In one case, for instance, a pump impeller returned for overhaul defied attempts at decontamination until it was discovered that a repair had been made to a presumably integral wear ring by undercutting and shrinking on a new ring. The interface was barely perceptible, but once it had been found and the ring had been removed, the impeller was readily decontaminated.

Soluble contaminants are most easily removed by providing the pump with complete drainage features; that is, leaving no internal pockets that are not naturally drainable. Ease and speed of parts replacement are, therefore, also important items of design because they reduce the length of time service personnel are exposed to radiation.

The NRC provides specifications detailing the allowable radiation exposure to personnel. Any pump that produces radioactivity at rates that would exceed these limitations must either be repaired at a facility licensed to handle contaminated material or be decontaminated prior to being sent to a conventional repair shop.

After prolonged service in a nuclear plant, pumps may emit radioactivity at a rate in excess of 2 to 50 rem/h, far in excess of the NRC limits. Therefore, provisions to decontaminate nuclear pumps become an important design characteristic. Although not all nuclear pumps operate in highly radioactive environments, these pumps will usually require some degree of decontamination before they can be freely handled and repaired.

SEISMIC DESIGN

The ASME code requires that a seismic analysis be performed on all classes of nuclear pumps. Refer to Section 12.6.2 for more detailed information on this subject.

TESTING

Hydrostatic Testing Hydrostatic testing in accordance with ASME code rules is conventional except that specific documentation is required.

Performance Testing Performance of nuclear pumps is verified by procedures common to nonnuclear pumps. In addition, however, it may be necessary to demonstrate pump performance under some emergency condition, such as loss of cooling water or seal injection water. Also, a test under simulated total-loss-of-power conditions may be required, in which the reactor coolant pump must coast safely to a stop and withstand loss of both cooling and seal injection water for a finite time.

Periodic Testing The ASME code, under its rules for in-service inspection (Section XI), requires periodic testing of certain pumps installed in a nuclear plant. The pumps affected by these rules are those associated with the safety systems and those that may be required to function during an emergency or a reactor shutdown. Examples of such pumps are those for core spray, residual heat removal, boron injection, and containment spray. If pumps cannot be tested in their usual circuit, they must be supplied with bypass loops that can be valved off.

The field testing procedure involves obtaining a baseline set of values for head, flow, speed (if variable), vibration, inlet pressure, and bearing temperatures. The flow quantity that sizes the bypass loop is not specified in the code, but in the practical sense, the flow must be adequate to prevent overheating of the pump and overloading of the bearings. Also, because it is an off-design point, consideration is given to the potential for rough operation due to impeller internal recirculation and possible low-flow cavitation damage. Code rules call for a 5-min running period every 3 months and longer periods where it is necessary for bearing temperatures to stabilize. If the differences between periodic operating data and baseline data exceed permissible limits, the cause for the differences must be sought and corrected. Record-keeping is therefore a significant part of pump in-service testing.

12.6.2
NUCLEAR PUMP SEISMIC QUALIFICATIONS

D. NUTA
W. W. PARRY, JR.

Nuclear electric generating stations and their components shall be designed to resist the dynamic forces that could result from an earthquake at the site. This section covers the seismic qualification requirements for pumps classified according to function and importance based on safe operation and public safety.

SEISMIC CLASSIFICATION OF PUMPS

Pumps that must withstand a safe shutdown earthquake (SSE) and remain functional during and after such an event are designated seismic Category I. Pumps that are not required to remain functional during SSE are designated nonseismic Category I and may be classified as follows:

- Nonseismic Category I pumps are located so their failure or structural deformation might keep a seismic Category I piece of equipment or system from performing its intended function. The analysis methods and seismic input used on nonseismic Category I pumps to show that gross failure or excessive deformations will not occur are identical to those used for seismic Category I equipment.
- Nonseismic Category I pumps are located so their failure or structural deformation cannot affect any seismic Category I system or component. Seismic-related requirements for these pumps, if any, are established using a less stringent seismic environment with the aim of minimizing possible replacement expenses following a credible event.

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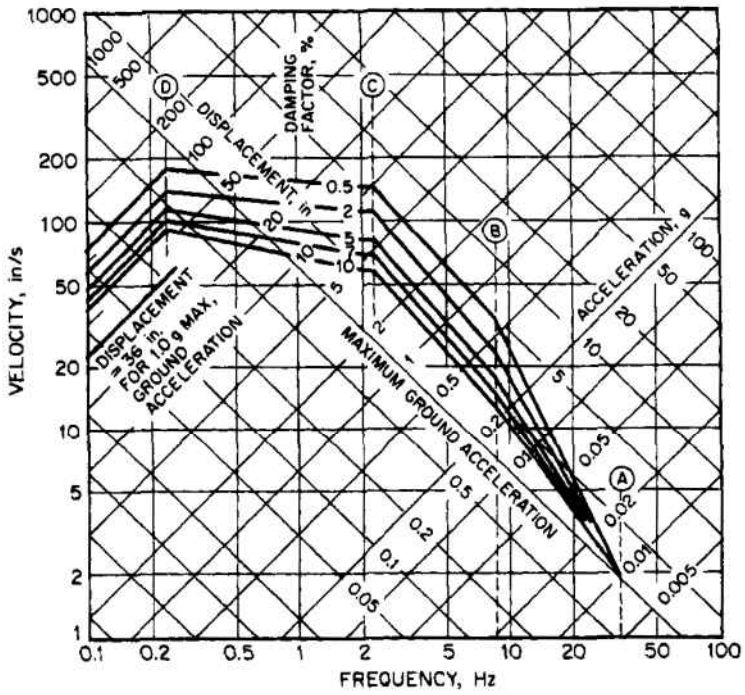


FIGURE 1 Horizontal design response spectra scaled to 1*g* ground acceleration (1 in = 2.54 cm; g = 32.17 ft/s² = 9.807 m/s²)

DEFINITIONS

Critical Damping This is the minimum damping for which a vibrating system has no vibratory motion. For analysis purposes, damping is expressed as a percentage of critical damping. The amplification of input motion, or magnitude of response, is inversely proportional to the damping ratio of the vibrating system. Figure 1 shows responses obtained for damping ratios, which vary from 0.5 to 10% of critical damping.

Damping Damping is the dissipation of energy in a vibrating system. Damping is a function of many factors, including material characteristics, stress levels, and geometric configurations and takes place as a result of the transformation of input energy in the form of seismic motion to heat, sound waves, or other energy forms. For analysis purposes, damping is assumed to be viscous; that is, the damping force is proportional to velocity and acts in the opposite direction.

Degrees of Freedom The degrees of freedom are the coordinate(s) necessary to adequately describe the behavior of a structure or component. A node, or nodal point, may have as many as six degrees of freedom, consisting of three translations and three rotations (such as Δ_x, Δ_y, Δ_z, θ_x, θ_y, θ_z, as shown in Figure 2). This complex representation in terms of degrees of freedom is used when performing static analysis to obtain forces, stresses, or displacements.

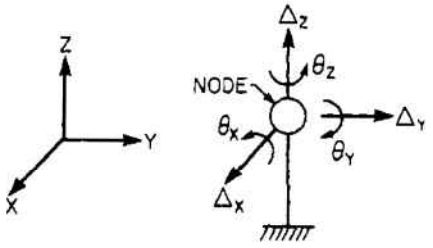


FIGURE 2 Nodal degrees of freedom

Previous Page

Finite Element Method The finite element method of analysis involves the idealization of a structure or component, such as a pump, as an assemblage of elements of finite size. This mathematical model can be a lumped-mass model, a shell element model, a three-dimensional solid model, or a combination of all three. Figures 3 and 4 contain typical shell element models of horizontal and vertical pumps respectively.

Floor Acceleration This is the acceleration at a structural floor or any acceleration applicable at the equipment mounting location.

Ground Acceleration Ground acceleration is the acceleration of the ground associated with earthquake motion.

Modal Analysis Modal analysis is a type of dynamic analysis in which the response of a vibrating system is obtained as a combination of responses of the system's mode shapes.

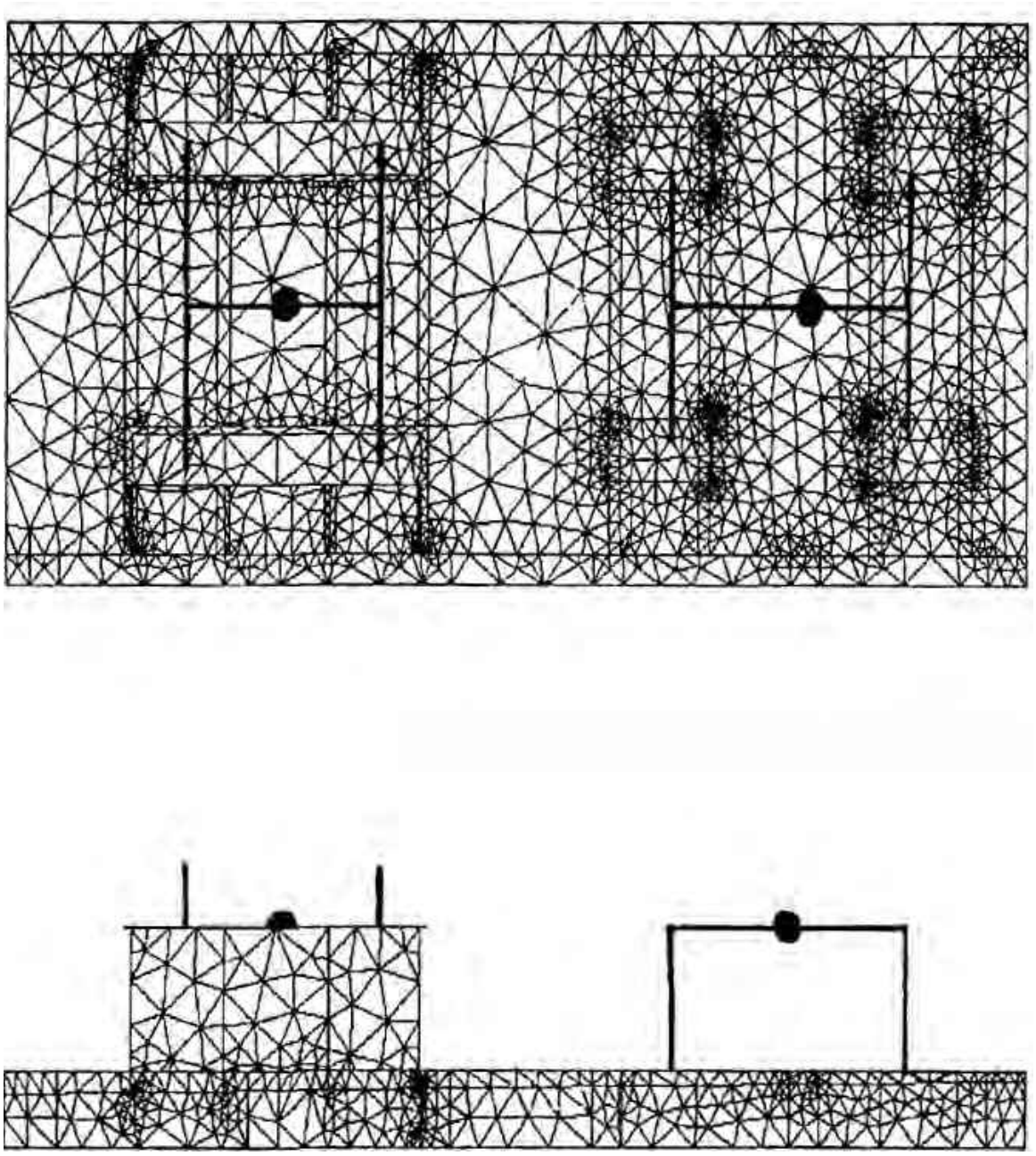


FIGURE 3 Typical FEM mesh of horizontal pump/motor assembly pump and motor are modeled as lumped-mass elements whereas bedplate is modeled with shell elements.

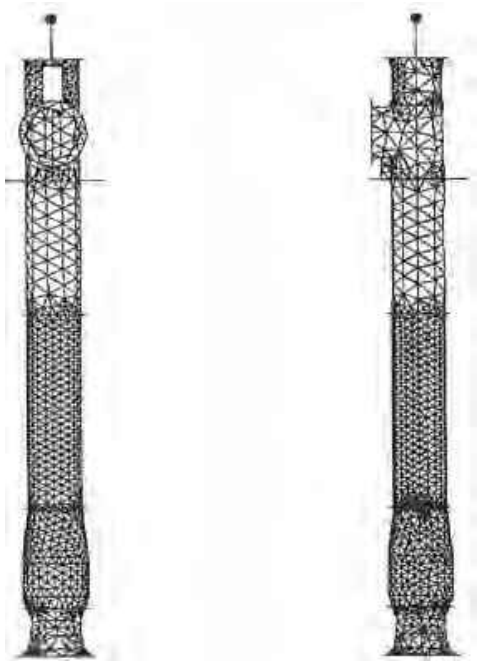


FIGURE 4 Typical FEM mesh of vertical circulating water pump. The pump is modeled with shell elements while the motor is modeled with lumped-mass elements.

(Specific requirements on the combination of modal responses is contained in U.S. Nuclear Regulatory Commission Guide 1.92.) Modal analysis may be used in conjunction with a response spectrum or time history analysis.

Natural Frequency The natural frequency is the frequency at which a body vibrates due to its own physical characteristics and to the elastic restoring forces developed when it is displaced in a given direction and then released while restrained or supported at specific points.

Operating Basis Earthquake (BE) An operating basis earthquake is an earthquake that could reasonably be expected to affect a plant site during the operating life of the plant. The systems and components necessary for continued operation of a nuclear plant without undue risk to public health and safety are designed to remain functional when subjected to the vibratory ground motion produced by an operating basis earthquake.

Required Response Spectrum (RRS) The RRS is the response spectrum for which the equipment (pump assembly) has to be dynamically qualified.

Response Spectrum The response spectrum is the plot of the maximum response (acceleration, velocity, displacement) of a series of damped single-degree-of-freedom (SDF) oscillators versus the natural frequencies (or, sometimes, periods) of the oscillators when subjected to the vibratory motion input (time history) at their supports. Figure 5 represents a plot of response spectra obtained for 0, 2, 5, and 10% of critical damping with an earthquake time history as input. The plot is on tripartite log paper, and only a limited number of SDF oscillators are shown (usually, more than 70 are needed to correctly define a spectrum).

In order to facilitate the understanding of how to use the Figure 5 plots, the 0% of critical damping response spectrum is presented in Figure 6. Assuming an SDF oscillator having a frequency of 2 cycles per second (Hz), the maximum displacement of the mass point when subjected to the earthquake record is 7 in (17.8 cm), the maximum velocity is 50 in/s (1.27 m/s), and the maximum acceleration is $2g$, where g = gravitational acceleration = 32.2 ft/s^2 (9.807 m/s^2). A line is drawn perpendicular to the frequency axis, and from the intersection of this line with the spectrum curve, a line parallel to the acceleration lines

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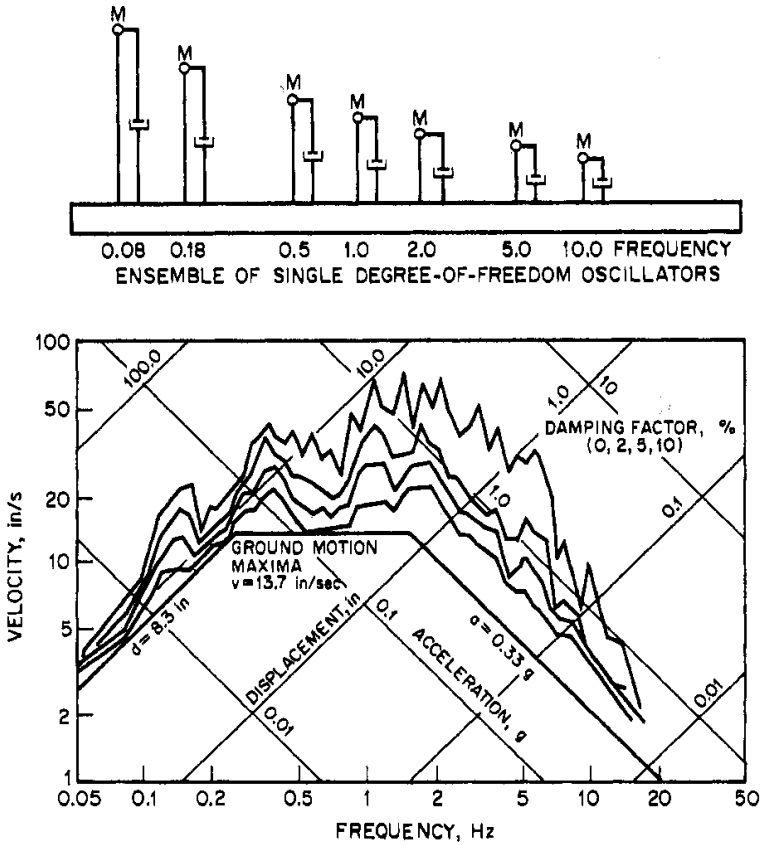


FIGURE 5 Response spectra plots for various dampings (1 in/s = 0.0254 m/s; 1 in = 2.54 cm; 1g = 32.17 ft/s^2 = 9.807 m/s^2)

will provide the acceleration a ≈ 2g, a line parallel to the displacement lines will provide the displacement d 7 ≈ in (17.8 cm), and a line perpendicular to the velocity axis will provide the velocity v ≈ 50 in/s (1.27 m/s).

In general, response spectra are presented as plots of acceleration versus frequency. Given a time history of acceleration, such as the one presented in Figure 7, the response spectrum is obtained as described previously. The response acceleration at frequencies above

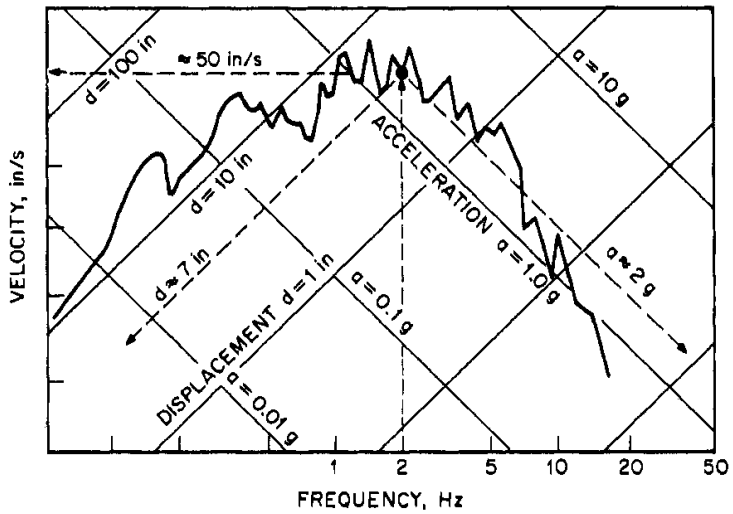


FIGURE 6 Response displacement, velocity, and acceleration versus frequency (1 in/s = 0.0254 m/s; 1 in = 2.54 cm; 1g = 32.17 ft/s^2 = 9.807 m/s^2)

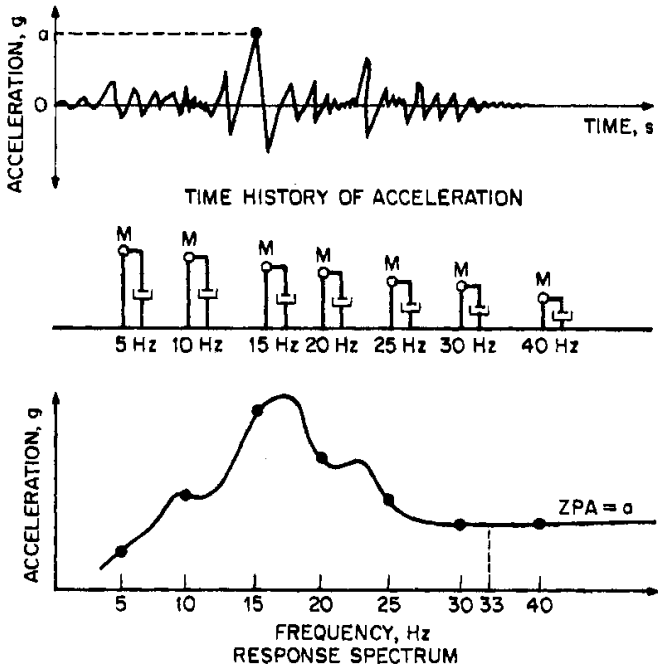


FIGURE 7 Development of acceleration response spectrum ($1g$ = 32.17 ft/s² = 9.807 m/s²)

33 Hz (also called zero period acceleration) is equal to the peak acceleration of the time history record regardless of the damping ratio used (the value a is specified in Figure 7).

Response Spectrum Analysis The response spectrum analysis is a modal dynamic analysis that uses response spectra as vibratory input at the pump attachment points.

Safe Shutdown Earthquake (SSE) An SSE is an earthquake that produces the maximum vibratory ground motion for which certain structures, systems, and components are designed to remain functional. These structures, systems, and components are those necessary to assure: (1) the integrity of the reactor coolant pressure boundary, (2) the capability to shut down the reactor and maintain it in a safe shutdown condition, or (3) the capability to prevent or mitigate the consequences of accidents that could result in potential offsite exposures to radioactive material. Various safety classes are assigned to structures, systems, and components as a function of which of these three categories they belong to.

Single-Degree-Of-Freedom Oscillator (SDF) The SDF is a vibrating system consisting of a single node point where the mass is concentrated and supported by a beam and spring element. If the single dynamic degree of freedom is along the axis of the beam, the frequency of the oscillator is

$$f = \frac{1}{2\pi}\sqrt{\frac{K}{M}}$$

where f = frequency, cycles per second or Hz

$K = EA/L$ = spring stiffness, lb/in (N/mm)

E = Young's modulus of elasticity of support beam, lb/in² (Pa) A = cross-sectional area of support beam, in² (mm²)

L = length of support beam in (mm)

M = mass lb (s²/in (N · s²/mm)

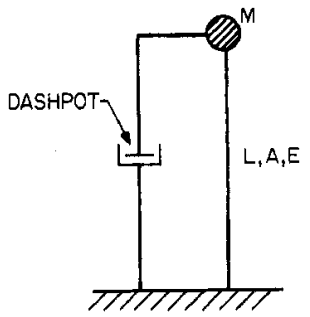


FIGURE 8 Single-degree-of-freedom oscillator

With this idealization, the frequency of the SDF oscillator may be varied by changing the length of the beam element and maintaining *E, A,* and *M* constant (Figure 8). A horizontal degree of freedom would be used in conjunction with beam bending and shearing.

Damping effects are simulated by a dashpot, which can be assigned different damping ratios. Although there is little effect on the frequency of the SDF oscillator when the dashpot is assigned damping ratios below 10 or 15% of critical damping, there is a significant effect when higher values are used. Thus, if the dashpot is assigned 100% of critical damping, the oscillator will behave as a rigid body, duplicating, without amplification, the support input motion.

Test Response Spectrum (TRS) The TRS is a response spectrum that is either theoretically constructed or derived using spectrum analysis based on the actual motion of the shake table (during dynamic testing).

Time History Analysis The time history analysis is an analysis performed in the time domain using time history vibratory input. A direct integration or modal analysis may be performed.

Time History of Acceleration The time history of acceleration is a variation of acceleration as a function of time during the postulated duration of the seismic event.

Zero Period Acceleration (ZPA) The ZPA is a response acceleration at frequencies above 33 Hz.

Although these definitions will facilitate the understanding of the seismic considerations discussed herein, the reader is encouraged to review the works listed at the end of this section for a more complete list of terms and definitions used in the process of dynamic qualifications.

SEISMIC QUALIFICATION REQUIREMENTS

The seismic qualification of seismic Category I pumps is performed to demonstrate the ability of the equipment to perform its intended function during and after the time it is subjected to an SSE, which could occur before or after several OBEs. The value of the SSE or OBE response spectra shall be supplied by the equipment buyer (nuclear facility). The simultaneous effects of the seismic accelerations in the two orthogonal horizontal directions and the vertical direction have to be considered.

The seismic qualification can be achieved by

- Analytical methods (static or dynamic analysis)
- Testing the equipment under simulated seismic conditions
- Using a combination of tests and analysis

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The information that is provided in the IEEE 344 standard on these qualification methods is applicable to pump assemblies. The information presented hereafter either emphasizes some of the IEEE 344 requirements or is pertinent to pumps and pump assemblies in particular.

Qualification by Analysis The initial step in qualifying a pump assembly by analytical methods is to determine the rigidity of the assembly. If the assembly has its lowest natural frequencies greater than the ZPA frequency, typically 33 Hz, the assembly is considered rigid. On the other hand, if one of its natural frequencies falls below the ZPA, the assembly is considered flexible. Therefore, prior to conducting the qualification analysis, the natural frequency of the pump assembly must be determined by either a modal analysis or an equipment "bump" test. The FEM models as shown in Figures 3 and 4 can be used for the modal analysis.

After the rigidity of the pump assembly has been determined, the type of qualification analysis can be selected. The allowable types of analysis are:

- **Static analysis** If it can be shown that the equipment is rigid, a static seismic analysis may be performed to determine the stresses and deformations due to the dynamic seismic loads. The dynamic forces shall be determined by multiplying the mass of the assembly times the maximum floor acceleration (ZPA from the response spectra). These forces are then applied through the center of gravity of the assembly. The resulting stresses from each acceleration force are combined by taking the square root of the sum of the squares (SRSS) to yield the total dynamic stress. The dynamic deflections shall be calculated in the same manner.
- **Simplified dynamic analysis** If it has been determined that the equipment is flexible and the customer specification permits, a simplified dynamic analysis may be completed applying the same method as the static analysis but using different values for the dynamic floor accelerations. The applicable floor acceleration shall be obtained by multiplying the acceleration values corresponding to the fundamental natural frequency from the appropriate response spectra curve by 1.5. If the fundamental natural frequency is not known, the maximum peak value of the response spectras shall be multiplied by 1.5. The 1.5 factor will conservatively account for possible participation of higher modes.
- **Detailed dynamic analysis** When justification for a static analysis cannot be provided, a detailed dynamic analysis is required, unless a conservative factor is used to account for the participation of higher modes (simplified dynamic analysis). A mathematical model of the equipment is required to determine the dynamic behavior of the equipment. This mathematical model may be a lumped-mass model, a shell model (see Figure 4), a solid element model, or a combination such as shown in Figure 3. The model should include, when applicable, the hydrodynamic mass that represents the contained water or the effects of submergence on vertical pumps. This model is then analyzed using either response spectra modal analysis or time-history (modal or step-by-step) analysis. The various modal contributions shall be combined by taking the SRSS of the individual modal stress and deformations.

These dynamic analyses are performed with computer programs in the public domain. Such programs include ANSYS, MSC/NASTRAN, ALGOR, and COSMOS/M, which incorporate acceptable methods of combining modal responses, or with a multitude of other finite element programs that are either commercially available or developed specifically for the analysis. Further information regarding specific items related to dynamic analysis, such as damping ratios, combination of modal responses, and modeling techniques may be obtained from the works listed at the end of this section. Although recent advancements in computer technology and software have made this type of analysis more readily available, when performing detailed dynamic analyses, a thorough understanding, obtained from experience, is imperative.

After these seismic stresses and deformations have been obtained, they must be combined with the other equipment stresses and deformations, resulting from all of the applicable loads, in order to determine the acceptability of the equipment.

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Qualification by Testing The equipment is tested by using the applicable seismic information (floor response spectrum or time history). The input test motion should have a TRS that closely envelopes the RRS, and the test should simulate the occurrence of five OBEs followed by one SSE.

Both the pump and the motor are tested under conditions that simulate as closely as possible actual operating conditions and operability. The operability of the pump, which is tested without the piping system attached and in the absence of any fluid, is established by the absence of undue large deflections during testing (which implies low stress levels) and by performance at the end of the test.

The input motion used must have a maximum acceleration equal to or greater than the ZPA and cannot include frequencies above the frequency associated with the ZPA. The input motion is applied to one vertical and one horizontal axis (resultant) simultaneously unless it can be proved that coupling effects are negligible. In general, multiaxis testing using independent random inputs is required.

Unless it has complete symmetry about its vertical axis, the equipment is rotated 90° and retested.

The mounting and support details used during testing should simulate the service mounting as closely as possible and should be designed so the mounting will not cause dynamic coupling to the test assembly or alter the input motion.

Combination of Test and Analysis If the motor assembly is tested separately and the pump is qualified by analysis, RRS at the motor assembly attachment points are developed from the pump assembly seismic analysis and used in the motor assembly testing. If both the pump assembly and the motor have to be tested but it is not feasible to test them together, dummy weights are used to replace the assembly part not included in the test and RRS at the location are established and used in testing the missing part.

DESIGN AND FUNCTIONAL REQUIREMENTS

The preceding sections provided the procedures for determining the seismic response of pumping equipment. However, in order to determine the acceptability of these results, they have to be combined with the pump's normal operational loads and compared to predetermined stress limitations. In accordance with the ASME code, the customer's design specification shall advise the applicable load combinations and allowable stress values.

Typically, the pump load combinations can be divided into four levels or service limits. Table 1 lists these service limits in ascending order dependent upon the magnitude of component deformation resulting from operation at the prescribed condition. In addition, Table 2 lists the ASME code allowable stress limits for each of these conditions. If no allowable stress limits are specified in the customer's design specification, no allowable stress increases should be used for the increasing load combinations, including those for OBE and SSE operation.

In addition to the specific requirements for stress levels or functionality of pressure containing components, the following items should be addressed as part of any seismic qualification of pump-motor assemblies:

- Secondary stress loading caused by differential movement of connected components should be determined.
- Displacements and deflections should be measured during testing or calculated, if qualified by analysis, in order to assure they are not excessive.
- Loads to be used in the design of the pump-motor assembly foundation and anchorage should be established.
- Seismic loads must be added to other normal shaft loads in order to verify stress levels, bearing loads, running clearances and possibly coupling misalignment.

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TABLE 1 Typical load combinations and stress limits

Plant Condition	Design/ Normal	Upset	Emergency	Faulted
Concurrent Loads	PD + SL	PD + SL + SOT	PD + SL + SOT + OBE	PD + SL + SOT + SSE
Design/Stress Limits	Level A	Level B	Level C	Level D

Where:

PD = Design pressure and temperature
SL = Sustained loads including dead weight and applied nozzle loads
SOT = System operating transients for upset, emergency, and faulted conditions per customer specifications
OBE = Operating basis earthquake
SSE = Safe shutdown earthquake
Level A = Those loadings that the pump may be subjected to in the performance of its specified function
Level B = Those loadings that the pump must withstand without damage requiring repair
Level C = Those loadings that may result in large deformations that necessitate the removal of the pump from service for inspection or repair
Level D = Those loadings that result in gross general deformations with some consequent loss of dimensional stability and damage requiring repair, which may require removal of the unit from service

TABLE 2 Stress limits

Service limits	Stress limits (maximum normal stress)
Design and Level	$\sigma_m \leq 1.0S$
	$(\sigma_m \text{ or } \sigma_L) + \sigma_b \leq 1.5S$
Level B	$\sigma_m \leq 1.1S$
	$(z\sigma_m \text{ or } \sigma_L) + \sigma_b \leq 1.65S$
Level C	$\sigma_m \leq 1.5S$
	$(\sigma_m \text{ or } \sigma_L) + \sigma_b \leq 1.8S$
Level D	$\sigma_m \leq 2.0S$
	$(\sigma_m \text{ or } \sigma_L) + \sigma_b \leq 2.4S$

σ_m = General membrane stress; equal to average stress across solid section under consideration; excludes discontinuities and concentrations and is produced only by pressure and other mechanical loads
σ_L = Local membrane stress; same as σ_m except that σ_L includes effect of discontinuities
σ_b = Bending stress; equal to linearly varying portion of stress across solid section under consideration; excludes discontinuities and concentrations and is produced only by mechanical loads
S = Allowable stress, given in Tables I-7.0 and I-8.0 of the ASME code; corresponds to highest metal temperature of section during condition under consideration

SOURCE: ASME Boiler and Pressure Vessel Code, Section III, Nuclear Power Plant Components, Division I, Subsection NC, 1992, New York.

- For electric motors, deflections or component dislocations should be closely monitored in order to assure that:
  - the rotor will not grind into the stator.
  - the fan blades will not damage the winding insulation.
  - there is no loss of lubricant and no slipped winding or winding temperature detector.

DOCUMENTATION

Refer to IEEE 344 for a list of documents that might be required for dynamic qualification of the pump-motor assembly (by test or analysis) and for general guidance in preparing the pump assembly purchase specification.

FURTHER READING

U.S. Nuclear Regulatory Commission guides (available from NRC, Washington, D.C.).

Regulatory Guide 1.100, "Seismic Qualification of Electrical Equipment for Nuclear Power Plants."

Regulatory Guide 1.61, "Damping Values for Seismic Design of Nuclear Power Plants."

Regulatory Guide 1.70, "Standard Format and Content of Safety Analysis Reports of Nuclear Power Plants."

Regulatory Guide 1.89, "Qualification of Class 1E Equipment for Nuclear Power Plants."

Regulatory Guide 1.92, "Combining Modal Responses and Spatial Components in Seismic Response Analysis."

Industry Standards (Available from the Institute of Electrical and Electronics Engineers, New York.)

IEEE 323, "Qualifying Class 1E Equipment for Nuclear Power Generating Stations."

IEEE 344, "IEEE Recommended Practices for Seismic Qualification of Class 1E Equipment for Nuclear Power Generating Stations."

Other Another source that may be helpful comes from the American Society of Civil Engineers.

American Society of Civil Engineers. "Structural Analysis and Design of Nuclear Plant Facilities." Manuals and Reports on Engineering Practices, No. 58, New York, 1980.

SECTION 12.7
CHEMICAL INDUSTRY

GARY CLASBY
THOMAS C. SPENCE

SOLUTION CHARACTERISTICS

Chemical industries can be broadly defined as those that make, use, or dispose off chemicals. The pumps used in these industries are different from those used in other industries primarily in the materials from which they are made. Although cast iron, ductile iron, carbon steel, and aluminum- or copper-based alloys will handle a few chemical solutions, most chemical pumps are made of stainless steel, Hastelloy, the nickel-based alloys, or the more exotic metals, such as titanium and zirconium. Pumps are also available in carbon, glass, porcelain, rubber, and a variety of engineering polymers, including the thermoplastics, thermosets, epoxies, vinyl ester, and fluorocarbons. Each of these materials has been incorporated into pump design for just one reason—to eliminate or reduce the destructive effect of the chemical liquid on the pump parts.

Because the type of corrosive liquid to be pumped determines which of these materials is most suitable, a careful analysis of the chemical solution to be handled is the first step in selecting the proper materials for pump construction.

Major and Minor Constituents Foremost in importance in a study of the characteristics of any solution are the constituents of the solution. This means not only the major constituents but the minor ones as well, for in many instances the minor constituents will be the more important. They can drastically alter corrosion rates, and therefore a full and detailed analysis is most critical.

Concentration Closely allied to what the constituents are is the concentration of each. Merely stating "concentrated," "dilute," or "trace quantities" is basically meaningless because of the broad scope of interpretation of these terms. For instance, some interpret *concentrated* as meaning any constituent having a concentration of greater than 50% by weight, whereas others interpret any concentration above 5% in a like manner. Hence it is always desirable to cite the percentage by weight of each and every constituent in a given solution. This eliminates multiple interpretations and permits a more accurate evaluation. It is also recommended that the percentage by weight of any trace quantities be cited, even if this involves only parts per million. For example, titanium be completely suitable in a given environment in the absence of fluorides. If, however, the same environment contained even a few parts per million of fluorides, the titanium would suffer a catastrophic corrosion failure.

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Temperature Generalized terms such as *hot, cold,* or even *ambient* are ambiguous. The preferred terminology would be the maximum, minimum, and normal operating temperature. In general, the rate of a chemical reaction increases approximately two times with each 18°F (10°C) increase in temperature. Because corrosion can be considered a chemical reaction, the importance of temperature or temperature range is obvious.

A weather-exposed pump installation is a good illustration of the ambiguity of the term *ambient.* There could be as much as a 150°F (83°C) difference between an extremely cold climate and an extremely warm climate. If temperature cannot be cited accurately, the ambient temperature should be qualified by stating the geographic location of the pump. This is particularly important for materials that are subject to thermal shock in addition to increased corrosion rate at higher temperatures.

Acidity and Alkalinity More often than not, little consideration is given to the pH of process solutions. This may be a critical and well-controlled factor during production processing, and it can be equally revealing in evaluating solution characteristics for material selection. One reason the pH may be overlooked is that it generally is obvious whether the corrosive substance is acidic or alkaline. However, this is not always true, particularly with process solutions in which the pH is adjusted so the solutions will always be either alkaline or acidic. When this situation exists, the precise details should be known so a more thorough evaluation can be made. It is also quite important to know when a solution alternates between acidic and alkaline conditions because this can have a pronounced effect on materials selection. Some materials, although entirely suitable for handling a given alkaline or acidic solution, may not be suitable for handling a solution whose pH is changing.

Solids in Suspension Erosion-corrosion, velocity, and solids in suspension are closely allied in chemical industry pump services. Pump design is a very critical factor when the solution to be pumped contains solids. It is not uncommon for a given alloy to range from satisfactory to completely unsatisfactory in a given chemical application when hydraulic design is the only variable. Failure to cite the presence of solids on a solution data sheet is not an uncommon occurrence. The concentration of solids should be referred to as percent by volume or weight. This undoubtedly is the reason for many catastrophic erosion-corrosion failures because most corrosion resistant materials are soft and do not handle abrasion well.

Aerated or Nonaerated The presence of air in a solution can be quite significant. In some instances, it is the difference between success and failure in that it can conceivably render a reducing solution oxidizing and require an altogether different material for pump construction. A good example of this would be a self-priming nickel-molybdenum-alloy pump for handling commercially pure hydrochloric acid. This alloy is excellent for the commercially pure form of this acid, but any condition that can induce even slightly oxidizing tendencies renders this same alloy completely unsuitable. The very fact that the pump is a self-primer means that aeration is a factor to contend with, and extreme caution must be exercised in using an alloy that is not suitable for an oxidizing environment. The presence of air will not only affect the head-flow rate performance, but also the NPSHR. The maximum amount of air a conventional centrifugal pump can handle is approximately 5% by volume.

Transferring or Recirculating This item is important because of the possible buildup of corrosion product or contaminants, which can influence the service life of the pump. Such a buildup of contaminants can have a beneficial or deleterious effect, and for this reason it should be an integral part of evaluating solution characteristics.

Inhibitors or Accelerators Both inhibitors and accelerators can be intentionally or unintentionally added to the solution. Inhibitors reduce corrosivity, whereas accelerators increase corrosivity. Obviously, no one would add an accelerator to increase the corrosion rate on a piece of equipment, but a minor constituent added as a necessary part of a process may serve as an accelerator; thus the importance of knowing the presence of such constituents.

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Purity of Product Where purity of product is of absolute importance, particular note should be made of any element that may cause contamination problems, whether it be discoloration of product or solution breakdown. In some environments, pickup of only a few parts per billion of certain elements can create severe problems. This effect is particularly important in pump applications where velocity effects and the presence of solids can alter the end result, as contrasted with other types of process equipment where the velocity or solids may have little or no effect.

When a material is basically suitable for a given environment, purity of product should not be a problem. However, this cannot be an ironclad rule, particularly with chemical pumps.

Continuous or Intermittent Duty Depending upon the solution, continuous or intermittent contact can have a bearing on service life. Intermittent duty in some environments can be more destructive than continuous duty if the pump remains half full of corrosive during periods of downtime and accelerated corrosion occurs at the air-liquid interface. Stagnant conditions for certain chemicals can also cause pitting corrosion of some metals. Perhaps of equal importance is whether the pump is flushed or drained when not in service.

CORROSIVES AND MATERIALS

Metallic or Nonmetallic Materials for chemical industry pump applications can, in general, be divided into two very broad categories: metallic and nonmetallic. The metallic category can be further divided into ferrous and nonferrous alloys, both of which have extensive application in the chemical industry. The nonmetallic category can be further divided into natural and synthetic rubbers, polymers, ceramics and glass, carbon and graphite. These materials have definite application in the handling of heavy corrosives. In particular, polymers in recent years have gained widespread acclaim for their ability to handle chemicals. For a given application, a thorough evaluation of not only the solution characteristics but also the materials available should be made to ensure the most economical selection.

Source of Data To evaluate material for chemical pump services, various sources of data are available. The best source is previous practical experience within one's own organization. It is not unusual, particularly in large organizations, to have a materials group or corrosion group whose basic responsibility is to collect and compile corrosion data pertaining to process equipment in service. These sources should be consulted whenever a materials evaluation program is being conducted. A second source of data is laboratory and pilot-plant experience. Though the information from this source cannot be as valuable and detailed as plant experience, it certainly can be very indicative and serve as an important guide. The experience of suppliers can be a third source of information. Though suppliers cannot hope to provide data on the specific details of a given process and the constituents involved, they normally can provide assistance and materials for test to facilitate a decision. Technical journals and periodicals are a fourth source of information. A wealth of information is contained in these publications, but if an excellent information retrieval system is not available, it can be very difficult to locate the information desired.

Reams of information have been published in books, tables, charts, periodicals, bulletins, and reports pertaining to materials selection for various environments. It is not the intent of this section of the handbook to make materials recommendations. However, it is deemed advisable to provide some general comments and to point out a few applications having unusual characteristics. The Hydraulic Institute Standards present a very comprehensive guide for polymer material selection.

Sulfuric Acid This is the most widely used chemical in industrial applications today, and much time is spent in evaluating and selecting materials for applications involving sulfuric acid with and without constituents. The following are some of the applications that merit special consideration.

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DILUTION OF COMMERCIALLY PURE SULFURIC ACID When sulfuric acid is diluted with water, there is considerable evolution of heat. At times, the mixing of the acid and water takes place not in the mixing tank but in the pump transferring the acid. This means that heat is evolved as the solution is passing through the pump. Temperatures of 200°F (93°C) or higher are reached, depending upon the degree of dilution and the amount of mixing taking place in the pump. Thus, the heat evolved in the dilution would restrict material selection. Very few metallics or nonmetallics are resistant to 70% sulfuric acid at temperatures approaching 200°F (93°C). Refer to Section 8.2 for material guidelines for sulfuric acid.

SULFURIC ACID SATURATED WITH CHLORINE It is a well-known fact that any solution involving wet chlorine is extremely corrosive. In a solution containing sulfuric acid and chlorine, the specific weight percentage of sulfuric acid determines whether the solution will accelerate corrosion. Because of the hygroscopic nature of concentrated sulfuric acid, it will absorb moisture from the chlorine. Thus, when a sulfuric acid-chlorine solution contains at least 80% sulfuric acid, there need be little concern for the chlorine because dry chlorine is essentially noncorrosive. In such a case, a material selection can be made as if sulfuric acid were the only constituent. If the solution is saturated with chlorine but contains less than approximately 80% sulfuric acid; however, the material selection must be based not only on the sulfuric acid but also on the wet chlorine. This, of course, is a very corrosive solution, and extreme caution must be exercised in selecting the material to be used.

SULFURIC ACID CONTAINING SODIUM CHLORIDE It is quite apparent that the addition of sodium chloride to sulfuric acid will result in the formation of hydrochloric acid and thus necessitate a material that will resist the corrosive action of hydrochloric acid also. Though this may seem obvious, it is amazing how often it is ignored. This is particularly true in 10 to 15% sulfuric acid pickling solutions to which sodium chloride has been added to increase the rate of pickling, with little or no consideration being given to the destructive effect of the salt on the process equipment handling the pickling solution.

PIGMENT MANUFACTURE A slurry of titanium dioxide in sulfuric acid is one of the processing steps in the manufacture of pigment. A variety of metallics and nonmetallics would be suitable for this application in the absence of the titanium dioxide solids, but the presence of the solids circulating in a pump renders practically all of the normal sulfuric acid-resistant pump materials unsuitable. Special consideration must be given to materials that will resist the severe erosion-corrosion encountered in this type of service.

SULFURIC ACID CONTAINING NITRIC ACID, FERRIC SULFATE, OR CUPRIC SULFATE The presence of these compounds in sulfuric acid solutions will drastically alter the suitability of materials that can be used. Their presence in quantities of 1% or less can make a sulfuric acid solution oxidizing, whereas it would normally be reducing. Their presence, singly or in combination, could serve as a corrosion inhibitor, thus in certain instances allowing a stainless steel, such as type 316, to be used. On the other hand, the same compounds could serve as a corrosion accelerator for a nonchromium bearing alloy, such as nickel-molybdenum alloys, and thus render it completely unsuitable.

Nitric Acid In the concentrations normally encountered in chemical applications, nitric acid presents fewer problems than sulfuric acid. The choice of metallic materials for various nitric applications is somewhat broader than the choice of nonmetallic materials. Nitric acid, being a strongly oxidizing acid, permits the use of stainless steel quite extensively, but its oxidizing characteristics restrict the application of nonmetallics in general and plastics in particular. Requiring special evaluation are such aggressive solutions as fuming nitric acid; nitric-hydrofluoric; nitrichydrochloric (some of which fall into the aqua regia category); nitric-adipic combinations; and practically any environment consisting of nitric acid in combination with other constituents. Invariably, additional constituents in nitric acid result in more aggressive corrosion; hence material selection becomes quite critical.

Hydrochloric Acid Both commercially pure and contaminated hydrochloric acid present difficult situations in selecting pump materials. The most common contaminant that

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creates problems is ferric chloride, the presence of which can render this otherwise reducing solution oxidizing and thus completely change the material of construction that can be used. Addition of a very few parts per million of iron to commercially pure hydrochloric acid can result in the formation of enough ferric chloride to cause materials such as nickel-molybdenum, nickel-copper, and zirconium to be completely unsuitable. Conversely, the presence of ferric chloride can make titanium completely suitable. Nonmetallics find extensive application in many hydrochloric acid environments. Often the limiting factors for the nonmetallics are temperature, mechanical properties, and suitability for producing pump parts in the design desired. With the nonmetallics, the near-complete immunity from corrosion in such environments subordinates corrosion resistance to other factors. Refer to Section 8.2 for material guidelines for hydrochloric acid.

Phosphoric Acid The increasing use and demand for all types of fertilizers have made phosphoric acid a very important commodity. In the wet process of producing phosphoric acid, the phosphate rock normally contains fluorides. In addition, at various stages of the operation the solution will also contain sulfuric, hydrofluoric, fluosilicic, and phosphoric acids as well as solids. In some instances, the water used in these solutions may have an exceptionally high chloride content, which can result in the formation of hydrochloric acid, which further aggravates the corrosion problem. It is also common for certain of these solutions to contain solids, which of course create an erosion-corrosion problem. Pure phosphoric and superphosphoric acids are relatively easy to cope with from a material standpoint, but when the solution contains all or some of the aforementioned constituents, a very careful materials evaluation must be conducted. Such environments are severely corrosive in the absence of solids and cause severe erosion-corrosion and a drastically reduced service life when solids are present. This is particularly significant with any type of chemical pump.

Chlorine Little need be said about the corrosivity of chlorine. Wet chlorine, in addition to being extremely hazardous, is among the most corrosive environments known. Dry chlorine is not corrosive and carbon steel is commonly used, but there are those who contend that dry chlorine does not exist. A common definition of dry chlorine for carbon steel use is less than 50 ppm water. Chlorine vapor combined with the moisture in the atmosphere, for instance, can create severe corrosion problems. In any case, selecting the most suitable material for any type of chlorine environment requires very careful evaluation.

Alkaline Solutions With some exceptions, alkaline solutions, such as sodium hydroxide or potassium hydroxide, do not present serious corrosion problems at temperatures below 200°F (93°C). However, in certain applications, purity of product is of utmost concern, necessitating selection of a material that will have essentially no corrosion rate. Among the exceptions to the rule that alkaline solutions are relatively noncorrosive are bleaches, alkaline brines, and other solutions containing chlorine in some form.

Organic Acids Organic acids are much less corrosive than inorganic acids. This does not mean, however, that they can be taken lightly. For instance, acetic, lactic, formic, and maleic acids all have their corrosive characteristics and must be treated accordingly when evaluating metallics and nonmetallics.

Salt Solutions Normally considered neutral, salt solutions do not present a serious corrosion problem. In some instances, process streams adjust pH to maintain a slightly alkaline environment, and such solutions are even less corrosive than when they are neutral. On the other hand, when a process stream has a pH adjustment to maintain a slightly acidic environment, the liquid becomes considerably more corrosive than neutral salt solutions. This condition requires that more effort be expended in evaluating the solution before making a material selection.

Organic Compounds Most organic compounds do not present corrosion problems of the same magnitude as inorganic compounds. This does not mean that any material arbitrarily selected will be a suitable choice. It does mean that there will be more materials

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available to choose from, but each application should be considered on its own merits. Of particular concern in this area are chlorinated organic compounds and those that will produce hydrochloric acid when moisture is present. Plastics, categorically, possess excellent corrosion resistance to inorganic compounds within their temperature limitations, but they do exhibit some weaknesses in their corrosion resistance to organic compounds.

Water Water is less corrosive than most of the other mediums encountered in the chemical and allied industries. For the term *water* to be meaningful, however, it is extremely important to know the specific kind of water: demineralized, fresh, brackish, salt, boiler feed, mine. These waters and the various constituents in them can demand a variety of materials, indicated, for example, by the spectrum of materials being studied and used in desalination programs. Because they are likely to have a very pronounced effect on our total economy, precise materials evaluation and selection are integral parts of these programs.

TYPES OF PUMP CORROSION

The types of corrosion encountered in chemical pumps may at first appear to be unusual compared with those found in other process equipment. Nevertheless pumps, like any other type of chemical process equipment, experience basically only eight forms of corrosion, of which some are more predominant in pumps than in other types of equipment. It is not the intent here to describe in detail these eight forms of corrosion, but it is desirable to enumerate them and provide a brief description of each so they can be recognized when they occur.

General, or Uniform, Corrosion This is the most common type, and it is characterized by essentially the same rate of deterioration over the entire wetted or exposed surface. General corrosion may be very slow or very rapid, but it is of less concern than the other forms of corrosion because of its predictability. However, predicting the general corrosion rate in a pump can be a difficult task because of the varying velocities of the solution in the pump.

Concentration Cell, or Crevice, Corrosion This is a localized form of corrosion resulting from small quantities of stagnant solution in areas such as threads, gasket surfaces, holes, crevices, surface deposits, and the underside of bolt and rivet heads. When concentration cell corrosion occurs, the concentration of metal ions or oxygen in the stagnant area is different from the concentration in the main body of the liquid. This causes electric current to flow between the two areas, resulting in severe localized attack in the stagnant area.

Pitting Corrosion This is the most insidious form of corrosion, and it is very difficult to predict. It is extremely localized and manifested by small holes, and the weight loss due to the pits will be only a small percentage of the total weight of the equipment. Chlorides in particular are notorious for inducing pitting of stainless steels. Pitting is common in areas other than stagnant areas, whereas concentration cell corrosion is basically confined to areas of stagnation.

Stress Corrosion Cracking This is localized failure caused by a combination of tensile stresses in a specific alloy in a specific medium. Fortunately, castings, because of their basic overdesign, seldom experience stress corrosion cracking. Corrosion fatigue, which can be classified as stress corrosion cracking, is of concern in chemical pump shafts because of the repeated cyclic stressing. Failures of this type occur at stress levels below the yield point as a result of the cyclic application of the stress.

Intergranular Corrosion This is a selective form of corrosion at and adjacent to grain boundaries. It is associated primarily with stainless steels but can also occur with other

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alloy systems. In stainless steels, it occurs when the material is subjected to heat in the 800 to 1600°F (427 to 871°C) temperature range. Unless other alloy adjustments are made, this form of corrosion can be prevented only by heat-treating. It is easily detectable in castings because the grains are quite large relative to those in wrought material of equivalent composition. In some instances, uniform corrosion is misinterpreted as intergranular corrosion because of the etched appearance of the surfaces exposed to the environment. Even in ideally heat-treated stainless steels, very slight accelerated attack can be noticed at the grain boundaries because these areas are more reactive than the grains themselves. Care should be taken to avoid confusing general and intergranular corrosion. Stainless steel castings will never encounter intergranular corrosion if they are properly heat-treated after being exposed to temperatures in the 800 to 1600°F range (427 to 871°C).

Galvanic Corrosion This occurs when dissimilar metals are in contact or are otherwise electrically connected in a corrosive medium. Corrosion of the less noble metal is accelerated, and corrosion of the more corrosion-resistant metal is decreased. The farther apart the metals or alloys are in the electromotive series, the greater the possibility of galvanic corrosion. When it is necessary to have two dissimilar metals in contact, the total surface area of the less resistant metal should far exceed that of the more resistant material. This tends to prevent premature failure by simply providing a substantially greater area of the more corrosion-prone material. This form of corrosion is not common in chemical pumps but may be of some concern with accessory items in contact with pump parts and exposed to the environment.

Erosion-Corrosion This type of failure is characterized by accelerated attack resulting from the combination of corrosion and mechanical wear. It may involve solids in suspension or high velocity. It is quite common with pumps where the erosive effects prevent the formation of a passive surface on alloys that require passivity to be corrosion resistant. The ideal material for avoiding erosion-corrosion in pumps would possess the characteristics of corrosion resistance, strength, ductility, and extreme hardness. Few materials possess such a combination of properties.

Selective Leaching Corrosion This, in essence, involves removal of one element from a solid alloy in a corrosive medium. Specifically, it is typified by dezincification, dealuminumification, and graphitization. This form of attack is not common in chemical pump applications because the alloys in which it occurs are not commonly used in heavy chemical applications.

TYPES OF CHEMICAL PUMPS

The second step in selecting a chemical pump is to determine which type of pump is required, based on the characteristics of the liquid and on the desired head and flow rate. It should also be noted that not all types are available in every material of construction, and the final selection of pump type may depend on the availability of designs in the proper material. Centrifugal, regenerative turbine, rotary, and diaphragm pumps are commonly used in the chemical industry.

Centrifugal Pumps Centrifugal pumps (Figure 1) are the most commonly used type in the chemical industry because of their suitability in practically any service. They are available in a wide range of corrosion-resistant materials. Centrifugal pumps with flow rates as high as 7000 gpm (1600 m^3/h) are commonplace. Heads range as high as 990 ft (300 m) at 60 Hz 2-pole electric motor speeds. Centrifugal pumps are normally mounted in the horizontal position, but they may also be installed vertically, suspended in a tank, or hung in a pipeline similar to a valve. They are simple, economical, dependable, and efficient. Disadvantages include reduced performance when handling liquids of more than 500 SSU (108 cSt) viscosity and the tendency to lose prime when comparatively small amounts of air or vapor (3 to 5%) are present in the liquid.

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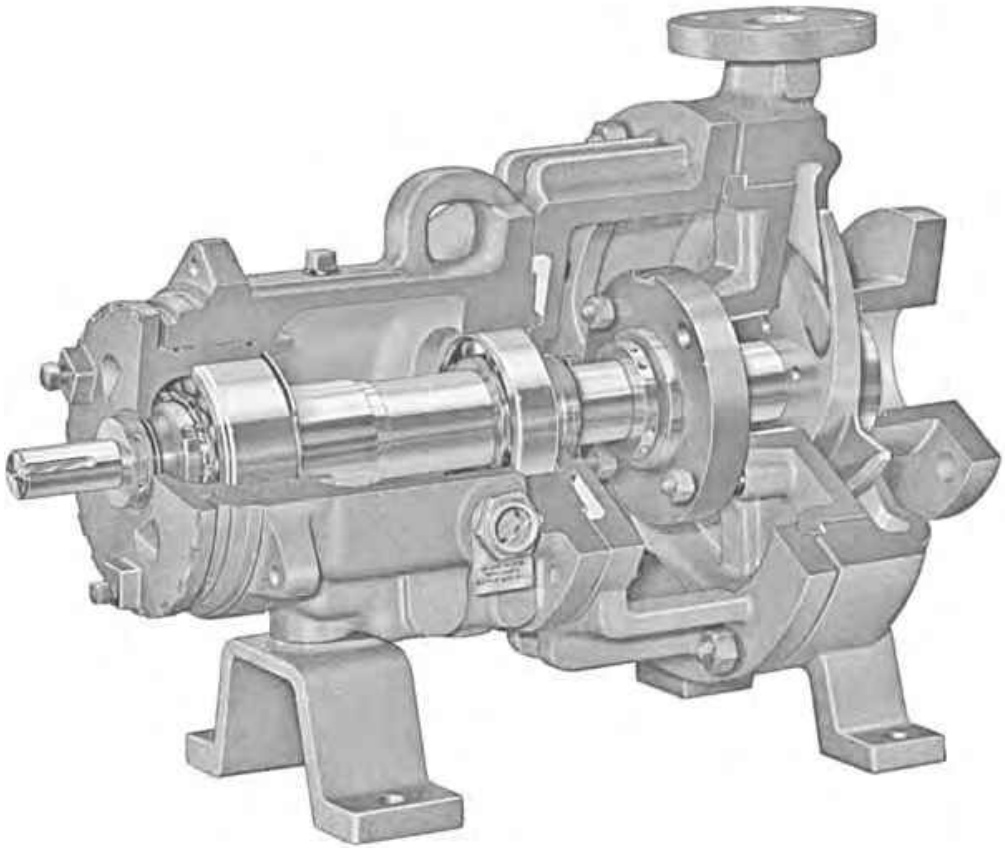


FIGURE 1 Cutaway of a typical centrifugal chemical pump (Flowserve Corporation)

Regenerative Turbine Pumps Flow rates up to 390 gpm (88 m^3/h) and heads up to 6900 ft (2100 m) are available with this type of pump at 60 Hz 2-pole electric motor speeds. When it is used for chemical service, the internal clearances must be increased to prevent rubbing contact, which results in decreased efficiency. These pumps are generally unsuitable for solid-liquid mixtures of any concentration.

Rotary Pumps Gear, screw, flexible-member, sliding-vane, piston, lobe, and cam types are particularly adept at pumping liquids of high viscosity or low vapor pressure. Because they operate on the positive displacement principle, they are inherently self-priming and their positive displacement makes them ideal for use in metering controlled volumes of liquid. The types that have close running internal clearances are generally limited to use on clear, nonabrasive liquids, and when built of materials that tend to gall or seize on rubbing contact, the clearances between mating parts must be increased, resulting in decreased efficiency.

Diaphragm Pumps These units are classed as positive displacement sealless pumps because the diaphragm acts as a limited displacement piston. Pumping action is obtained when the diaphragm is forced into reciprocating motion by mechanical linkage, compressed air, or oil from a pulsating external source. This type of construction eliminates any connection between the liquid being pumped and the source of energy and thereby eliminates the possibility of leakage. This characteristic is of great importance when toxic or very expensive liquids are being handled. Disadvantages include a limited selection of corrosion-resistant materials, limited head and capacity range, and the necessity of using check valves in the suction and discharge nozzles. Although air-operated diaphragm pumps are displacement pumps, they are not positive displacement pumps. The maximum pumping pressure cannot exceed the pressure of the compressed air powering the pump. Refer to Section 3.6 for more details on diaphragm pumps.

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CHEMICAL PUMP DESIGN CONSIDERATIONS

Casting Integrity Practically all the major components of chemical pumps are castings. There is probably more concern in chemical pump applications than in any other type of service because leakage, loss of product, and downtime can be extremely costly, as well as very dangerous.

Mechanical Properties There are several factors that determine whether a certain material can be utilized for a particular design. Materials may possess outstanding corrosion resistance but may be completely impossible to produce in the form of a chemical pump because of their mechanical properties. Accordingly, it is advisable to be aware of the mechanical properties of any material being considered in a corrosion evaluation program. Most materials are covered by ASTM or other specifications; such sources can be used for reference.

Nonmetallic Construction A number of nonmetallic materials have been used extensively in chemical pump construction (refer to Section 8.2). Their excellent chemical resistance makes them competitive with stronger metal alloys.

Weldments Welded construction should impose no limitation, provided the weldment is as good as or better than the base material. Materials requiring heat treatment to achieve maximum corrosion resistance must be treated after welding, or other adjustments must be made to make certain corrosion resistance is not sacrificed.

Section Thickness Pressure-containing parts are generally made thicker than required for handling a noncorrosive liquid so that full capability will be maintained even after the loss of some material to the corrosive medium. Parts that are subject to corrosion from two or three sides, such as impellers, must be made considerably heavier than their counterparts in water or oil pumps. Pressure-containing parts are also made thicker so they will remain serviceable after a specified amount of corrosive deterioration. Areas subject to high velocities, such as the cutwater of a centrifugal pump casing, are reinforced to allow for the accelerated corrosion caused by the high velocities.

Threads Threaded construction of any type in the wetted parts must be avoided whenever possible. The thread form is subject to attack from two sides, and a small amount of corrosive deterioration can reduce the holding power of the threaded joint.

Gaskets Gasket materials must resist being corroded by the chemical being handled. Compressed synthetic fibers and elastomers have been used extensively for corrosive services. Fluorocarbon resins are used because of their almost complete corrosion resistance.

Power End This assembly, consisting of the bearing housing, bearings, oil or grease seals, and bearing lubrication system, is normally made of iron or steel components; thus it must be designed to withstand a severe chemical plant environment. For example, where venting of the bearing housing is required, special means of preventing the entrance of water, chemical fumes, or dirt must be incorporated into the vent design.

The bearing that controls axial shaft movement is usually selected to limit shaft movement to 0.002 in (0.05 1 mm) or less. Endplay values above this limit have been found detrimental to impeller and mechanical seal operation.

Water jacketing or fan cooling of the bearing housing may be necessary under certain conditions to maintain bearing temperatures below 180°F (82°C), the limit used in most applications.

Maintenance Maintenance of a chemical pump in a corrosive environment can be a very costly and time-consuming item. When evaluating materials and design factors,

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maintenance aspects should be high on the priority list. The ease and frequency of maintenance are critical items and should be considered part of a preventive maintenance program. Such a program can be the most effective way of eliminating emergency shutdowns caused by pump failure. Furthermore, the knowledge gained in a routine preventive maintenance program can be of unlimited value when a breakdown does occur because repair personnel will have acquired a thorough knowledge of the construction details of the pump.

SEALING

The area around the sealing chamber probably causes more chemical pump failures than all other parts combined. The problem of establishing a seal between a rotating shaft and the stationary pump parts is one of the most intricate and vexing problems facing the pump designer.

Packings Braided fluorocarbon resins, aluminum, graphite, and many other materials or combinations of materials have been used to establish a seal (discussed in more detail in Section 2.2.2). A small amount of liquid must be allowed to seep through the packing to lubricate the surface between packing and shaft. This leakage rate is hard to control, and usually the packing is overtightened and the leakage is stopped. The unfortunate results of this condition is rapid scoring of the sealing surface, making it much harder to adjust the packing.

Mechanical Seals Mechanical shaft seals as described in Section 5.2 are used extensively on chemical pumps. The majority of chemical pumps in service today do not use packing. The primary consideration is selection of the proper materials for the type of corrosive liquid being pumped. Stainless steels, ceramics, graphite, and fluorocarbon resins are used to make most seal parts. Seals consist of rotating and stationary elements that, in the past, have been assembled into the pump as separate pieces. However, cartridge seals have now become an industry standard, where the seal, sleeve, and gland are preassembled as a unit, with no adjustment required, to simplify installation in the pump and improve seal operating life and reliability.

Seal Chambers Most chemical pumps are available with traditional convertible sealing chambers that can accept packing or mechanical seals. These "stuffing boxes" have been sized for packing and require narrow cross-section mechanical seals that are prone to early failure because of excessive temperature rise and trapped air. Chemical pumps are now available with oversized bore sealing chambers with additional liquid for cooling the seal and tapered bore sealing chambers with excellent seal cooling from fluid exchange with the pump and no trapped air, as they are self-venting.

Temperature One of the most important factors affecting the sealing medium is their operating temperature. Increased temperatures can result from high process temperatures and from heat generated by the sealing device. High temperature can increase the corrosive attack on the sealing chamber, the packing or the mechanical seal. Seal parts not designed for high temperature can also distort or crack and fail.

One answer to the heat/temperature problem is to cool the sealing chamber with a water jacket that surrounds it and the seal gland. These jackets tend to be poor heat exchangers, however, and water can be expensive—or nonexistent—in many locations. There is also the risk that the cooling water source will fail while the pump is running. These factors are contributing to the requirement that chemical pumps operating between 200 and 500°F (93 and 260°C) be selected with mechanical seals that can handle full process temperature. These pumps are then applied with oversized sealing chambers with product flush or tapered sealing chambers to maximize seal cooling and to avoid trapped air. In many cases, cooling water is no longer required.

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Pressure Seal chamber pressure varies with suction or discharge pressure depending on its location in the pump and impeller design. Variations in impeller design include those using vertical or horizontal seal rings in combination with balance ports, or the use of back vanes, pump-out vanes, or pump-out slots. All impeller designs depend upon a close running clearance between the impeller and the stationary pump parts. This clearance must be kept as small as possible to prevent excessive recirculation of the liquid and the resulting loss of efficiency. Unfortunately, most chemical pump materials tend to seize when subjected to rubbing contact, and the running clearances must therefore be increased considerably above those used in pumps for other industries.

At pressures above 100 lb/in^2 (690 kPa), packing is generally unsatisfactory. Mechanical seals incorporating a pressure balancing feature to relieve high face loading are the best means of sealing at pressures above 100 lb/in^2 (690 kPa).

Shaft Pump shaft bending—or deflection—can create additional sealing problems. Undersize shafts, or those made of materials that bend readily, will deflect from their true center in response to radial thrust on the impeller.

Mechanical seal operation is impaired when the shaft is bent or deflected during operation. Because the flexible member of the seal must adjust with each revolution of the shaft, excessive deflection results in shortened seal life. If the deflection is of more than nominal value, the flexible seal member will be unable to react with sufficient speed to keep the seal faces together, allowing leakage at the mating faces.

A limit of 0.002 in (0.05 mm) at the seal faces has been established as the maximum allowable shaft deflection consistent with good pump design and seal life. Operation of the pump outside the allowable operating flow region can also increase radial thrust and shaft deflection, shortening seal life.

Shaft Surface In the seal chamber region, the shaft surface must have corrosion resistance at least equal to and preferably better than that of the wetted parts of the pump. In addition, this surface must be hard enough to resist the tendency to wear under the packing or mechanical seal parts. Further, it must be capable of withstanding the sudden temperature changes often encountered in operation.

Often it is cost effective to make the shaft from high-strength carbon steel, and then add protective sleeve of stainless steel, plastic, carbon, glass, or a coating in the seal chamber area. Cylindrical sleeves are made so they can be removed and replaced when they become worn. Other designs utilize sleeves that are permanently bonded to the shaft. Where possible, shafts are increasingly being made from solid stainless steel to maximize the shaft diameter under the seal, reducing bending and increasing seal life. This is most effective when cartridge seals are used as they include their own sleeve.

Another method of obtaining a hard surface in this region is the welded overlay or spray coating of hard metals onto the base shaft. Ceramic materials applied by the plasma spray technique possess excellent corrosion resistance but cannot achieve the complete density required to protect the underlying shaft.

Composite shafts utilizing carbon steel for the power end and a higher alloy for the wet end have been used extensively where the high-alloy end has acceptable corrosion resistance. The two ends are joined by various welding techniques, and the combination of metals is therefore limited to those that can be easily welded together. On such assemblies, the weld joint and the heat-affected zone must be outside the wetted area of the shaft.

Sealless Pumps Elimination of the sealing chamber and its associated problems has been the objective of several pump designs. Refer to Section 2.2.3 for sealless canned motor pumps and magnetically driven pumps. Diaphragm pumps previously mentioned are sealless pumps.

Vertical immersion pumps utilize sleeve bearings in the area immediately above the impeller to limit the flow of liquid along the shaft. For chemical service, the problem of materials associated with this bearing and its lubrication has to be addressed on an application basis.

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CONSTRUCTION OF NONMETALLIC PUMPS

A number of nonmetallic materials have been used extensively in chemical pump construction (refer to Section 8.2). Their excellent chemical resistance makes them competitive with stronger metal alloys.

CHEMICAL PUMP STANDARDS

In 1962, a committee of the Manufacturing Chemists Association (MCA) reached agreement with a special committee of the Hydraulic Institute on a proposed American Standards Association (ASA) standard for chemical process pumps. This document was referred to as the American Voluntary Standard or the Manufacturing Chemists Association Standard. In 1971, it was accepted by the American National Standards Institute (ANSI) and issued as ANSI Standard B123.1. This ANSI standard was renumbered to ANSI B73.1 in 1974, to ANSI/ASME B73.1M in 1984, to ASME B73.1M in 1991, and to ASME B73.1 in 2001.

It is the intent of this specification for horizontal end suction centrifugal pumps for chemical process that pumps of similar size from all sources of supply shall be dimensionally interchangeable with respect to mounting dimensions, size, and location of suction and discharge nozzles, input shafts, base plates, and foundation bolts. Table 1 lists the pump dimensions that have been standardized, and a cross-sectional assembly of a pump meeting these criteria is shown in Figure 1.

It is also the intent of this standard to outline certain design features that will minimize maintenance problems. The standard states, for instance, that the pump shaft should be sized so the maximum shaft deflection shall not exceed 0.005 in (0.127 mm) at the centerline of the impeller when the pump is operating within its design region. It does not specify shaft diameter because impeller diameter, shaft length, and provision for operation with liquids of high specific gravity would determine the proper diameter.

The standard also states that the minimum bearing life shall be 17,500 h within the design region of the pump. Bearing size is not specified but is to be determined by the individual manufacturer and will be dependent upon the load to be carried.

Additional specifications in the standard include hydrostatic test pressure, shaft finish at rubbing points, packing space, and seal chamber space.

ASME B73.2 is the specification for vertical in-line centrifugal pumps for chemical process. Dimensional criteria are shown in Table 2, and a pump meeting these requirements is shown in Figure 2.

ASME B73.3 is the specification for sealless horizontal end suction centrifugal pumps for chemical process. It gives dimensional and other technical requirements for both magnetic drive and canned motor pumps.

ASME B73.5 is the specification for thermoplastic and thermoset polymer material horizontal end suction centrifugal pumps for chemical process. A typical composite pump meeting these requirements is shown in Figure 3.

Standards for acceptance tests are given in ANSI/HI 1.6, the American National Standard for Centrifugal Pump Tests.

The International Organization for Standardization (ISO) also publishes standards for chemical pumps. ISO 2858 gives the nominal duty point and dimensions for end suction horizontal centrifugal pumps. ISO 3069 gives the dimensions of cavities for mechanical shaft seals and soft packing for pumps conforming to ISO 2858. ISO 3661 gives the dimensions of baseplates for pumps conforming to ISO 2858. ISO 5199 gives the technical specifications of pumps conforming to ISO 2858.

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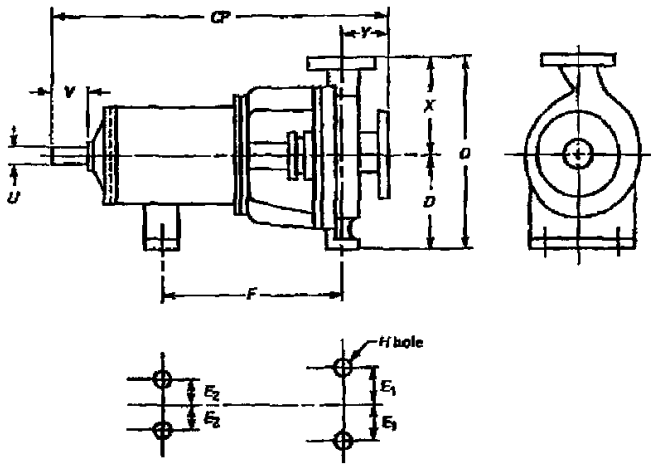


TABLE 1 *ASME B73.1 Horizontal pump dimensions*

Dimension Designation	Size; Suction × Discharge × Nominal Impeller Diameter	CP	D	$2E_1$	$2E_2$	F	H	O	U [Note (1)] Diameter	U [Note (1)] Keyway	Vmin	X	Y
AA	1.5×1×6 (40×25×150)	17.5 (445)	5.25 (133)	6 (152)	0	7.25 (184)	0.625 (16)	11.75 (298)	0.875 (22.23)	0.188×0.094 (4.76×2.38)	2 (51)	6.5 (165)	4 (102)
AB	3×1.5×6 (80×40×150)	17.5 (445)	5.25 (133)	6 (152)	0	7.25 (184)	0.625 (16)	11.75 (298)	0.875 (22.23)	0.188×0.094 (4.76×2.38)	2 (51)	6.5 (165)	4 (102)
AC [Note (2)]	3×2×6 (80×50×150)	17.5 (445)	5.25 (133)	6 (152)	0	7.25 (184)	0.625 (16)	11.75 (298)	0.875 (22.23)	0.188×0.094 (4.76×2.38)	2 (51)	6.5 (165)	4 (102)
AA [Note (2)]	1.5×1×8 (40×25×200)	17.5 (445)	5.25 (133)	6 (152)	0	7.25 (184)	0.625 (16)	11.75 (298)	0.875 (22.23)	0.188×0.094 (4.76×2.38)	2 (51)	6.5 (165)	4 (102)
AB [Note (2)]	3×1.5×8 (80×40×200)	17.5 (445)	5.25 (133)	6 (152)	0	7.25 (184)	0.625 (16)	11.75 (298)	0.875 (22.23)	0.188×0.094 (4.76×2.38)	2 (51)	6.5 (165)	4 (102)
A10	3×2×6 (80×50×150)	23.5 (597)	8.25 (210)	9.75 (248)	7.25 (184)	12.5 (318)	0.625 (16)	16.5 (420)	1.125 (28.58)	0.25×0.125 (6.35×3.18)	2.625 (67)	8.25 (210)	4 (102)
A50	3×1.5×8 (80×40×200)	23.5 (597)	8.25 (210)	9.75 (248)	7.25 (184)	12.5 (318)	0.625 (16)	16.75 (425)	1.125 (28.58)	0.25×0.125 (6.35×3.18)	2.625 (67)	8.5 (216)	4 (102)
A60	3×2×8 (80×50×200)	23.5 (597)	8.25 (210)	9.75 (248)	7.25 (184)	12.5 (318)	0.625 (16)	17.75 (450)	1.125 (28.58)	0.25×0.125 (6.35×3.18)	2.625 (67)	9.5 (242)	4 (102)
A70	4×3×8 (100×80×200)	23.5 (597)	8.25 (210)	9.75 (248)	7.25 (184)	12.5 (318)	0.625 (16)	19.25 (490)	1.125 (28.58)	0.25×0.125 (6.35×3.18)	2.625 (67)	11 (280)	4 (102)
A05 [Note (2)]	2×1×10 (50×25×250)	23.5 (597)	8.25 (210)	9.75 (248)	7.25 (184)	12.5 (318)	0.625 (16)	16.75 (425)	1.125 (28.58)	0.25×0.125 (6.35×3.18)	2.625 (67)	8.5 (216)	4 (102)
A50	3×1.5×10 (80×40×250)	23.5 (597)	8.25 (210)	9.75 (248)	7.25 (184)	12.5 (318)	0.625 (16)	16.75 (425)	1.125 (28.58)	0.25×0.125 (6.35×3.18)	2.625 (67)	8.5 (216)	4 (102)
A60	3×2×10 (80×50×250)	23.5 (597)	8.25 (210)	9.75 (248)	7.25 (184)	12.5 (318)	0.625 (16)	17.75 (450)	1.125 (28.58)	0.25×0.125 (6.35×3.18)	2.625 (67)	9.5 (242)	4 (102)

(Continued)

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TABLE 1 ASME B73.1 Horizontal pump dimensions (*Continued*)

Dimension Designation	Size; Suction × Discharge × Nominal Impeller Diameter	*CP*	*D*	$2E_1$	$2E_2$	*F*	*H*	*O*	*U* [Note (1)] Diameter	*U* [Note (1)] Keyway	*V*min	*X*	*Y*
*A*70	4×3×10 (100×80×250)	23.5 (597)	8.25 (210)	9.75 (248)	7.25 (184)	12.5 (318)	0.625 (16)	19.25 (490)	1.125 (28.58)	0.25×0.125 (6.35×3.18)	2.625 (67)	11 (280)	4 (102)
*A*40	4×3×10 (100×80×250)	23.5 (597)	10 (254)	9.75 (248)	7.25 (184)	12.5 (318)	0.625 (16)	22.5 (572)	1.125 (28.58)	0.25×0.125 (6.35×3.18)	2.625 (67)	12.5 (318)	4 (102)
*A*50 [Note (3)]	6×4×10 (150×100×250)	23.5 (597)	10 (254)	9.75 (248)	7.25 (184)	12.5 (318)	0.625 (16)	23.5 (597)	1.125 (28.58)	0.25×0.125 (6.35×3.18)	2.625 (67)	13.5 (343)	4 (102)
*A*20 [Note (2)]	3×1.5×13 (80×40×330)	23.5 (597)	10 (254)	9.75 (248)	7.25 (184)	12.5 (318)	0.625 (16)	20.5 (520)	1.125 (28.58)	0.25×0.125 (6.35×3.18)	2.625 (67)	10.5 (266)	4 (102)
*A*30	3×2×13 (80×50×330)	23.5 (597)	10 (254)	9.75 (248)	7.25 (184)	12.5 (318)	0.625 (16)	21.5 (548)	1.125 (28.58)	0.25×0.125 (6.35×3.18)	2.625 (67)	11.5 (292)	4 (102)
*A*40	4×3×13 (100×80×330)	23.5 (597)	10 (254)	9.75 (248)	7.25 (184)	12.5 (318)	0.625 (16)	22.5 (572)	1.125 (28.58)	0.25×0.125 (6.35×3.18)	2.625 (67)	12.5 (318)	4 (102)
*A*80 [Note (3)]	6×4×13 (150×100×330)	23.5 (597)	10 (254)	9.75 (248)	7.25 (184)	12.5 (318)	0.625 (16)	23.5 (597)	1.125 (28.58)	0.25×0.125 (6.35×3.18)	2.625 (67)	13.5 (343)	4 (102)
*A*90 [Note (3)]	8×6×13 (200×150×330)	33.875 (860)	14.5 (368)	16 (406)	9 (229)	18.75 (476)	0.875 (22)	30.5 (775)	2.375 (60.33)	0.625×0.313 (15.88×7.94)	4 (102)	16 (406)	6 (152)
*A*100 [Note (3)]	10×8×13 (250×200×330)	33.875 (860)	14.5 (368)	16 (406)	9 (229)	18.75 (476)	0.875 (22)	32.5 (826)	2.375 (60.33)	0.625×0.313 (15.88×7.94)	4 (102)	18 (457)	6 (152)
*A*105 [Note (3)]	6×4×15 (150×100×380)	33.875 (860)	14.5 (368)	16 (406)	9 (229)	18.75 (476)	0.875 (22)	30.5 (775)	2.375 (60.33)	0.625×0.313 (15.88×7.94)	4 (102)	16 (406)	6 (152)
*A*110 [Note (3)]	8×6×15 (200×150×380)	33.875 (860)	14.5 (368)	16 (406)	9 (229)	18.75 (476)	0.875 (22)	32.5 (826)	2.375 (60.33)	0.625×0.313 (15.88×7.94)	4 (102)	18 (457)	6 (152)
*A*120 [Note (3)]	10×8×15 (250×200×380)	33.875 (860)	14.5 (368)	16 (406)	9 (229)	18.75 (476)	0.875 (22)	33.5 (851)	2.375 (60.33)	0.625×0.313 (15.88×7.94)	4 (102)	19 (483)	6 (152)
*A*105 [Note (3)]	6×4×17 (150×100×430)	33.875 (860)	14.5 (368)	16 (406)	9 (229)	18.75 (476)	0.875 (22)	30.5 (775)	2.375 (60.33)	0.625×0.313 (15.88×7.94)	4 (102)	16 (406)	6 (152)
*A*110 [Note (3)]	8×6×17 (200×150×430)	33.875 (860)	14.5 (368)	16 (406)	9 (229)	18.75 (476)	0.875 (22)	32.5 (826)	2.375 (60.33)	0.625×0.313 (15.88×7.94)	4 (102)	18 (457)	6 (152)
*A*120 [Note (3)]	10×8×17 (250×200×430)	33.875 (860)	14.5 (368)	16 (406)	9 (229)	18.75 (476)	0.875 (22)	33.5 (851)	2.375 (60.33)	0.625×0.313 (15.88×7.94)	4 (102)	19 (483)	6 (152)

GENERAL NOTES:

(a) Dimensions in parentheses are approximate equivalents in millimeters.

(b) All other dimensions are in inches.

NOTES:

(1) "*U*" Diameter may be 1.625 in (41.28 mm) in *A*05 through *A*80 sizes to accomodate high torque values.

(2) Discharge flange may have tapped bolt holes.

(3) Suction flange may have tapped bolt holes.

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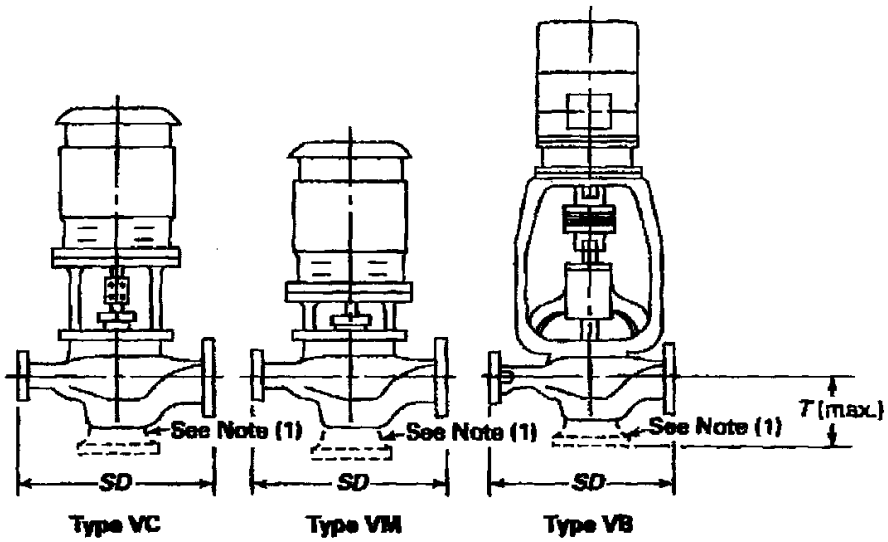


TABLE 2 ASME B73.2 Vertical in-line pump dimensions

Dimensions, mm					Dimensions, in				
Standard Pump Designation [Note (2)]	Metric Flange Sizes		SD 2.5 –2.0	*T*, Max.	Standard Pump Designation [Note (2)]	ANSI 125, 150, 250, or 300 Flange Sizes		SD +0.10 –0.08	*T*, Max.
	Suction	Discharge				Suction	Discharge		
50-40-380	50	40	380	175	2015/15	2	1.5	14.96	6.89
50-40-430	50	40	430	175	2015/17	2	1.5	16.93	6.89
50-40-480	50	40	480	175	2015/19	2	1.5	18.90	6.89
80-40-380	80	40	380	200	3015/15	3	1.5	14.96	7.87
80-40-480	80	40	480	200	3015/19	3	1.5	18.90	7.87
80-40-610	80	40	610	200	3015/24	3	1.5	24.02	7.87

(*Continued*)

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TABLE 2 ASME B73.2 Vertical in-line pump dimensions (*Continued*)

Dimensions, mm					Dimensions, in				
Standard Pump Designation [Note (2)]	Metric Flange Sizes		SD 2.5 –2.0	*T*, Max.	Standard Pump Designation [Note (2)]	ANSI 125, 150, 250, or 300 Flange Sizes		SD +0.10 –0.08	*T*, Max.
	Suction	Discharge				Suction	Discharge		
80-50-430	80	50	430	200	3020/17	3	2	16.93	7.87
80-40-510	80	50	510	200	3020/20	3	2	20.08	7.87
80-50-610	80	50	610	200	3020/24	3	2	24.02	7.87
100-80-560	100	80	560	225	4030/22	4	3	22.05	8.86
100-80-635	100	80	635	225	4030/25	4	3	25.00	8.86
100-80-710	100	80	710	225	4030/28	4	3	27.95	8.86
150-100-610	150	100	610	250	6040/24	6	4	24.02	9.84
150-100-710	150	100	710	250	6040/28	6	4	27.95	9.84
150-100-760	150	100	760	250	6040/30	6	4	29.92	9.84

NOTES:

(1) Optional separate pedestal.

(2) Pump designation defines design, flange sizes, and SD dimension (e.g., VC, VB 50-40-380).

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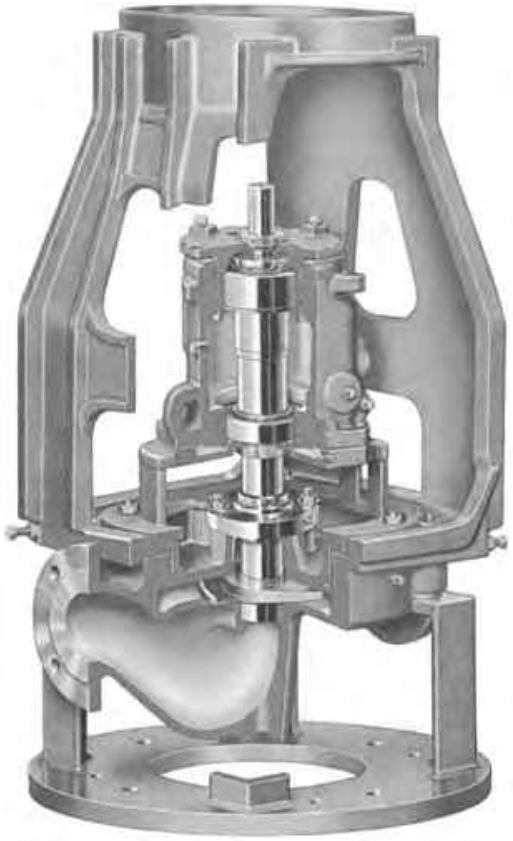


FIGURE 2 Vertical in-line centrifugal pump for chemical process applications (Flowserve Corporation)

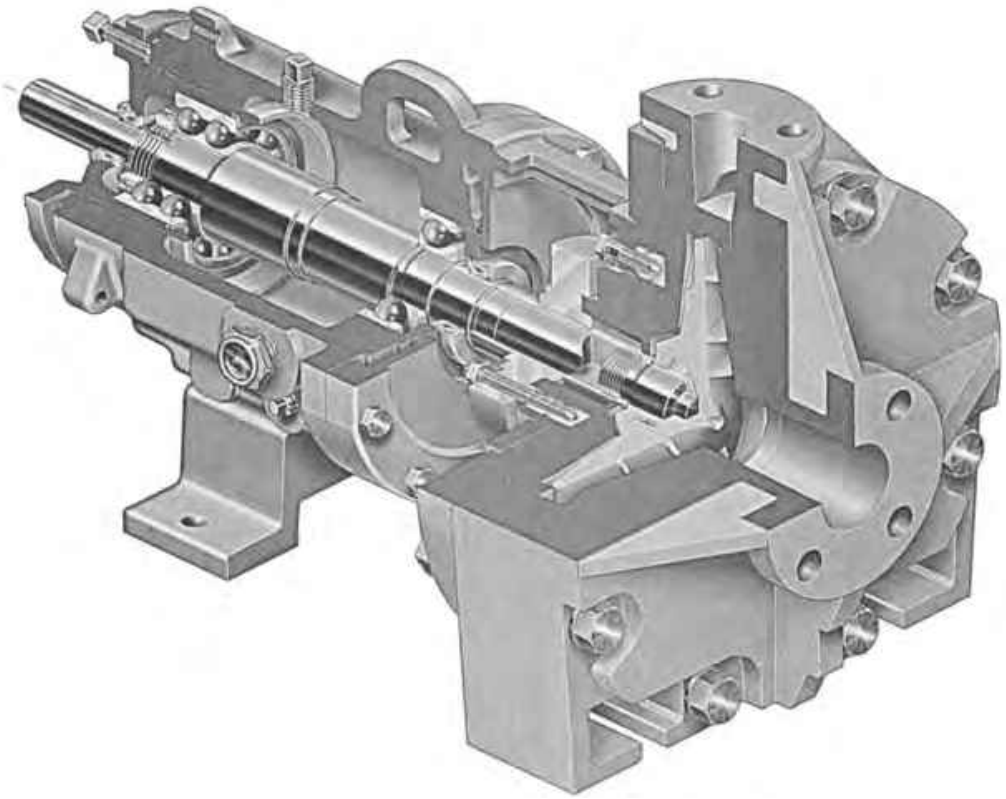


FIGURE 3 A typical composite pump (Flowserve Corporation)

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SECTION 12.8
PETROLEUM INDUSTRY

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Every phase of petroleum production, transportation, and refining requires a variety of pump types.

Production pumps include reciprocating units for mud circulation during drilling operations and sucker-rod, hydraulic rodless, and motor driven submersible centrifugal units for lifting crude to the surface. The most common use of centrifugal pumps in production is for water flooding (secondary recovery, subsidence prevention, or pressure maintenance).

Transportation pumps include units for gathering, on and offshore production, pipelining crude and refined products, loading and unloading tankers, tank cars, barges, or tank trucks, and servicing airport fueling terminals. The majority of the units are centrifugal-type pumps.

While refining units use all types of pumps, the vast majority of them are centrifugal as well. They vary from single-stage centrifugal units to horizontal and vertical multistage barrel-type pumps handling a variety of products over a full range of temperatures and pressures. Centrifugal pumps are also used for auxiliary services, such as boiler feedwater, cooling water, and rainwater transfer.

Except for some comments about the use of displacement pumps for handling viscous liquids, this section is restricted to centrifugal pumps. It also includes an overview of the requirements for some of the principal types of centrifugal pumps.

REFINERY PUMPS

Major refinery processes include crude distillation, vacuum tower separation, catalytic cracking, alkylation, hydrocracking, catalytic reforming, coking, and hydrotreatment for the removal of sulfur and nitrogen. The products resulting from these processes include motor gasoline, commercial jet fuel and kerosene, distillate fuel oil, residual fuel oil, and lubricating oils. The American Petroleum Institute (API) lists over a thousand distinct refinery products. The American Petroleum Institute Standard 610, "Centrifugal Pumps

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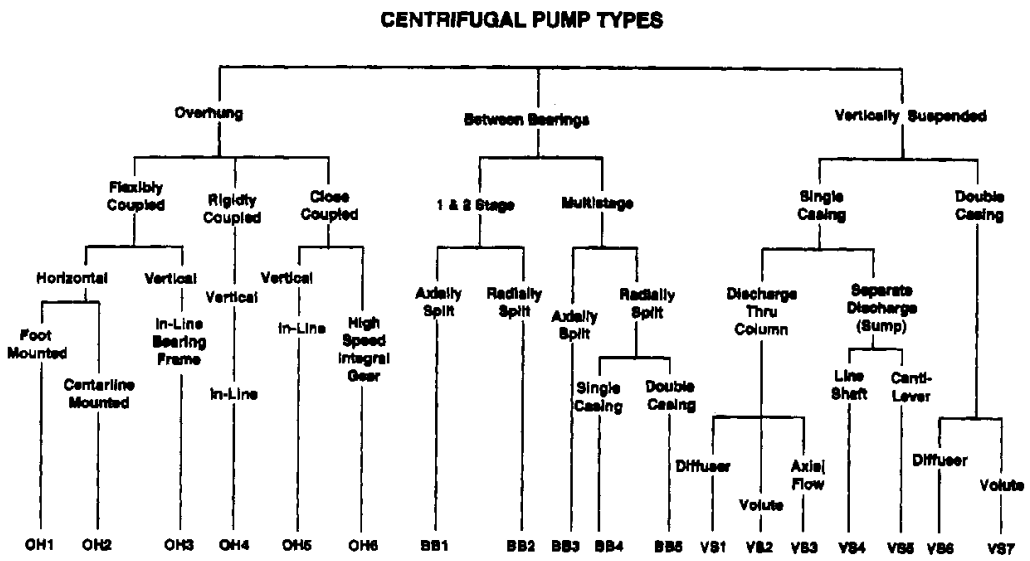


FIGURE 1 Pump classification type identification (Courtesy of the American Petroleum Institute[1])

for Petroleum, Petrochemical and Natural Gas Industries (API 610), has established specifications for the design features required for centrifugal pumps used for general refinery service.

API 610 first edition was published in October of 1954. In its initial form, the document contained only 15 pages of text and applied to single stage overhung pumps only. Since the original publication, the standard has been updated on a regular basis, approximately every five to six years. The tenth edition (2004), contains 89 pages of text, 91 pages of annexes, and covers the specification of as many as 18 different pump types. The eighth edition (1995) showed, for the first time, a "family tree" (Figure 1) of pump types that are potentially covered by the standard and a document organization that makes it clear which requirements apply to what pumps.

The tenth edition of API 610 is technically identical to the ninth edition other than editorial and formatting changes. These latter changes were implemented to allow the standard to be published as ISO 13709, making it for the first time a true, globally accepted international standard.

Construction Figure 2 illustrates the details of a single-stage overhung refinery process pump meeting API 610 requirements. Such a pump is referred to as a Type OH2. The suction nozzle may be located either at the end or at the top. A spacer coupling with a minimum 5 inch long spacer is used between the pump shaft and the driver shaft so the bearing bracket and cover may be removed without disconnecting the suction or discharge piping. This is referred to as a back pull-out design. Figure 3 illustrates the details of a between bearings, two-stage, radially split pump of type BB2. Type BB2 pumps can have one or two stages. Figures 4 and 5 show horizontal multistage pumps. Figure 4 is an axially split machine, Type BB3, and Figure 5 is a radially split or barrel-type pump, Type BB5. It is a requirement of all present day horizontal refinery pumps that the mechanical seal can be changed and the rotor removed without disconnecting the piping or moving the driver.

API 610 requires standardized seal chamber dimensions for all pumps. For overhung and vertically-suspended designs, there is only one seal chamber and seal to prevent leakage to the atmosphere. For between bearings designs, there are two seal chambers and two seals. In single-stage overhung pumps, the seal chamber pressure during pump operation is between suction and discharge pressure, depending on the design of the wear rings and impeller balance holes. Between-bearings pumps of two or more stages usually have a breakdown bushing with bleed-off to suction at the high-pressure end to balance the pressure at the shaft seals. Vertically suspended pumps are usually also equipped with a leak-off arrangement to reduce seal chamber pressure from discharge pressure.

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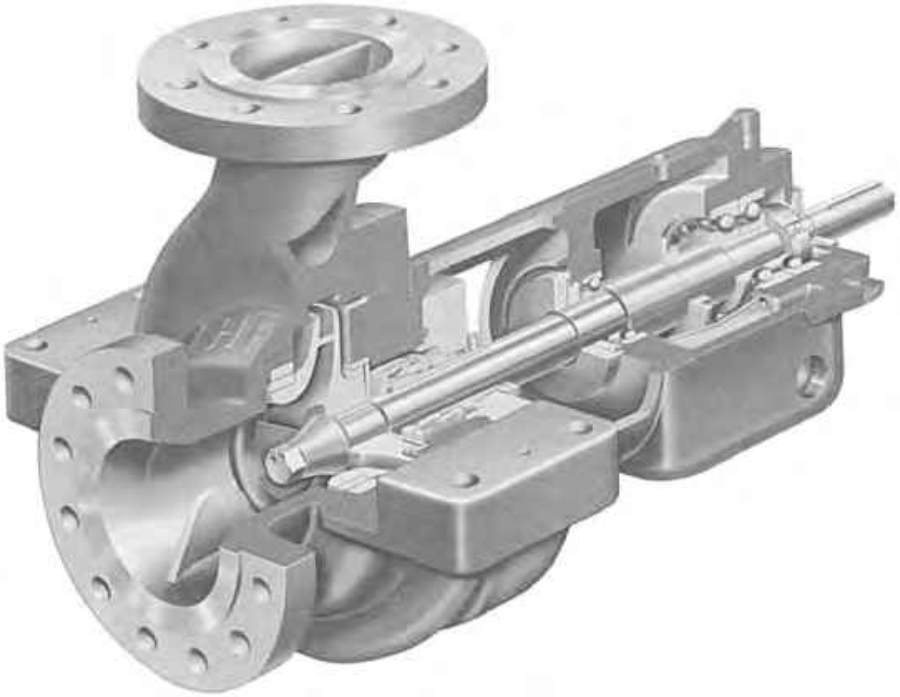


FIGURE 2 Single-stage overhung process pump—API Type OH2 (Flowserve Corporation)

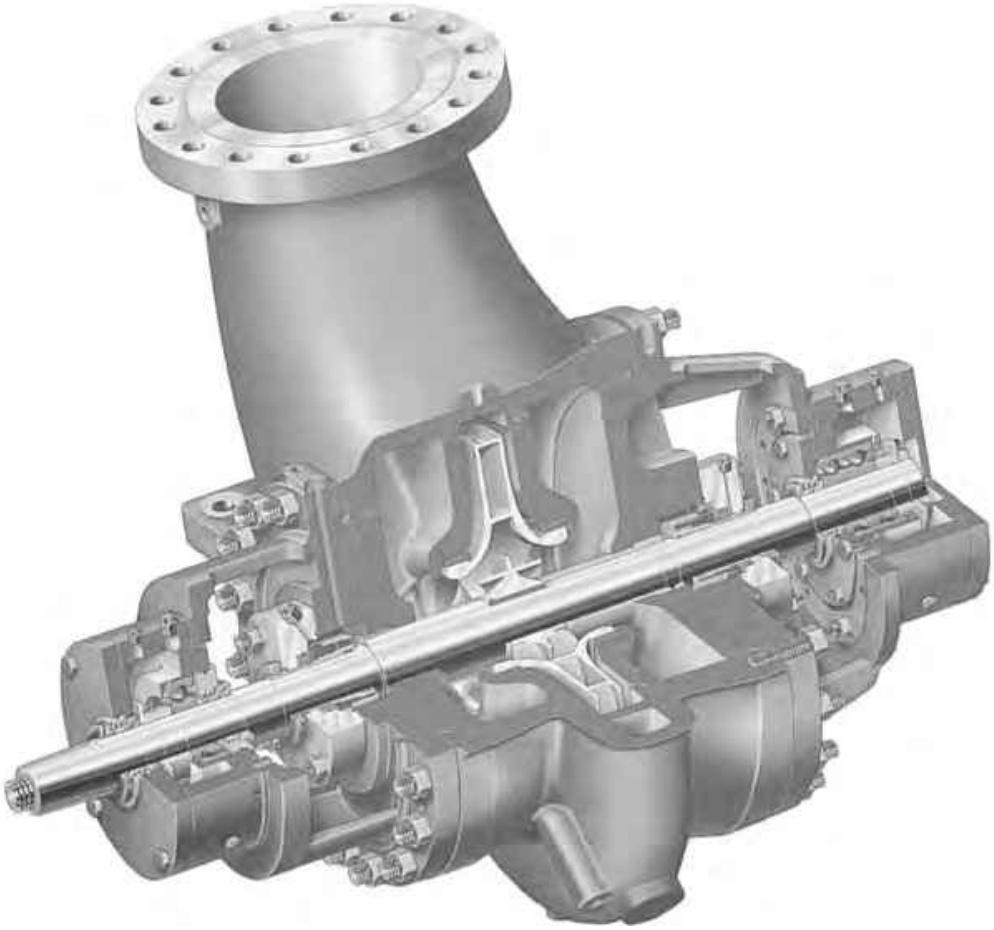


FIGURE 3 Between bearings, single-stage, double suction, radially-split process pump—API 610 Type BB2 (Flowserve Corporation)

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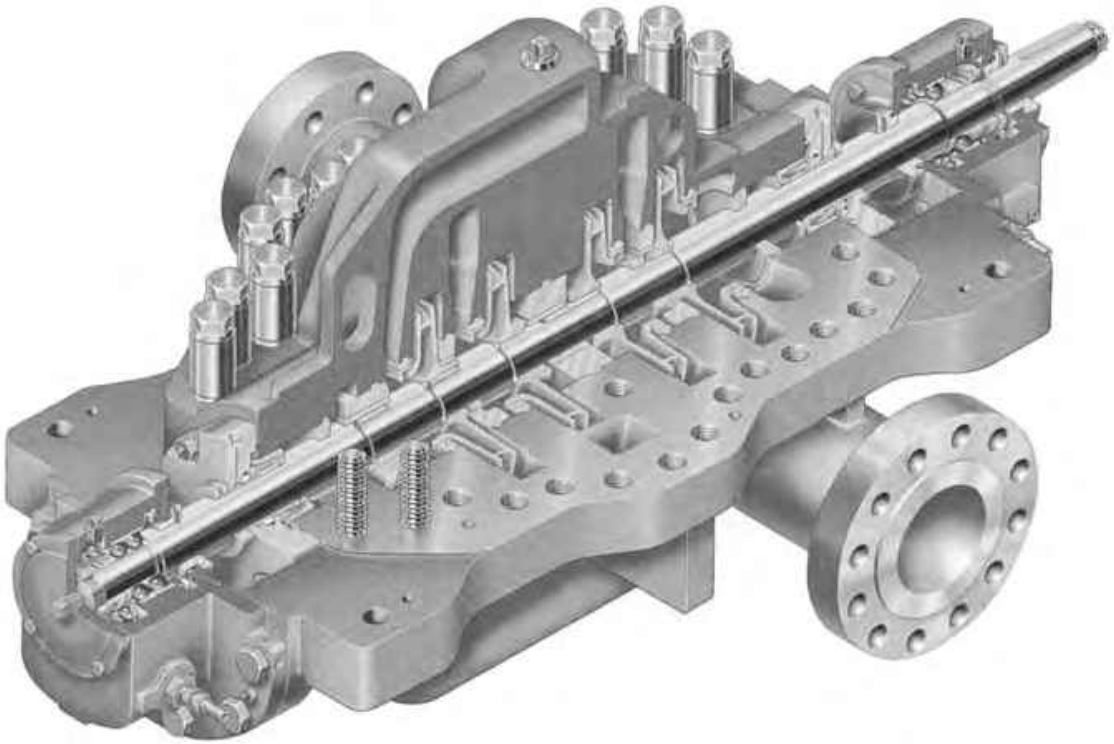


FIGURE 4 Axially split multistage pump API 610 Type BB3 (Flowserve Corporation)

In API 610 ninth edition, mechanical seal piping and lubricating systems (when separate from the pump itself) were removed from the standard. The mechanical seal requirements were moved into API 682, "Shaft Sealing Systems for Centrifugal and Rotary Pumps" and the piping and lubrication system requirements were moved into API 614 "Lubrication, Shaft Sealing, and Control-Oil Systems and Auxiliaries for Petroleum, Chemical, and Gas Industry Services". These API standards have been adopted by the International Standards Organization (ISO), API 682 as ISO 21049, and API 614 as ISO 10438.

Although most refiners continue to strongly prefer single seals for cost and reliability reasons, environmental requirements increasingly drive them toward dual seals. From the mid 90s to the present approximately 40% of refinery pumps in hydrocarbon services have been specified to have dual seals. In the last few years this percentage has climbed to the point that a new refinery project in detailed design has specified more than 90% of hydrocarbon pumps to have dual seals.

Regardless of whether the seal is a single or dual, for each seal, one of the sealing rings is made of carbon, and the mating ring is either silicon carbide or tungsten carbide. API 610/ISO 13709 requires that seals and their support systems comply with API 682/ISO 21049. API 682/ISO 21049 gives standardized auxiliary piping plans for the support of all liquid-type mechanical seals. Dual mechanical seals have proved progressively more reliable while the seals and their support systems have become progressively more expensive. It is not unusual for the mechanical seal and its support system in a single-stage overhung pump to cost more than the bare pump, driver, and baseplate.

The thrust bearing for single-stage overhung pumps is subject to axial loading caused by exposure of one end of the shaft to suction pressure and forces on the impeller due to the differential between suction pressure and discharge pressure. When single-stage overhung pumps are used for suction pressures in excess of 250 psig (17 MPa), it is normal practice to "unbalance" the impeller wear rings. This is done by making the diameter of the back wear rings smaller than that of the suction side wear rings, or by eliminating the back wear rings completely. This produces an unbalanced axial thrust load in the opposite direction to that created by high suction pressure and, thus, decreases the net

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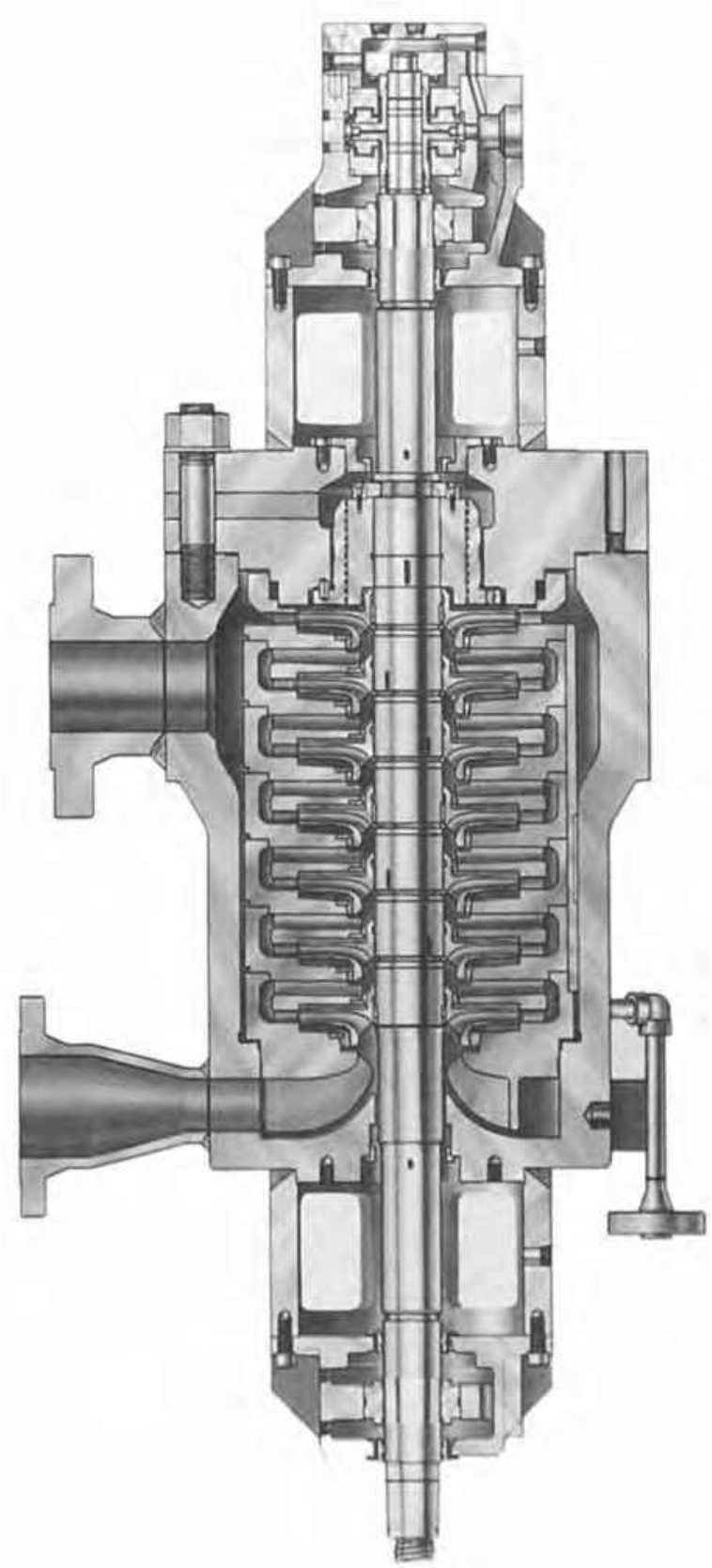


FIGURE 5 Radially split or barrel type pump—API 610 Type BB5 (Flowserve Corporation)

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axial load on the thrust bearing. In the case of multistage pumps, the differential pressure across the impellers builds with each stage and some method must be employed to reduce the load on the thrust bearing. In some designs, axial thrust is balanced by means of a balance piston/leak-off arrangement; in others, an opposed impeller arrangement helps to balance axial loads. In all cases, to meet API 610 requirements, the bearings must be designed for a minimum life of 25,000 hours of continuous operation at rated conditions and 16,000 hours at maximum radial and axial loads, at rated speed.

All single-stage overhung pumps and most one and two stage between bearings pumps are designed with "families" of bearing brackets/housings. This results in the bearings being selected for some "worst case" or "highest load" set of hydraulics. Fatigue bearing life is rarely a concern in API 610 pumps and bearing life is generally a function of lubricant quality and operational effectiveness of the owner.

It is well recognized in the oil and gas industry that for expensive sealing systems to deliver very low fugitive emissions and high reliability, radial movement at the seal faces must be held to a minimum. API 610 addresses this issue with a number of requirements. Among these are:

- Shaft deflection at the seal faces is limited to 0.002 in under the worst conditions.
- One and two stage pumps are required to have a dry bending first critical above 120% of maximum continuous speed.
- Seal chamber dimensions are standardized.
- Bearings must be designed for a minimum life of 25,000 hours at rated conditions, 16,000 hours at maximum radial and axial loads and rated speed.

The shaft deflection criteria, the dry critical criteria and the bearing minimum life criteria all drive shaft stiffness up. The seal chamber dimensions set the minimum overhang, the pump can have and limit how stiff the shaft can be. Note that Hydraulic Institute Standard 1.3 2000 Section 1.3.4.7.2 provides standard calculations to check shaft deflection and dry bending critical speed as well as bearing loads. With bearing loads known the several bearing manufacturers provide very simple tools for calculating bearing life on their websites.

Having noted the importance of minimizing shaft deflection at the seal faces, API 610 has no requirement for pumps to be designed with double volutes. However, commercial interests of pump manufacturers make it almost universal that single-stage overhung pumps have double volutes over a certain size, typically a 3-in discharge and, for larger pumps, even some 3-in discharge sizes are available with a double volute. The most significant exceptions are the smaller pumps, where it is simply not cost effective to cast a double volute.

To aid in maintaining alignment at various operating temperatures, pump mounting feet are located on the case at the same centerline as the shaft for all horizontal pumps. Earlier editions of API 610 also suggested using bearing or seal chamber cooling at elevated pumping temperatures. However, the industry has moved toward less and less cooling water usage because of cost and availability of quality water sources in many areas. Currently, API 610 leaves cooling as an item to be jointly agreed upon by the purchaser and the pump vendor. It should also be noted that the quality and stability of lubricating oils has significantly improved over the last few decades, and oils simply perform better and more adequately without cooling than before.

In general, pumps meeting API 610 are designed for operation at elevated temperatures. In all services, but particularly in high temperature services, pumps are subject to loads at the nozzles due to thermal movements of piping, fabrication errors in the piping, and movements of the piping supports. These forces and moments tend to "move" the pump relative to its driver and result in misalignment of the pump and driver shafts. This misalignment can increase vibration levels and impose forces on the shafting that contribute to poor reliability. API 610 prescribes minimum forces and moments, by nozzle size, that complying pumps must be able to withstand. Further, it outlines a test procedure to allow verification that a pump meets these minimum nozzle load requirements. The coordinate system for orientation of nozzle loads is in accordance with ISO 1503 standard convention.

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Inline pumps have been developed, in part, to avoid the problem of misalignment due to nozzle loads. Figure 6 shows a single stage, overhung, inline pump with a separate bearing frame, Type OH3. Figure 7 shows a high-speed integrally geared inline pump, Type OH6. For a given flange size, inline pumps are required to withstand twice the magnitude of nozzle loads allowed for horizontal pumps. In addition, installation is simplified and less expensive because a block foundation is not required, and the pump mounts in a pipeline

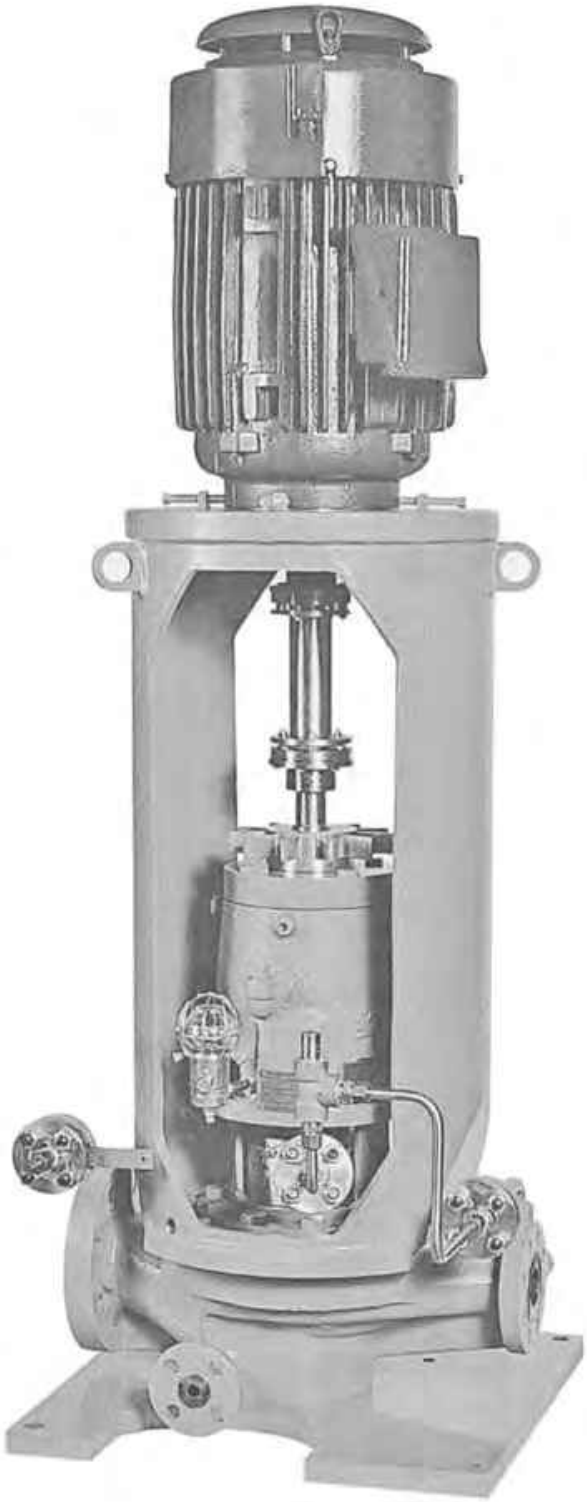


FIGURE 6 Single-stage, overhung, vertical inline pump with separate bearing frame—API 610 Type OH3 (Flowserve Corporation)

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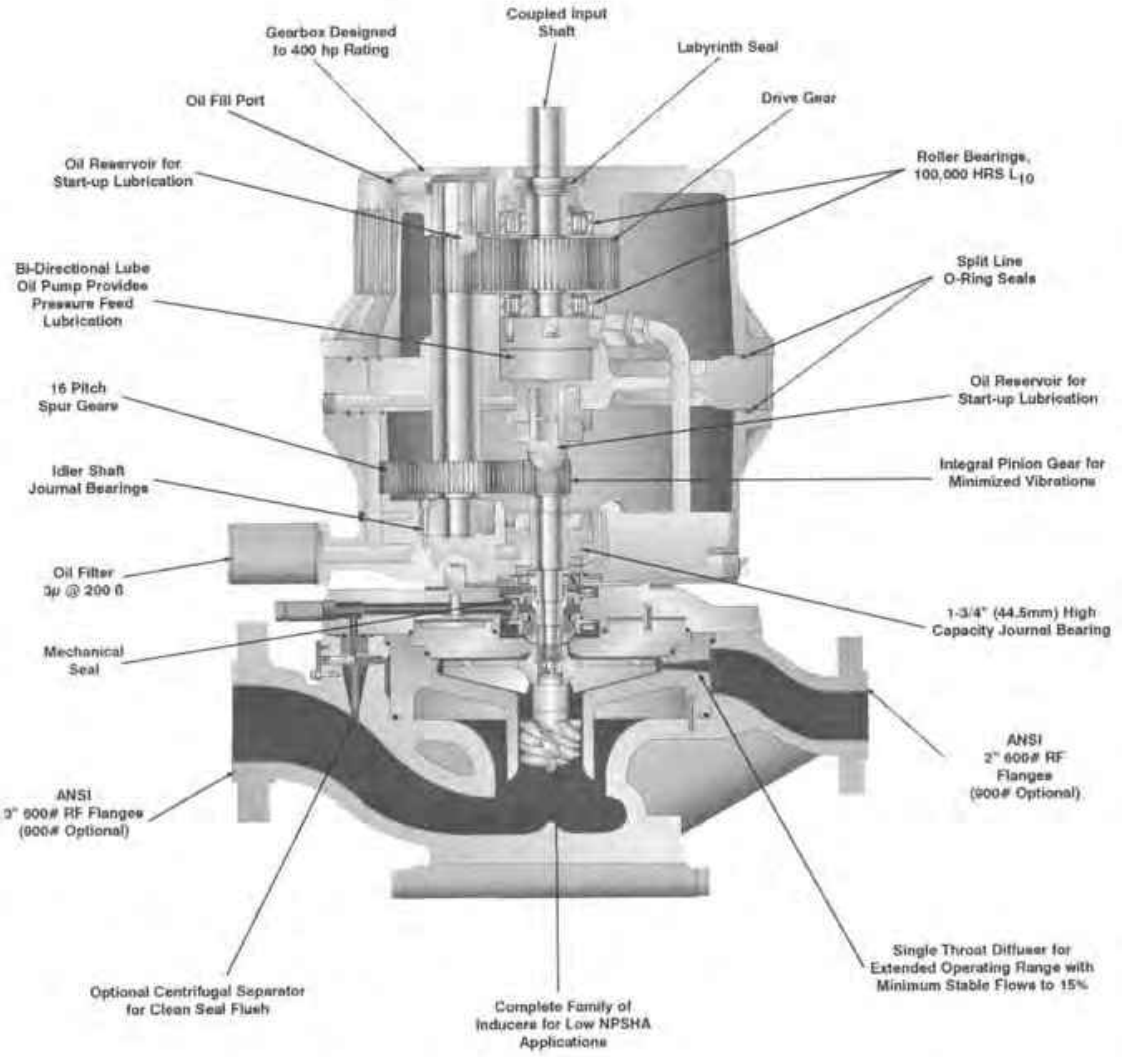


FIGURE 7 Integrally geared inline pump—API 610 Type OH6 (Sundyne Corporation)

similar to a valve. It is noted that an in-line pump is the equivalent of a very heavy valve, and most users supply a support pad or other support arrangement beneath the pump. Furthermore, the vertical arrangement causes the pump to take up much less space. The Type OH6 pump (Figure 7) has a gear box that increases the speed at which the impeller spins and typically produces high heads at relatively low flows. This type of design has many advantages in certain low flow, high head services. Some alternative inline pump designs utilize high-speed motors that eliminate the need for the gearbox. High-speed pumps may also be supplied with inducers in front of the typically radially bladed impellers to improve suction (NPSH) performance. When this is done, the range of operation of the pump may be restricted to avoid off-design flow instability. This should be recognized and the required range of pump operation should be addressed during the applications stage of pump selection for such pumps.

Pumps of overhung shaft construction are nominally limited by most manufacturers to drivers rated below approximately 500 HP (375 kW). Units with greater power requirements are usually designed with bearings on both ends of the shaft and the impeller—or impellers—in between the bearings (designated by API 610 as between bearings, or Type BB, pumps). Ball bearing construction, in compliance with requirements of API 610, is used to a limit of a bearing Nd_m factor of 500,000. The Nd_m factor is the product of the pump operating speed (N), in revolutions per minute and the mean bearing diameter (d_m), equal to the bearing bore plus the bearing outside diameter, divided by 2 (all dimensions in millimeters). For values of Nd_m above 500,000, or where the bearing basic life rating (L_{10h}) does not meet the minimum life requirements mentioned earlier, hydrodynamic

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bearings are required. This may result in a pump with sleeve radial and rolling element thrust bearings, or with both hydrodynamic radial and thrust bearings. API 610 also establishes an energy density limit of four million, above which hydrodynamic radial and thrust bearings are always required.

To ensure safety and reliability, it is an API 610 requirement that pump pressure casings and flanges be designed for maximum pump discharge pressure plus allowances for head increases at the rated pumping temperatures. API 610 also establishes minimum pressure ratings for all covered pumps. For horizontal radially split pumps, the minimum pump pressure rating is that of an ANSI/ASME B16.5 Class 300 (ISO 7005-1 PN50) steel flange of the material grade corresponding to that of the casing, or 600 psig (4 MPa), whichever is less. Furthermore, all pumps are subjected to a hydrostatic test at 1.5 times the case rating. It is a requirement for one- and two-stage pumps that the suction and discharge flanges have the same pressure rating and that the casing have a minimum pressure rating equivalent to the flange rating.

Performance One- and two-stage centrifugal pumps for API services cover a wide range of operating conditions—up to 1200 ft (365 m) of head and 11,000 gpm (2500 m^3/h), varying with manufacturer. Vertical in-line pumps have a range similar to that of their horizontal single-stage counterparts. Multistage centrifugal pumps are available for both large flow and extremely high heads, probably exceeding anything we could state here by the time this handbook is published.

API 610 requires a performance test in accordance with Hydraulic Institute (HI) standards for all pumps. The only deviation from HI standards is that efficiency is calculated for information and is not a rating or guarantee value. Power is the guarantee quantity and includes all losses (such as for mechanical seals and bearings). Vibration levels are taken during the test and acceptance criteria are established for both bearing housing and shaft measurements on the pump manufacturer's test stand. In general, refiners have lowered vibration levels in their plants through conscientious maintenance practices, particularly in the areas of balance and alignment. The vibration acceptance criteria are based upon approximately 150% of the average vibration of spared pumps in one of the larger refineries in North America in the early 1990s. Therefore, the vibration acceptance criteria are somewhat more liberal than typical plant values because of the temporary nature of a manufacturer's performance test setup. If critical service suction conditions warrant it, an NPSH test may also be conducted.

Materials Refinery pumps handle a variety of products with specific gravities ranging from 0.3 to 1.3 and viscosities from values below that of water to as high as 15,000 SSU (3240 centistokes) for centrifugal pumps and even higher for rotary pumps, over a wide range of temperatures. The product may be as innocuous as a lubricating-type oil or extremely corrosive. Many materials are utilized to satisfy these varied product requirements; the most common material classes are given in Table 1 (API 610 Table G-1: Material Class Selection Guide). Material classes and specifications are given in Table 2 (API 610 Table H-1: Materials for Pump Parts).

Drivers Drivers for refinery pumps are usually either electric motors or steam turbines. Centrifugal pumps running backward as hydraulic turbines are also used. The vast majority of refinery pumps are run at a fixed, 2-pole speed of either 3600 rpm in 60 cycle locations or 3000 rpm in 50 cycle locations.

PIPELINE PUMPS

Centrifugal pumps are used on every major liquid pipeline in the world to transport a variety of fluids, including crude oil, motor gasoline, fuel oil, jet fuel, liquefied petroleum gases, and anhydrous ammonia. Pump efficiency is of primary importance because of the power required to transport the liquid. Most pipeline systems install pumps in series because the differential head requirement is primarily energy loss due to friction, and an

TABLE 1 Material class selection guide (Courtesy of the American Petroleum Institute)[1]

Service	Temperature range °C	Temperature range (°F)	Pressure range	Materials class	Ref. note
Fresh water, condensate, cooling tower water	<100	<212	All	I-1 to 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 feed water					
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 reformate, 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

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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_2S)	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
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

[a]The materials for pump parts for each material class are given in Annexe H.

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.

[a]Oxygen content and buffering of water should be considered in material selection.

[b]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).

[c]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.

[d]All welds shall be stress-relieved.

[e]UNS N08007 of Ni-Cu alloy pump material should be used.

[f]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.

[g]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).

[h]Materials selected for low-temperature services shall meet the requirements of 5.12.4 and 5.12.1.6. Casting alloy ASTM A 352, Grades LCB, LC2, & LC3 is shown only for reference. Use equivalent materials for wrought alloys.

[i]Material alloys based on aluminium, bronze, aluminium bronze, and nickel, may also be considered for temperatures as low as −196°C (−320°F).

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TABLE 2 Materials for pump parts (Courtesy of the American Petroleum Institute)[1]

PART	Full compliance materials[b]	Material classes and abbreviations													
		I-1	I-2	S-1	S-3	S-4	S-5	S-6	S-8[l]	S-9[l]	C-6	A-7	A-8	D-1[j]	D-2[j]
		CI[a]	CI	STL	STL	STL	STL	STL	STL	STL	12 % CHR	AUS	316 AUS	Duplex	Super Duplex
		CI	BRZ	CI	NI-RESIST	STL	STL 12 % CHR	12 % CHR	316 AUS	Ni-Cu Alloy	12 % CHR	AUS[c, d]	316 AUS[d]	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
Interstage sleeves[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]
Interstage bushings[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]
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[i]	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]

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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 [h]	Bronze	Filled carbon	Nitrile butadiene [h]	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	316 AUS	316 AUS	Ni-Cu Alloy	316 AUS	316 AUS	316 AUS	Duplex	Super Duplex

[a] The 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

[b] See 5.12.1.4

[c] Austenitic stainless steels include ISO Types 683-13-10/19 (AISI Standard Types 302, 303, 304, 316, 321, and 347).

[d] For 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).

[e] Unless 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 non-galling materials or non-metallic materials, depending on the corrosiveness of the pumped liquid.

[f] For 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).

[g] If 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.

[h] Alternative materials may be substituted for liquid temperatures greater than 45 °C (110 °F) or for other special services.

[i] Unless otherwise specified, AISI 4140 steel may be used for non-wetted case and gland studs.

[j] Some 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 $\geq$ 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.

[k] Non-metallic 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).

[l] 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).

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outage of one of the units would result in only a partial loss in the throughput capacity. For pipelines where the differential head is mostly static, such as in mountainous areas, pumps are often installed in parallel. A series installation arrangement in a static system would be unsuitable because the differential head required could not be obtained unless all of the units were operating. In certain cases, however, where differential head requirements exceed levels practical for a single multistage pump, a combination of series and parallel arrangements may be desirable.

Construction A single-stage double-suction pump, a two-stage double-suction pump, and a four-stage pipeline pump are shown in Figures 8, 9, and 10, respectively. All the units are double volute and are of axially split construction. The single-stage double-suction unit (Figure 8) is typical of those used in series operation on large diameter pipelines with capacities as high as two million barrels per day. One pipeline alone has installed more than one hundred 5000 HP (3730 kW) pumps of this construction. The two-stage double-suction pump (a smaller unit is shown in Figure 9 with an engine driver) is typical for higher head applications, and sizes are available for power ratings to 20,000 HP (15,000 kW). For lateral and smaller diameter pipeline pumping stations, multistage units (Figure 10) are used. The number of stages chosen corresponds to the differential head required. Multistage units may be destaged initially to produce lower heads, with a subsequent power saving, or the nozzling may be arranged for series or parallel operation of a portion of the stages. The exact arrangement depends on the system characteristics and initial and ultimate capacities to be pumped.

All units are equipped with sleeve radial bearings and either ball or hydrodynamic thrust bearings, as dictated by power (size) and rotating speed. The single- and two-stage double suction pumps are inherently balanced axially. The multistage units utilize opposed impeller design to obtain axial balance. All modern units are equipped with mechanical seals in the seal chamber. Because most pipeline stations are now designed for unattended outdoor service, a number of safeguard controls are used. These include warnings for low suction pressure, high discharge pressure, high bearing or case temperature vibration, and excessive seal leakage. Tank farms utilize single- and double-suction inline vertical pumps or, if the tank is to be pumped dry, vertical canned pumps.

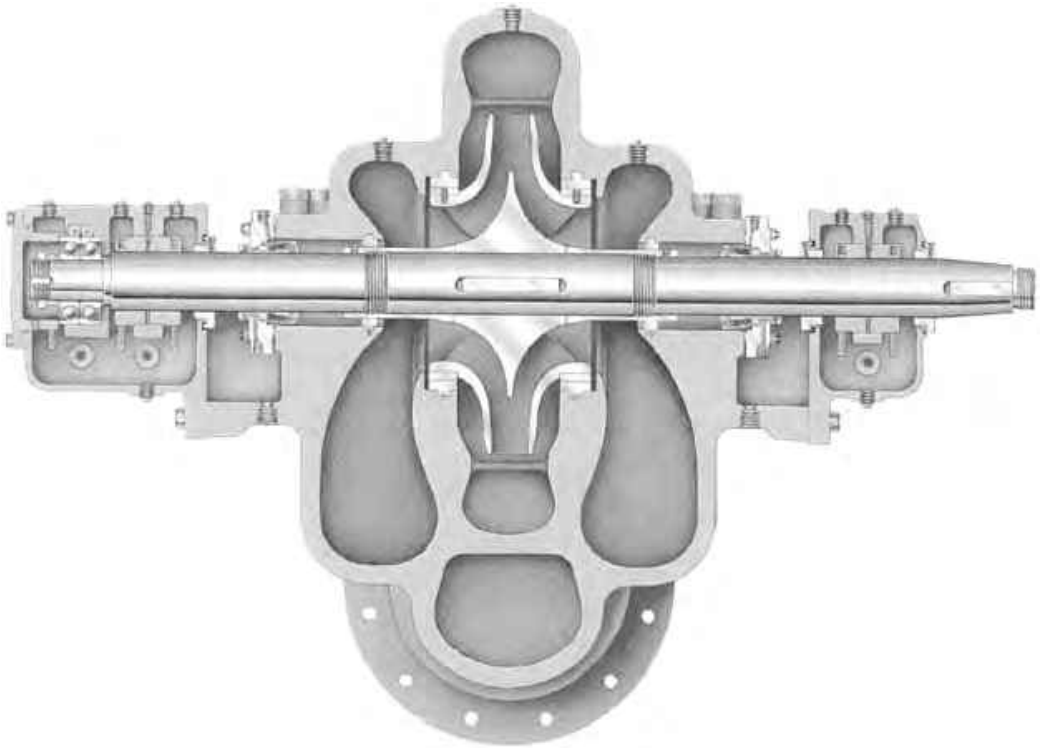


FIGURE 8 Single-stage double suction, axially split pump (Flowserve Corporation)

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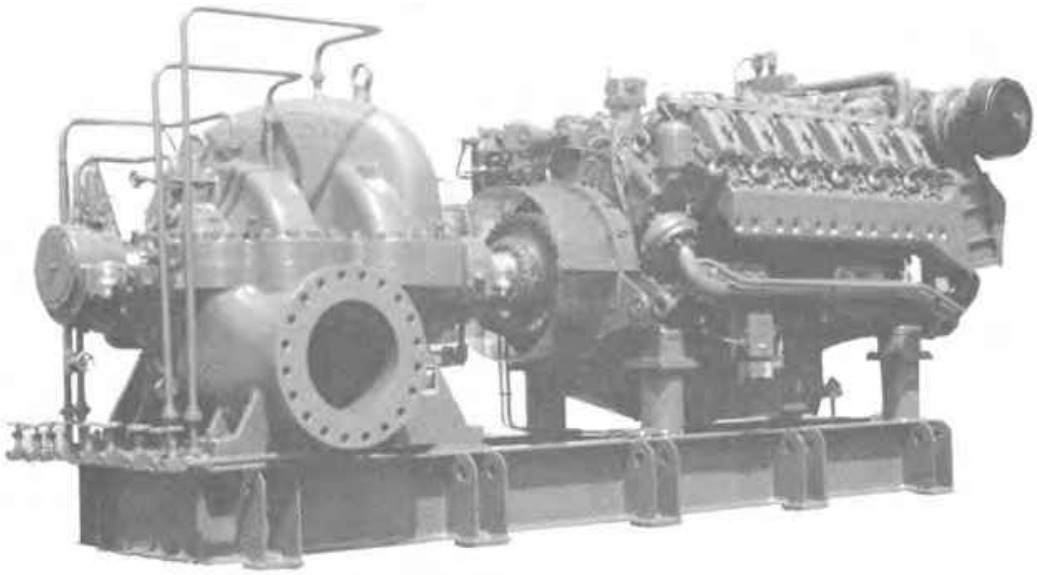


FIGURE 9 Two-stage, double suction pump with engine drive (Flowserve Corporation)

Performance The typical available performance range for single- and two-stage pipeline pumps is up to 1200 ft (365 m) of head and 8000 gpm (1800 m^3/h). For multistage units, the range is up to 8000 ft (2,300 m) and 10,000 gpm (2400 m^3/h). Pipeline units are custom designed, and the range is constantly being extended. Electric motors are normally used as drivers. With the utilization of gas turbines, the maximum power employed has increased dramatically in the last 30 years. Another use of gas turbines on existing pipelines is for peak capacity booster station service, in which case pumps are direct-driven at operating speeds as high as 6000 rpm.

Materials Pump part materials commonly specified for crude and product pipeline services are shown in Table 2 under columns S-1 through S-9. For the more aggressive services and liquids, the higher numbered material columns are preferred. All these material columns utilize carbon steel pump casings; the impellers, wear rings, and shaft materials are changed as the material grades increase in alloy and hardness (wear rings) with advancing material column grade number.

Pumping Stations The typical piping of a main line pump station in which the units are arranged in series is shown in Figure 11. As pipeline capacities have increased, one of the major problems has been the pressure loss at each station. For the station shown, three discharge valves (one 16 in (406 mm) ball valve manifolded in parallel with two 24 in (610 mm) control valves) were used. This arrangement limited the calculated pressure drop to that of a length of 36 in (914mm) pipe equal to the distance across the manifold and still allowed the use of control valves of proven size. When line conditions require throttling, the 16 in (406 mm) ball valve is first completely closed. If additional throttling is needed, the two 24 in (610 mm) control valves are closed to produce the required pressure drop. In order to move scrapers or batch separators through each station without interrupting flow through the pumps, signals (PIG SIG), hydraulically operated 24-in (610 mm) valves, and sequence control wiring are used. The distance between PIG SIG 1 and PIG SIG 2 represents the volume of station loop to be displaced. The tripping of PIG SIG 3 and PIG SIG 4 controls the valve opening and closing required to divert flow from the station discharge piping to behind the scraper or batch separator, thus forcing it to leave the station in the same relative position as it entered. Elbows at pump suction are arranged to avoid uneven flow distribution in the inlet of the double suction impeller.

Pipeline pumps are tested in accordance with HI standards. Reduced speed tests may be required because of the power limitations of manufacturing plants and available drivers, and such tests have proven to be extremely accurate indicators of full speed performance.

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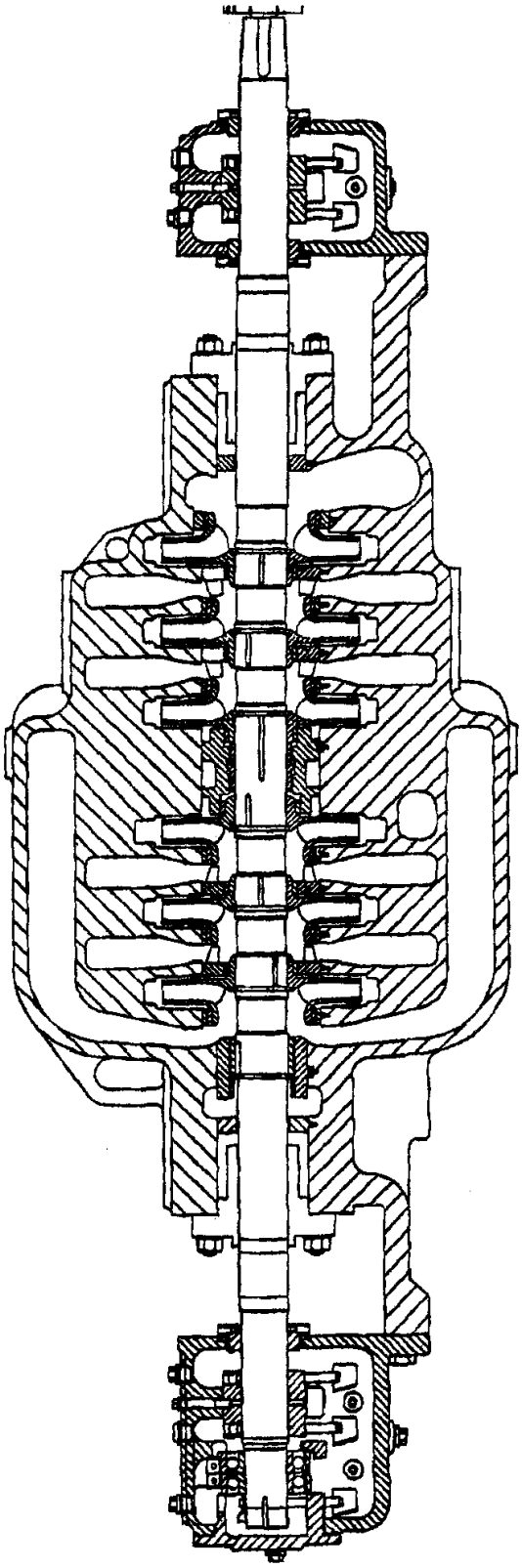


FIGURE 10 Single-suction, double-volute, axially split, six-stage pipeline pump (Flowserve Corporation)

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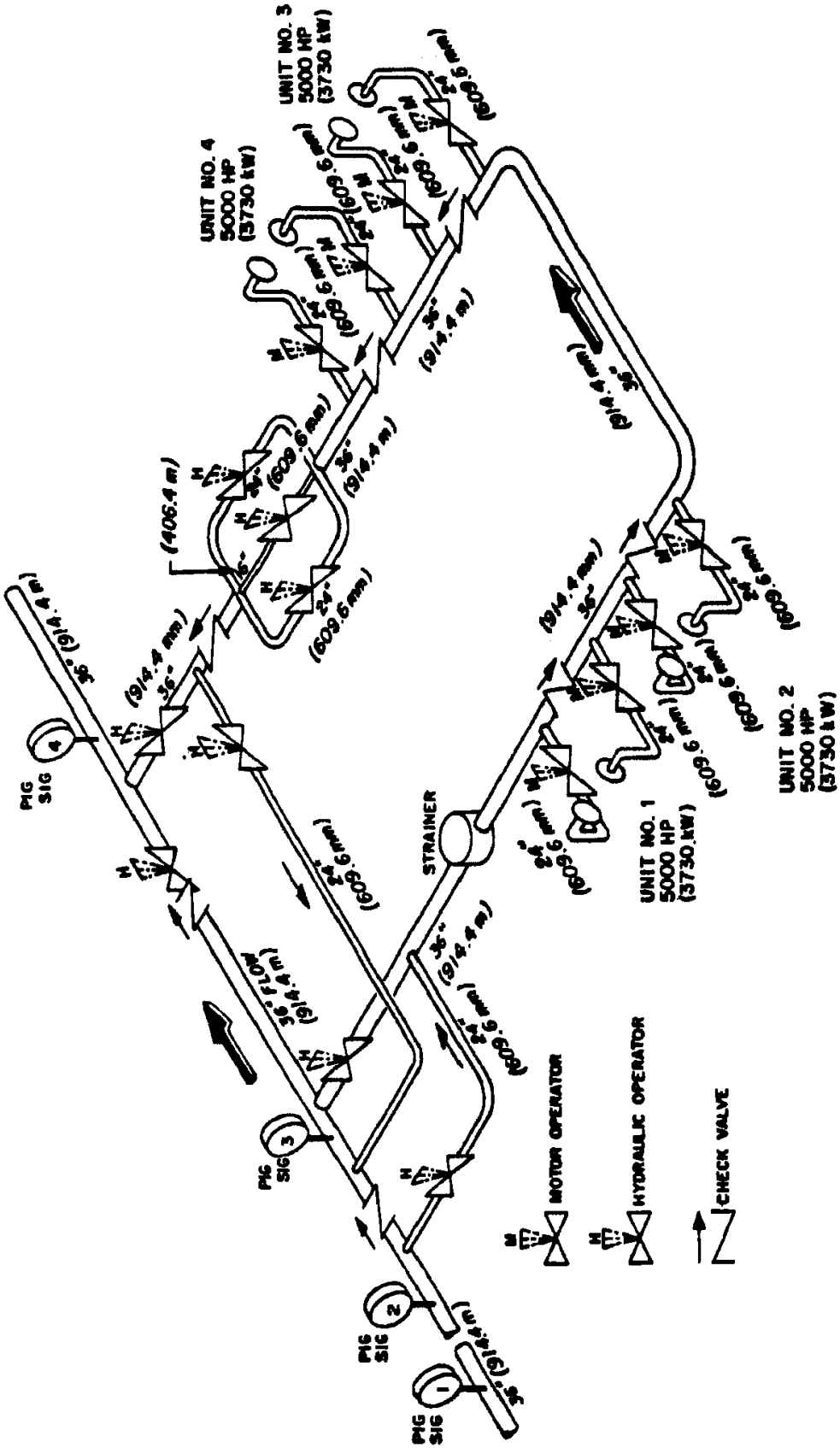


FIGURE 11 Main line pumping station piping layout

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SPECIAL SERVICES

Waterflood Centrifugal pumps are also used for water injection service when the capacity required is greater than 10,000 barrels per day. Injection pressures requirements vary considerably from initial startup with operating time. For high-pressure services, usually above 2500 lb/in^2 (17Mpa), barrel-type pumps are used.

Waterflood, or reinjection, pumps are among the most technically interesting API pumps built in significant quantities today. These pumps push the envelope of what has been accomplished in centrifugal pumps. Barrel-type waterflood pumps (Figure 12) have been build up to 8500 lb/in^2 (586 MPa) discharge (equivalent to 20,000 ft/6000 m of head), 16,000 gpm (3600 m^3/h), with power reaching 36,000 HP (27 mW), at speeds up to 11,000 rpm. The barrel construction is typically of forged steel with an austenitic stainless steel welded overlay applied in areas of high velocity. Barrel-type casings are hydrostatically tested to 1.5 times the rated pressure so to as much as 12,750 lb/in^2 (863 MPa). At these high pressures, slightly lower test pressures are sometimes negotiated.

Most waterflood injection pressure requirements, however, are such that axially split multistage pumps of the design shown in Figure 4 may be used. As many as 14 stages in one pump are often utilized to obtain the required differential head. While many examples of very reliable pumps with these large numbers of stages can be found, most users are moving away from pumps with more and eight or ten stages. In addition, if still greater pressures are required, the units may be piped in series. Axially split pumps have been hydrostatically tested to 14,000 lb/in^2 (96 MPa). For such instances, it is becoming more common to hydrostatically test at 1.3 times the maximum allowable working pressure of the pump, following the latest ASME code requirements.

For applications where injection liquid is highly corrosive, materials of construction may include austenitic and duplex stainless steels. Commonly used materials for major pump components in waterflood applications are shown in Table 2 under material columns A-8 and D-1. Seal chambers for high pressure and high-speed services may contain a limited leakage breakdown bushing preceding a mechanical seal for increased reliability.

Reactor Feed Pumps One of the most difficult services encountered in the petroleum industry is the high-temperature, high pressure reactor feed, or charge pump. Type BB5 pumps, such as the one shown in Figure 5, are most commonly used for these services. Reactor feed pumps can be subject to operating temperatures in excess of 800°F (425°C) with discharge pressures up to 3000 lb/in^2 (20 MPa) gauge. To produce such high pressures requires either a large number of stages or high speed. Historically, refiners have preferred 3600 rpm (3000 rpm in 50 Hz areas) pumps with large numbers of stages. However, the trend more recently is for shorter, more robust pumps, with fewer stages operating at higher speeds. The trend for waterflood pumps is identical.

The high temperatures and corrosiveness of the pumped fluids often require the use of a number of dissimilar metals. Compensation must be made for the different rates of expansion of the various components at operating temperatures. This presents a critical design problem for the manufacturer and a significant operability issue for the refinery. Extreme care is required in the startup and operation of these units. Even so, it is common for refiners to buy only a single pump for such services (no spare), and for these pumps to run continuously for five or more years without maintenance.

LPG Pumps Liquefied petroleum gases (LPG) are pumped from their point of production to points of use. The pump services involved are product pumps (from production into a pipeline), pipeline (located along the pipeline as necessary to maintain the desired flow), and reactor feed when the particular LPG is the feed for a petrochemical process. API 610 type BB3 and BB5 pumps are used, typically BB5 for product and reactor feed, BB3 for pipeline. Rated discharge pressure for product and pipeline pumps today is typically 1800 to 2000 psig (125 to 140 MPa). Reactor feed pumps rated for discharge pressures on the order of 4500 psig (310 MPa) are in service. The difference in pump type with service is determined by location: on plot installations, product, and reactor feed, usually follow API

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FIGURE 12 Double-casing multistage barrel-type pump with radially split inner casing (Flowserve Corporation)

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610's guidelines on casing construction for pressure, SG and temperature quite faithfully, whereas pipeline operators have successfully used BB3 pumps for rated pressures to 2000 psig (140 MPa), SG down to 0.35. Beyond casing construction, there are two other critical aspects of LPG pumps: rotor design and shaft seal type.

Rotor design falls into two classes, slender shaft and large shaft. The former have static deflection greater than the internal running clearances and, therefore, rely on the Lomakin effect for support during operation. Because the rotor rubs in its running clearances during start-up and shutdown, and LPG has poor lubricity, special materials are used for the running clearances to tolerate the rubbing. The alternative is a large-shaft rotor designed to operate between its first and second dry bending critical speeds and have a finite clearance with its running clearances when the pump is shutdown. Provided one surface of the running clearances is serrated to accommodate a "touch," normal pump materials can be used. Large-shaft rotors are the same as those being employed in modern design reactor feed pumps for refining.

Shaft sealing of LPG has been rendered far simpler than it used to be since the development of vaporizing liquid dry gas seals. The usual design is a tandem seal with filtered seal gas upstream of the primary seal, nitrogen sweep through a labyrinth between the secondary and primary seals with primary seal leakage plus portion of the nitrogen to flare. The secondary seal normally runs on nitrogen. Seal gas, nitrogen, and primary seal leakage flows are monitored to alarm on high leakage, shut down the pump on high-high leakage. These seals, correctly engineered into the pump, have demonstrated 5 years operation at sealed pressure on the order of 1000 to 1500 psig (70 to 105 MPa).

PUMPING OF VISCOUS LIQUID

Viscosity of the pumped liquid is a very important factor that must be considered to properly select and size pumps in many petroleum industry applications. A thorough understanding of the relationship between liquid viscosity and pump performance is essential to proper sizing of both the pump and its driver.

Centrifugal pumps are routinely applied on services with liquids having viscosities below 3000 Saybolt Seconds Universal (SSU) or 660 centistokes (cSt), and may be used up to at least 15,000 SSU (3300 cSt). (For background information on viscosity, refer to Section 8.1.) Centrifugal pumps are sensitive, however, to changes in viscosity and will exhibit significant reductions in capacity and head and rather drastic reductions in efficiency at moderate to high values of viscosity. The extent of these effects may be seen in Table 3.

The pump manufacturer nearly always specifies centrifugal pump performance basis pumping clean, cold water, even when the pump has been specifically designed for petroleum industry applications. Cold water is a universal pumping medium. Pump selections, however, must necessarily be made to satisfy viscous conditions of service and require application of these correction factors in the reverse direction.

TABLE 3 Effect of viscosity on performance of a typical centrifugal pump operating at best efficiency point

Viscosity SSU (cSt)	Capacity gpm (m^3/h)	Total head ft (m)	Efficiency %	Brake power bhp (kW)[a]
Nil	3000 (681)	300 (91)	85	241 (180)
500 (110)	3000 (681)	291 (89)	71	279 (208)
2000 (440)	2900 (658)	279 (85)	59	312 (233)
5000 (1100)	2670 (606)	264 (80)	43	373 (278)
10,000 (2200)	2340 (531)	243 (74)	31	417 (311)
15,000 (3300)	2100 (477)	228 (69)	23	473 (353)

[a]All values of brake power based on liquid having a specific gravity of 0.90.

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The Hydraulic Institute Standard 9.6.7 (Reference 5) may be used to calculate viscous effects on pump performance for radial flow centrifugal and vertical pumps with open or closed impellers, handling Newtonian liquids within the pump's normal operating range and within the capacity limits stated in the standard.

DISPLACEMENT PUMPS Both rotary and reciprocating pumps are also frequently used for pumping viscous fluids, and some types are well suited for use where viscosities are beyond the limits for centrifugal pumps. In fact, many common designs are suitable only for use with liquids that are at least moderately viscous because they depend on the viscosity to maintain the lubricating and sealing films between various internal parts of the pumps. Most gear pumps and screw pumps are in this category.

As with centrifugal pumps, the performance of displacement pumps may be significantly affected by changes in liquid viscosity, but the nature of these changes may be quite different. At constant speed, changes in viscosity are likely to have little or no effect on pump capacity. Total head, or differential pressure across the pump, would probably increase with increasing viscosity because of higher system resistance. Thus, the brake power required would increase with even though pump efficiency would not suffer nearly so drastically, if at all, as for a centrifugal pump.

The influence of pump design on how performance varies with viscosity is greater in displacement pumps than in centrifugal pumps. Because there are so many designs, it is not practical to attempt to provide a general means of determining their viscous performance here. This should be done only on the basis of information provided by the pump manufacturer.

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SECTION 12.9
PULP AND PAPER MILLS

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GLOSSARY

accepts Pulp which has passed the screening process and is usable.

additive An additive is any material such as clay, filler, or color added to stock to contribute specific properties.

air dry Pulp which contains 10% moisture.

alkaline pulping Also refers to kraft process pulping or use of caustic; contrasts with the acid sulfite process.

alum Hydrated salts of aluminum sulfate used in papermaking process.

black liquor Liquor containing dissolved organic materials and residual chemicals resulting from alkaline cooking process.

bleaching Bleach is the process of removing the lignin and other color-forming substances from the stock to render it white.

blow Pulp stream discharged from digester when under pressure to atmospheric pressure.

bone dry Also referred to oven dry; contains no moisture.

brightness Indicates degree of whiteness; the reflectivity of a sheet of pulp or paper based on standardized conditions.

brown stock Pulp before bleaching.

cellulose Cellulose is a carbohydrate that is the primary substance in plant fibers. It is extracted from the plant to make paper products.

chest A chest is a vessel for storing pulp.

consistency Consistency is the proportion by mass of fibers in a mixture of fibers and water.

cooking Cooking is the action of the chemical used to break down lignin bond between cellulose fiber in wood and other organic matter.

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deinking Deinking is the process of removing ink particles from recycled paper such as newsprint or magazines.

digester A digester is a pressure vessel used to contain chemical action of cooking chemical and raw cellulose material. Digesters may be of the batch or continuous flow type.

dry end The end of the paper machine where the paper is dried, calendered (imparting a surface finish), and reeled.

evaporator In the recovery of chemicals from the cooking liquor in a chemical pulp mill, an evaporator is used to concentrate the liquor (remove water) so the concentrated liquor can be incinerated in the recovery furnace.

fiber Fibers are the vascular bundles found in plants that are extracted and used to make paper.

fourdrinier Fourdrinier is the continuous wire upon which pulp or paper sheet is produced.

freeness Freeness is a measure of degree of refinement of stock and, hence, of its ability to drain water.

furnish Mixture of raw materials which includes pulp and chemicals used to make a specific grade of paper.

groundwood Groundwood is a mechanical pulp formed by simple grinding to break down wood structure.

headbox A headbox is a specialized nozzle that takes flow from the fan pump and discharges a uniform layer of white water (stock) onto the forming wire of the paper machine.

kraft process The kraft process is a method of separating cellulose fiber from lignins by using caustic soda in presence of sulfur radical.

lignin Lignin is a generic term used to refer to the complex organic matter present in wood that acts as binding agent for cellulose fibers.

liquor *Black liquor:* Solution of water plus residual organic matter (or lignins) in the wood after washing of raw stock. *Green liquor:* Solution of smelt from recovery furnace when dissolved in either water or weak washed liquor. *White liquor:* Solution of caustic soda (or other alkali), sometimes in the presence of a sulfur radical; this is the liquor charged to the digester for cooking.

market pulp Pulp sold as a raw material.

neutral sulfite Neutral sulfite is made through a cooking process using a solution of about 10% sodium sulfite and 5% caustic soda mixed for cooking wood or agricultural fiber to produce high-yield pulp.

oven dry Also referred to as bone dry; contains no moisture.

paper Paper is a sheet of cellulose or other fibers formed on a screen from a mixture of fibers in water. Thin absorbent paper is known as tissue; thicker grades of paper are called paperboard.

pulpwood Logs trimmed to size for use by a pulp mill.

refining Refining is a mechanical process carried out on stock to improve the ability of the fiber to form paper sheet. Different techniques are employed—some designed to shorten the fiber, others to increase the amount of fibrils (or "whisker") on the fiber.

shives Bundles of fibers which have not been separated during the pulping process.

soda process A soda process is similar to kraft process but uses caustic soda without presence of sulfur.

stock Stock is a generic term for the suspension of cellulose fiber in water or chemicals. *Bleached stock:* The same after bleaching. *Brownstock:* The same before bleaching. *Raw stock:* The product discharged from digester(s) before any washing or other treatment. It is also referred to as pulp.

sulfite process A sulfite process is the method of separating cellulose fiber from lignin with acid. This method is becoming obsolete because of its inherent detrimental impact on the environment.

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vat A vat is a semicylindrical mold or container for holding stock during washing or sheet formation.

yield Yield is the percentage of cellulose fiber in the form of pulp produced from a given weight of wood or other raw material. *High yield:* The same when some lignins are allowed to remain in the finished product.

Common Conversion Factors

AD = Air Dry Stock (Contains 10% water)
OD = Oven Dry Stock (All water removed)
Also called Bone Dry (BD)
AD = 1.11 × OD
OD = 0.90 × AD
ADT/D = 1.11 × ODT/D
ODT/D = 0.90 × ADT/D
AD Consistency = 1.11 × OD Consistency
OD Consistency = 0.90 × AD Consistency
T/D or TPD or ST/D = Short Tons per Day (Short ton = 2000 lb)
MT/D = Metric Tons per Day (Metric ton = 2205 lb)
ADST/D = Air Dry Short T/D (adstpd)
ADMT/D = Air Dry Metric T/D (ADMtpd)
ADST/D = 1.1025 × Metric T/D
ADST/D = 1.11 × BDST/D
ADST/D = 1.225 × BDMT/D
Production rate normally expressed as ADT/D
Consistency normally expressed as OD% (or BD%)

$$\text{Flow Rate (GPM)} = \frac{P \times 15}{C}$$

where
P = Production Rate in ADST/D (adstpd)
C = % Consistency (OD or BD)

$$\text{Flow Rate (l/s)} = \frac{P \times 1.04}{C}$$

where
P = Production Rate in ADMt/d
C = % Consistency (OD or BD)

GENERAL

Apart from the petrochemical industry, there are few continuous-process plants with pumps as much in evidence and with reliability as essential as in the pulp and paper industry.

Before paper leaves the machine room, 100 to 200 tons of water will have been pumped to the mill for every ton of paper produced. These figures represent only the basic amount of water taken from a river and rejected as effluent; the amount of liquid circulated is several times greater.

Although many attempts have been made to utilize a dry process for papermaking, both pulping (the separating of crude fibers from the raw material), and papermaking (actual treatment of the fiber and mechanical formation of the sheet) still require water as the medium to convey fiber. Throughout the mill, pumps are used to transfer or circulate

fibers suspended in water (stock), chemicals or solids in solution (liquors), or residues and waste matter as slurries, as well as to supply water for general services.

There are at least 150 pumps installed in a modern pulp mill and another 50 or so in a paper mill. About 1000 kW · h is required for the production of one ton of pulp, and a further 500kW · h for the finished paper. Of this total, about 25% is used in pumping. This means that even in a medium-size mill, the installed power required for pumps alone is approximately 10,000 HP (7457 kW), and often it is much higher.

PULPING PROCESS

Raw Materials In the past, the traditional fibrous raw materials for the manufacture of paper were cloth and agricultural residues; today the vast bulk of cellulose pulp is made from wood. Although over half of this is produced from softwoods (long-fibered), an increasing amount is being now produced from hardwoods (short-fibered). Traditional raw materials such as linen, cotton waste, straw, and agricultural residues are still used in small quantities, particularly where the paper sheet requires special properties. These distinctions are important in the selection of pumps, for the liquors have different characteristics. For example, straw black liquor is much more viscous than wood black liquor, and the proper corrections must be made in calculating the pump performance and pipe frictional losses.

Recycle Fiber Many grades of paper include fiber recovered from waste paper—both pre- and post-consumer. Recently, because of environmental concerns, much emphasis has been placed on recycling post-consumer paper products. Specialized plants exist for the recovery and processing of recycled fiber before it is used for papermaking. Some of the different types of recycled fiber are ONP (old newsprint), OCC (old corrugated containers), and MOW (mixed office waste). Special care is needed in the selection of pumps to handle recycled stock because of the large amount of foreign matter present—rope, string, metal, synthetic fibers, and adhesive materials—all of which can cause problems in the process and pumping. Deinking is also widely used in the processing of recycled fibers. Flotation cell deinking, a popular method, uses large amounts of entrained air, which has a dramatic effect on centrifugal pump performance and must be accounted for.

Groundwood Pulp This type of pulp is produced by simply grinding away wood by mechanical action. Almost all of the wood is used in the pulp, including many of the resins and other complex organic compounds. The fibers are bruised so the pulp has inferior strength.

Large amounts of water are required for cooling and for carrying away the groundwood pulp; the latter is usually acidic (pH 4 to 5), and so corrosion-resistant materials must be used. The pulp is used primarily for newsprint and magazines. Depending on the end use, some mechanical treatment (refining) may be required to alter the characteristics of the pulp, particularly the viscosity. In some cases, a mild bleach may be used to improve the color.

Refiner Mechanical Pulp The refiner mechanical pulping process utilizes a disc refiner to reduce wood chips to fibers. It produces a longer fibered pulp than conventional groundwood, but not as long as the chemical pulps. The pulp is therefore somewhat stronger and freer than SGW, but not nearly as strong as kraft pulp. RMP is actually the first of many processes utilizing the disc refiner to produce pulp, The processes vary as to the number of refining stages, refining pressure, temperature, and pre-treatment of the chips (steaming, chemical pre-treatment, and so on). Two of the more popular versions are TMP (thermomechanical pulp), in which chips are pre-softened with steam, and CTMP (chemithermomechanical pulp), in which they receive an additional chemical pretreatment. RMP pulps do not require a rigorous bleaching process in contrast to chemical pulps.

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Chemical Pulping Wood is a complex, nonuniform material containing about 50% by weight cellulose fiber, 30% lignins, and 18 to 20% carbohydrate. The remainder is proteins, resin, and other complex organic compounds that vary from one species to another. Cellulose resists attack from most chemicals, whereas the carbohydrates and other organic materials generally form compounds with the chemical cooking liquor. Some paper products can use the carbohydrate fraction to contribute bulk to the sheet, and for such papers groundwood and RMP and other mechanical pulps are used. Where high strength is required, cooking is necessary to separate the fibers completely from the remainder of the wood.

Most cooking of wood is done in a pressure vessel at high temperature and pressure in the presence of an acid or alkali.

There is considerable tradition in chemical pulping, and a number of different processes are used. For many years the traditional method of producing pulp for high-grade papers was the acid sulfite process. This has been largely superseded in recent years by an alkaline process using sodium-based liquors in the presence of a sulfur radical; this is known as the sulfate or kraft process. The main reasons for the change to the sulfate process have been lower corrosion rates, ease of chemical recovery, and a stronger pulp. The properties of the liquids pumped in the two processes are different, and the pumps require different materials of construction.

Typical Sulfate Process Pulpwood logs are first chipped to about $\frac{3}{4}$ by $\frac{1}{8}$ in (19 by 3 mm) and then charged into either a continuous digester or a series of batch digesters. Digester capacities range from approximately 100 air-dried tons per day to over 2000 air-dried tons per day, necessitating a wide range of hydraulic coverage for digester pumps. Cooking liquor (NaOH plus up to 30% Na_2S) is then allowed to react with the wood chips for 2 to $2\frac{1}{2}$ h at a temperature up to 350°F (177°C) and a pressure in the digester of 80 to 100 lb/in^2 (551 to 689 kPa). In many mills, the heating of the chips and cooking liquor is by direct steam injection to the digester. In others, some form of indirect heating is used with a closed liquor recirculation system. In the latter case, the digester circulating pumps are a critical item because they must handle hot caustic solutions and entrained solid matter in a closed, pressurized circuit. After cooking, the contents of the digester are discharged to atmospheric pressure into a vessel called the blow tank, where the sudden expansion causes the fibers to separate from the liquid, which is now known as black liquor.

At this point, the process splits into two streams—one for fiber processing and the other for chemical recovery. The fiber is washed and screened and then formed into a pulp or paper sheet. The black liquor is washed from the pulp and treated for chemical recovery. Because the most troublesome liquors are to be found in the recovery process and bleach plant, the selection of these pumps is critical for the successful operation of the process.

The chemistry of the recovery process is as follows: After concentration of the black liquor in multiple-effect evaporators to about 50% total solids, the final concentration to 60 to 65% is done by direct contact with hot flue gas from the waste heat or recovery boiler. The 65% concentration black liquor is mixed with salt cake (Na_2SO_4) before being sprayed into the furnace under pressure generated by high-pressure pumps. The furnace atmosphere is maintained with a minimum of excess air so the Na_2SO_4 is reduced to Na_2S, and sodium carbonate (Na_2CO_3) is formed in the process. These molten chemicals run out as a smelt and are dissolved in a tank to form green liquor. This liquor is then causticized with lime to form caustic soda (NaOH), with the Na_2S still present along with other residual chemicals, thus forming the regenerated cooking liquor known as white liquor. The calcium carbonate ($CaCO_3$) formed is burned in a lime kiln for reuse in causticizing. Various lime slurries and residues are formed during this process. The white liquor is then clarified and reused in the digesters, completing the cycle, as shown in Figure 1.

There are a variety of other pulping processes in use, but the sulfate process offers so many advantages that almost all recent installations have been of this type.

Bleaching Bleaching may be considered an extension of the cooking process, the object being to remove the coloring matter, carbohydrate, and lignins to that the remaining pulp

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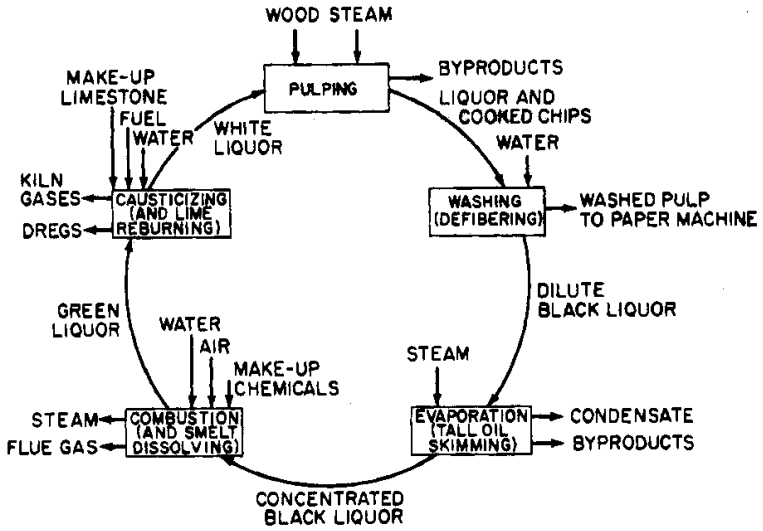


FIGURE 1 The recovery cycle in the sulfate process (Courtesy of ITT/Goulds Pumps, Inc.)

contains a maximum percentage of alpha cellulose, which is the purest cellulose form and the one most resistant to attack from normal chemicals. Bleaching is carried out to reach a degree of reflectance of monochromatic light and called pulp brightness. Because of resistance to attack, special highly reactive chemicals must be used for bleaching. The traditional chemical used for bleaching is chlorine, which is highly selective at attacking the lignin while resisting attack of the cellulose. Chlorine has a tendency, however, to form pollutants in the bleach plant effluent and is therefore being replaced with other chemicals such as chlorine dioxide, hydrogen peroxide, and gaseous oxygen and ozone. Chlorine dioxide produces highly corrosive liquors in the bleach plant.

LIQUIDS PUMPED IN A MILL

There are, broadly, three categories of liquids to be pumped in a paper mill:

1. Water and similar fluids
2. *Liquors* and *slurries*—mainly chemicals and solids in solution or suspension
3. *Stock*—suspension of cellulose fibers in water

Water Apart from the quantities involved, there are no special requirements concerning the water in pulp and paper mills because operating conditions are well within normal limits. However, iron and carbon steel piping should not be used in bleach pulp mills because of iron pickup.

Process water treatment is frequently used to purify process water for the mill and to remove undesirable elements such as iron. Higher-quality water is required for chemical reparation in the bleach plant and for boiler feedwater where demineralizer plants are used. Rubber- or epoxy-resin-lined pumps are used for those components in contact with the demineralized water.

PUMPS FOR MILL WATER For the majority of pumps, standard cast iron or stainless steel fittings are used except as noted for demineralized water. In many mills, however, stainless-steel-fitted pumps are standard because this permits a minimum number of spares to be held in stock for other duties.

In the paper mill, water used to form the sheet on the paper machine has a very low fiber content—$\frac{1}{2}$ to 1% consistency—and is known as white water. Fiber contents this low usually do not cause any pumping problems except in wear ring areas where flashing or slotting is used to keep leakage paths open and free from binding.

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Much of this water is recirculated, and where bleached products are produced, pumps must be constructed an austenitic stainless steel.

Liquors and Slurries Depending on the process and the particular point in that process, the liquor characteristics may require special pumps or special materials. Although the liquor cycle is a difficult one as far as the pumps are concerned, standard designs should be used whenever possible because this reduces the number of different types of pumps in the mill. In some cases, it may be necessary to use a higher material specification than necessary to achieve interchangeability.

Liquor and slurry pumps may be grouped as follows:

Group A—Standard designs suitable for most process uses where corrosion or erosion is not a major factor. Impellers are typically stainless steel with casings of cast iron.

Group B—Standard end-suction designs suitable for corrosive liquors. All liquid end components are typically 316 stainless steel. Duplex stainless steels may be used where erosion may be a factor.

Group C—Standard or nonstandard designs suitable for special services. Pumps are similar to group B for most applications but are of 317 or 317L stainless steel. For most corrosive services, glass-reinforced epoxy, resin, titanium, super austenitic stainless steels are used for both impeller and casing. Mechanical seals in place of packed boxes or dynamic seals are usually fitted to these pumps.

Recommendations for liquor and slurry pumps are

1. All liquor pumps should be classified as slurry type with open nonshrouded impellers of the end-suction and back pull-out type. Simply supported, double suction pumps are also used for fibrous slurries (stock)—particularly $\frac{1}{2}$% to 3% consistency stock on the paper machine. This includes most fan and cleaner pump applications.
2. On group A and B pumps, sealing is accomplished with dynamic seals, mechanical seals, or packed stuffing boxes.
3. For group C pumps, in particular, it may be necessary to depart from a standard design or type of centrifugal pump. For example, if a positive displacement characteristic is required, a screw-type pump may be used with confidence. In addition, all pumps handling stock with consistency above 6% must be regarded as nonstandard types.

After the pumps are grouped, it becomes necessary to decide which pump may be used for specific liquors. Requirements for individual mills will differ in detail, but the following may be taken as an indication of current practice, particularly in modern sulfate (kraft) mills. In every case, manufacturers should be made aware of the liquor characteristics and of the location of the pump in the process.

COOKING LIQUOR (WHITE LIQUOR—SULFATE PROCESS) This is essentially an alkaline solution made by causticizing green liquor. The liquor is prepared at concentrations over the range of 50 to 100 g/liter depending on the wood species, and the amount of active alkali (expressed as Na_2O) may be from 14 to 30% of the dry wood weight. White liquor is mainly sodium hydroxide, with a small percentage of sodium sulfide which depends on the mill sulfidity. Higher values of active chemical are used in bleached pulp mills. The term *sulfidity* is used to denote the ratio of chemicals present; it is frequently expressed as Na_2O and calculated from the expression

$$\frac{Na_2S}{NaOH + Na_2S}$$

The sulfidity value commonly used is from 20 to 30%; the higher values usually denote better chemical recovery. The specific gravity of the liquor will be approximately 1.2, and after clarification only small quantities of grit should be present. The liquor must be considered an abrasive that produces a high rate of wear on pump rotating elements. White liquor has a tendency to crystallize on internal surfaces of pipes and pumps, but there are

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no special viscosity problems and a pump head loss allowance of about 10% above that of water should be adequate. Group B pumps are recommended.

BLOW TANK DISCHARGE As the liquor introduced with the chips into the digester combines with the noncellulose and hemicellulose fractions of the wood, it changes from white liquor to black liquor before reaching the blow tank. In addition, the sudden release of pressure frees the cellulose fibers from the other matter, so the blow tank contains both raw stock (pulp) and black liquor. Pulp from the blow tank is often entrained with air, sand, and other contaminants. A stock pump, therefore, is required for this duty because the stock concentration is quite high.

BLACK LIQUOR For convenience these pumps are divided into three groups.

Weak Black Liquor (Total Solids Up to 20%) During washing, hot water is used to dissolve away the surplus organic matter from the pulp, and the liquor produced is termed *black liquor.* This liquor is a mixture of the lignins and carbohydrates in the original wood plus the cooking chemicals: it is alkaline with a solids content of 14 to 16% in a sulfate mill. The temperature will be about 180 to 190°F (82 to 87°C), and the specific gravity about 1.08. Washing is usually carried out with a minimum of three countercurrent stages, and the solids content given previously is representative of the liquor leaving the stages nearest the inlet; that is, where it is most concentrated. The quantity of recirculated liquor is quite high, and many mills have found group A pumps with stainless trim to be satisfactory and economical. With the low solids content, there are no special viscosity problems. This may be seen from Figure 2.

Black Liquor with Total Solids of 20 to 50% This liquor is formed by the evaporation of water from weak black liquor. The concentration is accomplished in multiple-effect evaporators, which usually discharge liquor with about 50% total solids at close to 200°F (93°C). In some odor-free installations, the solids concentration is much higher. Because of the nature of the evaporation, special pumps are usually required.

Liquor containing up to 50% solids is reasonably easy to pump, but allowance must be made for viscosity effects. In noting the values in Figure 2, it should be remembered that the plant must often start up cold, so cold liquor with a higher viscosity may have to be pumped. The liquor-specific gravity rises during evaporation from about 1.1 to 1.25. Group B pumps are recommended.

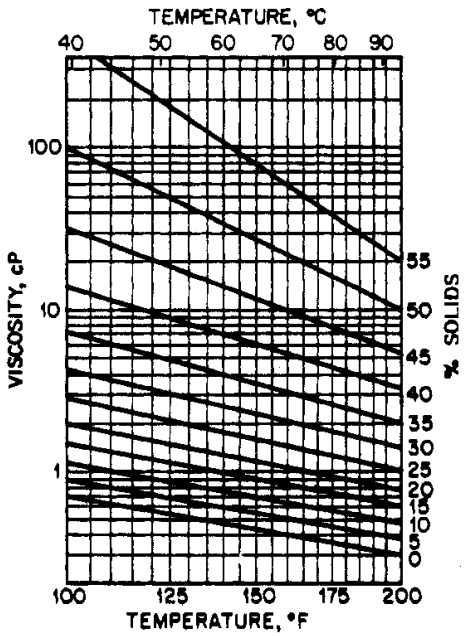


FIGURE 2 Black liquor viscosity (Courtesy of ITT/Goulds Pumps, Inc.)

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Black Liquor with Total Solids of 50 to 65% This is often referred to as heavy black liquor because the specific gravity rises to 1.35. The liquor is formed by further evaporation, either in the multiple-effect units or by contact evaporators using hot flue gas.

From a pumping standpoint, this liquor is probably the most difficult of all liquids to pump satisfactorily in pulp and paper mills. Continuous operation requires careful attention to pump sealing and maintenance. Steaming out at regular intervals of the evaporator and piping is particularly important to prevent solids buildup affecting NPSHA to the pumps.

No accurate figures are available for the viscosity of liquids with a solids concentration above 55% because there is wide variation in the liquors produced from different wood species and also in the liquors from the same wood of different ages. Hardwood species produce a more viscous liquor, especially eucalyptus, as well as more liquor per ton of pulp produced. Black liquor produced from straw pulping is even more viscous and, in addition, causes the deposition of silica on the walls of pumps and piping. An approximation of the viscosity of straw mill heavy black liquor may be determined from published figures, which give viscosities up to 2000 centistokes. This is probably at least 50% higher than liquor from normal long-fibered softwood.

During recovery, liquor is sprayed into the furnace for evaporation to dryness and burning. Prior to this, the make-up chemical (sodium sulfate or salt cake) is added and reduced to Na_2S in the reducing atmosphere of the furnace.

Little is known with certainty about heavy black liquor, but it does not seem to be very corrosive, and carbon steel is often used for pipework, although stainless steel pumps are fairly common. The pumps are subjected to severe duties—notably high heads, lumpy material, high temperature and pressure, and continuous service. Group B pumps are almost universally specified, often with casings of more wear-resistant material such as Alloy 20 or duplex stainless steels. In some cases, mills making straw pulp have not found suitable centrifugal pumps and have had to resort to gear pumps because of the very high viscosity of the liquor.

GREEN LIQUOR Green liquor is a solution of sodium carbonate and sodium sulfide plus other elements and compounds. One of these other compounds is iron sulfide in a colloidal form, which produces a greenish color. The liquor is formed by dissolving smelt from the causticizing process. Severe erosion takes place in green-liquor pumps, primarily because of the violent action inside the dissolving tank but also because of the gritty matter always present. Green liquor also builds up on the walls of pumps and piping, causing high frictional losses. The specific gravity is usually about 1.2, and an allowance of about 20% should be made for viscosity. Group B pumps are recommended for this service.

Liquor containing up to 50% solids is reasonably easy to pump, but allowance must be made for viscosity effects. In noting the values in Figure 2, it should be remembered that the plant must often start up cold, so cold liquor with a higher viscosity may have to be pumped. The liquor-specific gravity rises during evaporation from about 1.1 to 1.25. Group B pumps are recommended.

LIME SLURRIES In causticizing, various solutions and slurries are present that, apart from causing excessive wear in standard pumps, do not cause any problems. Thus, any normal slurry pump should prove satisfactory. In sulfate mills, the lime mud formed during green liquor causticizing presents the most serious problem, for approximately 1000 lb (500 kg) of mud may be formed for each ton (1000 kg) of pulp produced. Solid loads above 35% can occur, and frequent blockages are likely unless pumps are selected for minimizing downtime. For mild slurry duty, group B pumps with duplex stainless steel construction are satisfactory. For harsher applications, hard iron pumps like those used in the mining industry are employed.

BLEACH PLANT LIQUOR Most bleached pulp mills today use at least four stages of bleaching, and often six or more. Bleaching is used to remove residual lignins or to convert them to compounds that are stable regarding color and heat. The stages used include chlorination, either by hypochlorite, gaseous chlorine (both becoming obsolete), or chlorine dioxide

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(usually two stages), with an alkali extraction washing stage between. On occasions oxygen is also used. Bleach plant chemicals are usually prepared in the mill so solutions such as chlorine water, sulfuric acid, sodium chlorate, sodium chloride, sodium hydroxide, calcium hypochlorite, and chlorine dioxide all have to be pumped.

It cannot be emphasized too strongly that materials of construction are of vital importance in the chemical preparation area of the bleach plant.

In addition to the standard chemicals, some of the common pulp mill bleach substances, together with some chemical preparation systems, are as follows.

CHLORINE This is usually delivered to the mill in tank cars but is always vaporized to a gas before use.

CHLORINE WATER (HYPOCHLOROUS AND HYDROCHLORIC ACID) Concentrations cover the range from pH 2 to 10 for bleaching pulp. In some cases, the gas is mixed directly with pulp in special mixers. Group C lined pumps are essential.

SODIUM HYPOCHLORITE AND CALCIUM HYPOCHLORITE This mixture is made in the mill by permitting chlorine to react with either sodium or calcium hydroxide concentrated caustic (70%) diluted to 5 to 6% before chlorination. Calcium hypochlorite is made from a 10% solution of slaked lime at temperatures up to 150°F (66°C), but not normally exceeding 70°F (21°C). These liquors are corrosive to steel, and group C or lined pumps are necessary when handling solutions *to* the bleach plant; *after* bleaching the filtrate may still have residual hydrochloric acid.

CHLORINE DIOXIDE This is the most common bleach solution used because it gives an excellent brightness to the pulp and, despite corrosion problems, is usually cheaper than other bleach solutions.

After generation of the gas, during which absolute cleanliness is vital, the gas is stripped in a packed tower as an aqueous solution and stored in plastic tanks made of special resins that resist chemical attack. In modern plants, increasing use is made of glass-reinforced plastic with selected resins for piping, valves, and pump linings. This is sometimes a cheaper alternative than the use of exotic metals, such as titanium, for pumps. Pumps must be group C, and stainless steel is not satisfactory. Solution strengths of up to 11 g/liter are used.

SODIUM PEROXIDE AND HYDROGEN PEROXIDE These are used for bleaching groundwood pulp. Typical solutions contain sodium silicate (5%), sodium peroxide (2%), and sulfuric acid (1.5%). The latter controls the pH of the liquor. Concentrations of bleach liquors are up to 15%. Temperatures are usually less than 90°F (32°C). Group C pumps are necessary.

WASH LIQUORS In general, the filtrate from bleach washing stages will exhibit at least some of the properties of the stage immediately before washing, owing to slight excesses of chemical present. Filtrates are collected in corrosion-resistant pipes and vessels, usually made from glass-reinforced plastic, and the pumps used will be either group B or C, depending on the stage in question. The filtrate from the chlorine dioxide stages should be pumped with a super austenitic stainless steel case and trim pump because the filtrate is not as corrosive as the bleach solution.

Spent acid from chemical preparation plants is also highly corrosive, and usually stainless steel is not satisfactory for use with it.

Effluent from the bleach plant, on the other hand, is usually a mixture of several liquors, and experience has shown that 317 stainless steel is a suitable material for pumps that handle it.

CHLORINE DIOXIDE PREPARATION; SODIUM CHLORATE Chlorine dioxide is produced by permitting sodium chlorate to react with sulfuric acid and hydrochloric acid in a vessel into

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which a reducing agent such as NaCl, SO_2, or methanol is introduced in controlled quantities. Sodium chlorate solutions are usually from 43 to 46%, at which strength the specific gravity is about 1.38. Stainless steel pumps may be used, but epoxy-resin-lined pumps are superior.

FOUL CONDENSATE This arises from the evaporation of water from black liquor at the multiple-effect evaporators, as these units flash vapor from the liquor in one stage and use this to evaporate the liquid in the next stage. The vapor when condensed contains some carry-over from the black liquor, and thus the condensate is contaminated and corrosive. When a nickel cast iron casing and stainless trim are used, group A pumps should be satisfactory. Some liquors produce very corrosive vapors, and a stainless casing pump may prove necessary. Group B pumps are recommended.

Stock Stock is the term applied to the suspension of cellulose fiber in water. It first appears either after grinding (in the case of mechanical pulp) or after the blow tank (in the case of chemical pulp). Stock production rates may be converted to pump flow rates with the following formula:

$$\text{Flow (USGPM)} = \frac{\text{Production (air dried short tons per day)} \times 1.5}{\text{Consistency (\% oven dried)}}$$

After the separation of chemicals or impurities by washing and screening, the stock is given a mechanical treatment known as either beating or refining, depending on the nature of the treatment. This enhances the sheet properties. Additives such as starch, clay fillers, alum, and size are introduced to impart special characteristics, depending on the end use of the product.

Over the range of stock in normal use, the specific gravity may be considered constant for all practical purposes, with a value equal to that of water at the appropriate temperature.

Cellulose fibers have a specific gravity slightly greater than water, and constant agitation is required to ensure that stratification does not occur in storage. Agitation, however, can also introduce air, to the detriment of the stock.

The pH of stock varies over a wide range—from as low as 1.0 during some bleaching processes to 11 with others. In the paper machine room, the pH of the stock will usually range from 4 to 8. Thus from a corrosion viewpoint washed stock does not usually present special problems except when high-grade bleached products are produced. Stains will be caused by iron sulfides or oxides, and therefore stainless steel must be used—frequently 304 for washed stock, but 316 or 317 within the bleach plant before washing or where bleach liquor is likely to be present with the stock.

Unbleached paper mills generally do not experience corrosion with washed stock, except in the case of groundwood mills, where the pH is usually lower than in chemical pulp mills.

FIBER CHARACTERISTICS Stock made from softwoods will have a predominance of fibers 2.8 to 3.5 mm long and 0.25 to 0.3 mm wide; fibers from hardwoods will be about 1.0 to 1.3 mm long and 0.1 mm wide. Straw fiber will be still shorter—0.75 mm on the average—but flax can have fibers up to 9.0 mm long. These figures are typical and are of interest because of their effect on pump performance.

CONSISTENCY This is the amount of dry fiber content in the stock, expressed as a percentage by mass. Typical values will vary from about 0.1% for the feed to the headbox of a special paper machine to 16% for stock between some bleaching stages or in high-density towers. The critical stock consistency in the selection of pumps is 6%. Up to the 6% level, pumps may be selected on the basis of their water performance.

FREENESS When stock is beaten, or refined, it acquires an affinity for water, and the longer the stock is beaten, the longer the water retention period. The retention of water by the stock increases the friction factor of the flow of stock.

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Freeness is often measured by an instrument called the Canadian standard freeness tester. The range of values covers a scale from 0 to 900, with a higher freeness value indicating a less refined stock and thus a lesser affinity for water. This instrument measures the amount of water drained from a sample of stock under a regularly decreasing head. Its use is recommended by the Technical Association of the Pulp and Paper Industry (TAPPI), and it is commonly employed in North American mills.

STOCK PUMPS In stock pumps, consistency is not a major problem until a value of about 6% is reached. The essential requirement is to get the stock to the pump impeller, and every effort should be made to keep the piping as large and straight as possible. A typical open impeller, end suction stock pump is shown in Figure 3.

Above 6% consistency, special pumps are required, and they can be of the positive displacement screw type or centrifugal type. Air entrainment in the stock will reduce pump output. Air entrainment occurs from agitation in the chests, from flow over weirs, and from flow through restricted openings. How air entrained in water and in stock affects pump performance is shown in Figure 4.

PIPING ARRANGEMENT Piping should be as straight and short as possible. This is particularly important on the suction side of the pump to prevent dewatering of the stock. The diameter of the suction piping should be at least one pipe size larger than the diameter of the pump suction and should project into the stock chest. The inlet end of the suction pipe should be cut at an angle, and the bottom of the pipe should be at least $1\frac{1}{2}$ pipe diameters from the bottom of the chest. With the long side of the pipe on top, the probability of drawing air into the suction of the pump through vortices is reduced. Some manufacturers provide a lump breaker or screw feeder at the suction side of the pump for pumping stock above from 6% to 8% consistency.

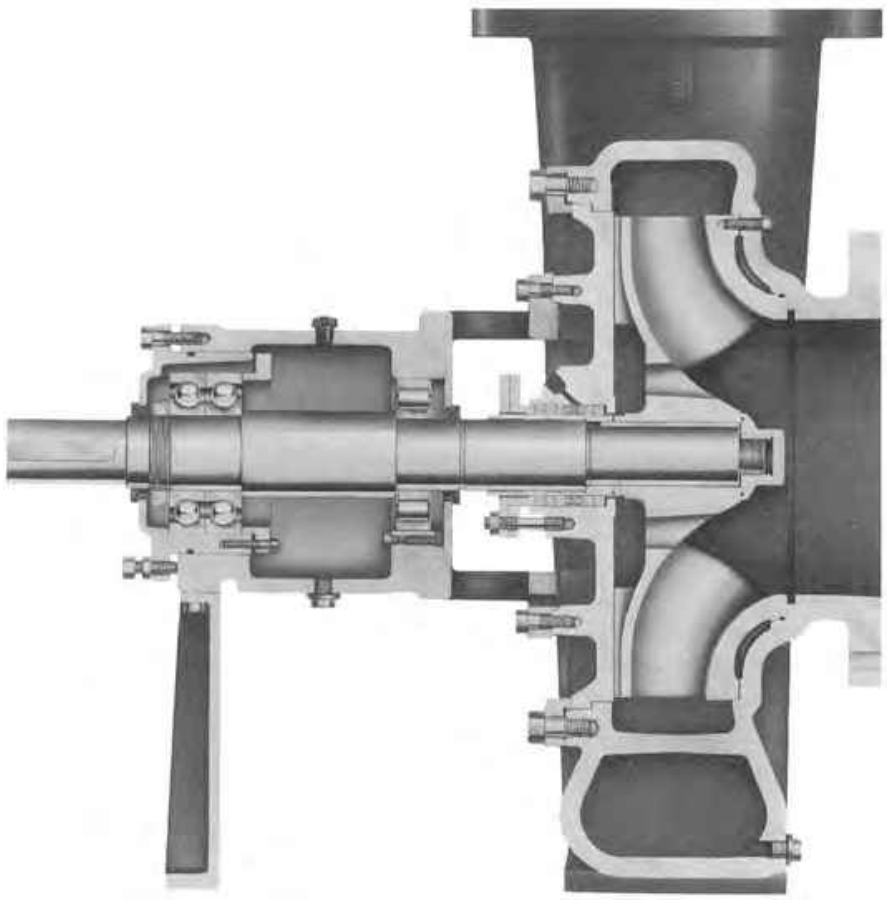


FIGURE 3 End suction stock pump (Courtesy of ITT / Goulds Pumps, Inc.)

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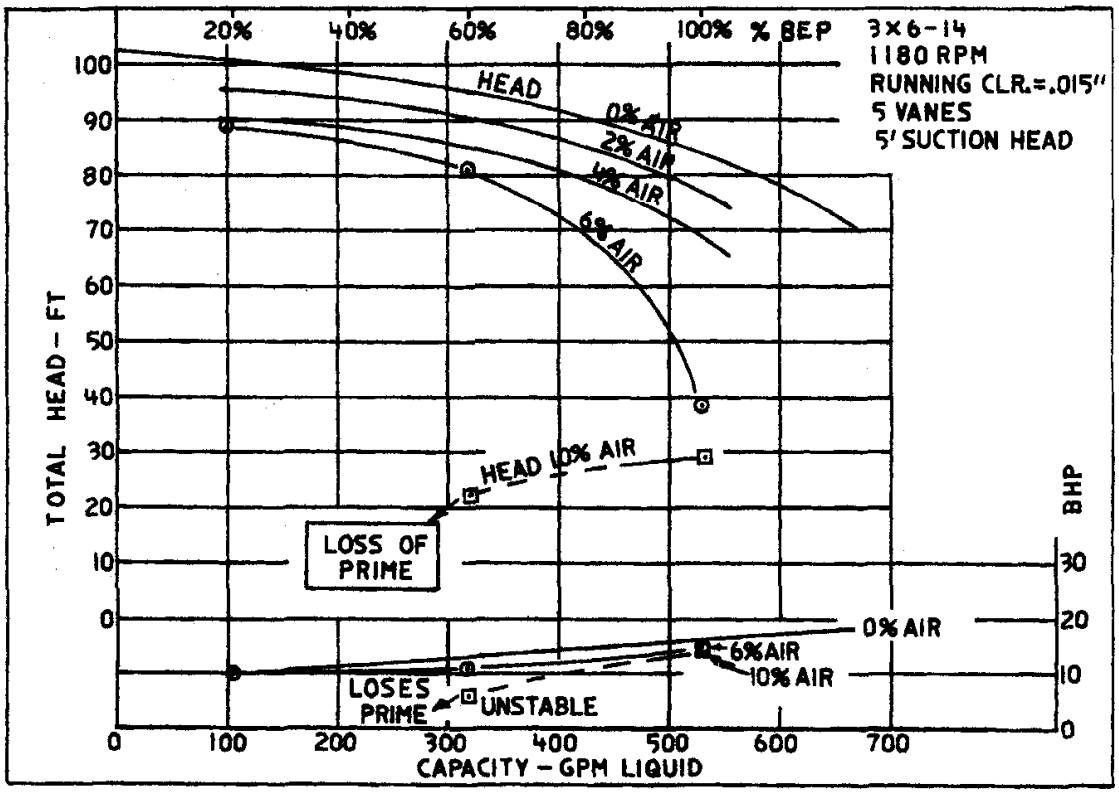


FIGURE 4 General effect of entrained gas on pump performance (Courtesy of ITT/Goulds Pumps, Inc.)

SIZE OF PUMPS It is important to estimate the performance requirements of stock pumps as accurately as possible. Oversizing of centrifugal pumps will cause an unbalanced radial thrust on the impeller resulting in excessive shaft deflection and reduced bearing and seal life. Oversizing can also result in impeller recirculation and the accompanying cavitation-like noise and damage to pump components.

PUMP TYPES AND SERVICES

The normal economic considerations of any continuous process apply equally well to pulp and paper mills, with a few points of difference. In pump installations, the improved cost figures that are possible from larger units are limited to some extent by the manufacturer's standard size units. Because the industry is capital intensive, the overriding factor in any pump installation is reliability. To achieve this, it does not necessarily follow that it is better to have two pumps installed, with one as a standby. One properly designed and serviced unit may well be better than two unknown units; this is especially true where the pump is in a portion of the process that cannot be interrupted without serious losses, either in raw materials or in quality of the finished product.

A duplication of pumps means complications in extra valves, pipework, connections for steam and viscous liquids, electric motors, cables, and starters. The result is that, in modern mills with good machinery and materials of construction, there is a strong tendency away from the duplication of pumps because of increased costs and questionable reliability. It follows that the important thing is to select the right pump and the right duty point in a particular range.

There may be several hundred pumps in a modern pulp and paper mill, but the cost of these pumps is probably less than 5% of the total equipment cost. It is unwise, therefore, to jeopardize mill reliability by compromising pump quality. Corrosion and erosion are major factors in pump life, and even with the best materials, the life of some components

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in severe service may be 12 months or less. Moreover, the power used by pumps is usually less than one-third of the mill demand. If one remembers that the cost of total power absorbed in a mill is only around 4%, even a 50% reduction in pump power will still be less than 1% net.

Efficiency The best point at which to operate a pump is, of course, its maximum efficiency, but this is not always possible, particularly in the case of stock pumps because of the wide range of process variables in most pulp and paper mills. Open impellers and excess clearances also reduce the efficiency, yet these factors are much more important in stock pumps than efficiency. Another important consideration is speed. Stock pumps should be chosen to run at as low a speed as possible to achieve stable operation, and this speed may not produce an efficient pump. The shape of the pump performance curve is much more important than the best efficiency quoted by a manufacturer. A flat or unstable head curve may produce surging or instability in the pump output. Good pump selection, therefore, must emphasize reliability as the first consideration and efficiency and costs as secondary considerations.

Pump Speeds Most of the pump duties in a mill can be accomplished by single-stage pumps and four-pole motor speeds. For liquids other than water, two-pole motor speeds should be avoided if possible. For special duties, including stock pumping, six- or eight-pole motors may be required unless some indirect or variable speed is used. Although it is true that lower speeds mean larger pumps and more expensive electric motors, lower speeds are justified because of reduced maintenance and greater reliability. Some deviation from these speeds may be necessary for pumps generating heads in excess of 150 ft (46 m), but this can often be taken care of by a larger impeller rather than a higher shaft speed.

Although not an option in every circumstance, the use of variable speed drive systems for pumps to replace control valves should also be considered. Reliable alternating current variable frequency drives are now readily available up to 2300 volts and are easily justifiable based on normal power costs at the 440-volt level. The commensurate speed reduction seen by the pump when replacing a control valve and avoiding its necessary pressure drop will also add to the reliability and life of the pump.

Multistage Pumps Except with boiler feedwater, the use of multistage pumps should be avoided. This is particularly true for stock and viscous liquors. The complicated pump design makes such units unacceptable for these services.

Pipeline Systems With black liquor, green liquor, and similar high-viscosity liquors, adequate provision must be made for steaming out and subsequent liquor drainage. The pumps must also be included in this system. Although the liquor pumps should be designed to pass some solid matter, motorized strainers should be used on the pumps for cyclone evaporators and recovery boiler-fuel pumps because both pumps discharge to spray nozzles. Dead pockets and other areas where liquor can collect should be avoided because solids from the liquor will build up in these areas and possibly break away to block pipelines or pump impellers. For protection at shutdowns, even for short periods, steaming out is essential.

Positive Displacement Pumps The principal use of the positive displacement pump is for consistency control of stocks above 5%. The normal measuring device used is quite satisfactory at low consistencies but is less reliable at the higher values. More satisfactory control may be achieved by using a screw pump, where the power is proportional to the pulp consistency at constant flow. Such pumps have been very reliable on consistency control.

Digester Circulating Pumps Digester circulating pumps are used with indirectly heated batch digesters to circulate the liquor at the digester pressure and temperature. Maintenance problems are common on these pumps because heads can be as high as 150 lb/in^2 (1034 kPa) and temperatures as high as 350°F (177°C). In addition, the circulating liquor contains some raw pulp even though screens are fitted to the digester outlets.

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Pumps for this service, therefore, should be centerline mounted, of very heavy construction, have closed impellers, and mechanical seals. In addition, there is often considerable pipework involved, for a digester may easily be 60 ft (18 m) high and pipe loads are often imposed on the pumps. Expansion joints and long-radius bends are used, but it is desirable to support the pumps on springs or slide bases.

Pumps for Heavy Black Liquor above 60% Solids A typical pump for this service is shown in Figure 11. An open impeller in a 316 stainless steel casing is recommended. A heavy sleeved shaft of 316 stainless steel with ample clearance between the rotating parts is also required for satisfactory operation. These pumps may be required to handle black liquor up to 2000 centistokes viscosity and should be provided with water cooling. Steam jacketing is not always satisfactory because the liquor may tend to bake on the walls of the casing. Pump speeds should be below 1800 rpm, if possible.

Multiple-Effect Evaporator Pumps Pumps in this service often operate in cavitation owing to problems in regulating the flow between evaporation stages. Level control valves in the suction line to the pump can alleviate this problem, but cavitation can still be expected in the pump. A self-priming pump may give longer life of the rotating elements than the condensate pump usually used on this service.

Diaphragm Pumps Diaphragm pumps are used in pumping lime mud slurries of high concentration. They consist of a rubber or neoprene diaphragm with a pulsating air supply on one side, controlled by a timer, and the slurry on the other.

Medium Consistency Stock Pumps Pumping of medium consistency stock (from about 8% to about 16% consistency) can be accomplished with positive displacement screw pumps or centrifugal pumps that have been specially adapted. Medium consistency stock is a thick viscoelastic material made up of strong networks of fibers called flocs. A high shear rate is necessary to render the stock capable of flow. Therefore all centrifugal medium consistency pumps are equipped with an inducer or feeder vane device in front of the pump suction to provide the shear necessary for the stock to flow into the impeller eye. There are also large amounts of entrained air in medium consistency stock and most centrifugal medium consistency pumps are equipped as well with an auxiliary vacuum pump to remove the air allowing for stable pump operation and high efficiencies. An example of a medium consistency centrifugal pump is shown in Figure 5.

Fan Pump Another specialized pump application in the paper mill is the pump that feeds the headbox or nozzle that spreads white paper onto the moving wire sheet that forms the paper. The *fan pump* is usually a large horizontally split double suction pump in austenitic (AISI 316) trim or all austenitic (AISI 316) stainless steel construction. To form an evenly distributed sheet, the headbox must be provided with flow that is free of pressure pulsations and flow disturbances. The fan pump rotating element (impeller, shaft, sleeves, and so on) is designed and built to special tolerances to guarantee a typical maximum pressure pulse of 0.5% of pump total dynamic head, peak-to-peak, at any frequency. A typical fan pump is shown in Figure 6.

Shower Services Shower services require pumps to operate at low flows and high pressures. Operation at low flows requires special pump designs to ensure good pump reliability. These pumps are usually of single-stage design with special, circular volute construction to minimize radial loads on the rotor and bearings. A typical low flow pump is shown in Figure 7. Some high pressure shower services require multistage or high speed (greater than 3600 rpm) pumps to reach the desired pressure. Generally, multistage and high speed pumps require more maintenance than single-stage units because of mechanical complexity.

Solid Handling Services There are many pump applications that require handling of various size solids, particularly in recycle services. Recycle services may contain tramp metal and plastics that can clog stock pumps. Recessed impeller pumps, sometimes

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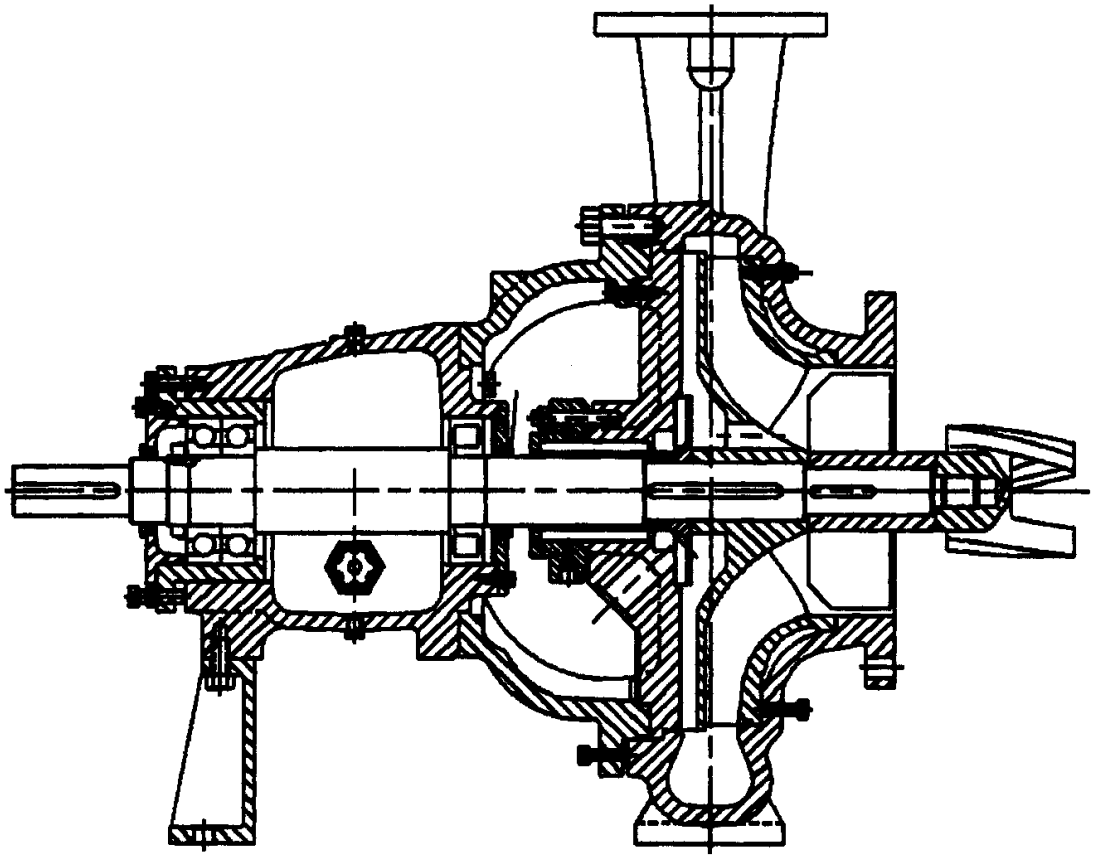


FIGURE 5 Medium consistency stock pump (Courtesy of ITT/Goulds Pumps, Inc.)

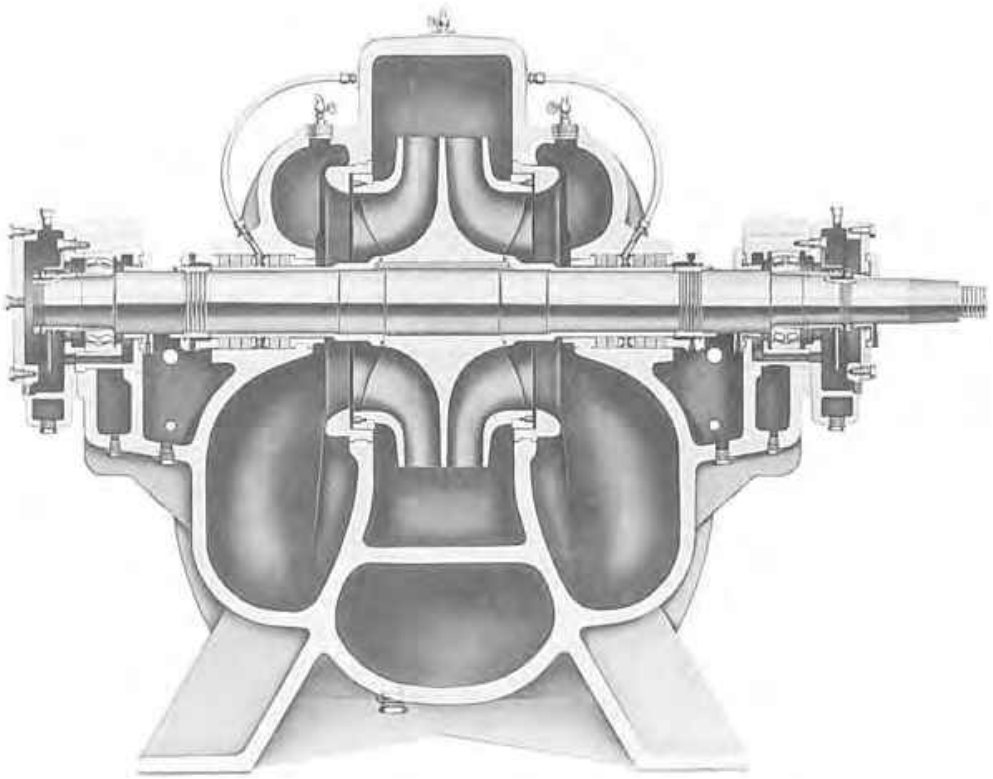


FIGURE 6 Fan pump (Courtesy of ITT/Goulds Pumps, Inc.)

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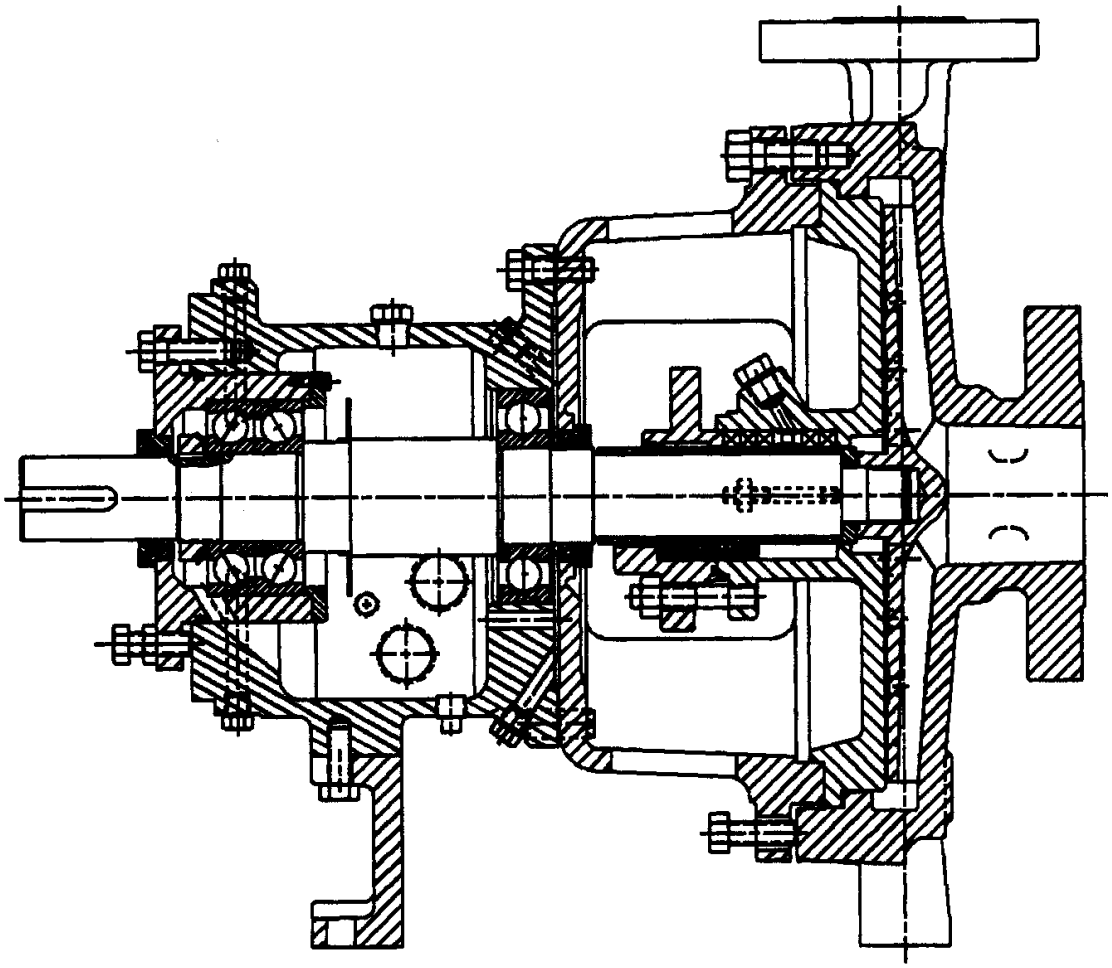


FIGURE 7 A typical low flow pump used for shower services (Courtesy of ITT/Goulds Pumps, Inc.)

referred to as vortex pumps, are particularly useful for these types of services because the impeller is recessed from the main flow allowing large solids to pass through the pump.

Sealless Pump Services In recent years, some of the severely corrosive chemical applications have been served well by sealless pumps. Magnetic drive pumps are particularly useful for tank car unloading or sodium hydroxide and other chemicals where mechanical seals have required high levels of maintenance (See Section 2.2.3).

Vacuum Pumps These are used to extract water from the sheet on the fourdrinier wire and at the suction presses by means of a vacuum up to a maximum of about 25 inHg (635 mmHg). Approximately 40,000 lb (18,100 kg) of water is extracted by this means for every ton (1000 kg) of paper produced, and this water is removed by entrainment with the air handled by the vacuum pumps. Frequently water separators are used to remove water; their use is a matter of economics, as a reduction of up to 10% in power may be achieved.

Vacuum pumps are of three basic types:

1. Water ring
2. Positive displacement
3. Centrifugal or axial-flow

Many engineers prefer the water-ring type, probably because of its simplicity. In general, however, this type uses more power, mainly because of the heating of the circulating water, which is then discharged to a drain.

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Centrifugal and axial-flow machines must be provided with water separators, but they are more efficient overall, particularly when the heat of compression is used in the machine room ventilating system. The machines run at high speed and are usually driven by a steam turbine.

The paper machine system requires vacuums at different levels, from a few inches (millimeters) of mercury to the maximum possible. Often pumps are connected to a common header, and orifice plates are used to divide the flow to ensure some measure of standby capacity. This involves throttling, however, and may create flow problems unless quantities are carefully estimated. The axial-flow machine permits extraction at any point along the rotor within fairly close limits and requires an accurate estimation of the quantities and specific vacuum required. A standby for axial-flow machines cannot usually be justified.

Stock and Liquor Pump Standardization Throughout the mill the duties of many pumps are similar, but different materials of construction may be used. If, at a slightly extra initial cost, the rotating elements of the pump can be standardized, this will reduce spare-parts inventories. This is also an advantage when purchasing pumps for a new mill. For example, if a standard arrangement consists of a complete rotating element, including bearings, only the impeller size and material need be different. Standardization is an additional reason for recommending that all pumps be stainless-fitted. Obviously, large pumps need individual evaluation. Standardization of stock pumps is less feasible, but up to about 6% stock, similar pumps can usually be specified.

Pump Selection Guidelines In many cases, an excess margin on head and capacity is specified. Where margins are excessive, mechanical troubles and cavitation often occur. The following guidelines will help to properly size a pump and motor in order to avoid these problems:

1. Carefully calculate the pump duty, using the TAPPI technical information sheet (TIS 410-14) for stock friction and the proper viscosity corrections for liquor friction. Always use a schematic of the actual piping system to be installed.
2. Properly size control valves to minimize the required pressure drop. Consider the use of variable speed drives.
3. Make the proper corrections to head and power for the presence of entrained gas in the fluid being pumped, per pump manufacturers recommendations.
4. Select a pump from the manufacturer's curve and note the impeller diameter and range for the pump. If the duty point falls near the end of the curve or if the impeller diameter is greater than 95% of the maximum impeller diameter, it is advisable to select the next size larger pump. For maximum reliability, select an impeller diameter equal to or nearly equal to: {(maximum impeller diameter—minimum impeller diameter) × 0.75} + minimum impeller diameter.
5. Allow 10% or 3 ft (whichever is larger) between NPSHA and NPSHR. Systems should be designed for a maximum available S value (suction specific speed) of 8,000.
6. Motors should be sized to be non-overloading for 105% of the impeller diameter chosen. Baseplates should allow for the installation of the next larger frame size.

SPECIAL PULP AND PAPER CONSIDERATIONS

Friction Loss of Pulp Suspensions In Pipe In any stock piping system, the pump provides flow and develops hydraulic pressure (head) to overcome the differential in head between two points. This total head differential consists of pressure head, static head, velocity head, and total friction head produced by friction between the pulp suspension and

the pipe, bends, and fittings. The total friction head is the most difficult to determine because of the complex, nonlinear nature of the friction loss curve. This curve can be affected by many factors.

The following analytical method for determining pipe friction loss is based on the TAPPI Technical Information Sheet (TIS) 410-14 (Reference 1), and is applicable to stock consistencies (oven-dried) from 2 to 6%. Normally, stock consistencies of less than 2% (oven-dried) are considered to have the same friction loss characteristic as water.

The friction loss of pulp suspensions in pipe, as presented here, is intended to supersede the various methods previously issued.

Figure 8 and Figure 9 show typical friction loss curves for two different consistencies ($C_2 > C_1$) of chemical pulp and mechanical pulp, respectively.

The friction loss curve for chemical pulp can be conveniently divided into three regions, as illustrated by the shaded areas of Figure 10.

These regions may be described as follows:

Region 1 (Curve AB) is a linear region where friction loss for a given pulp is a function of consistency, velocity, and pipe diameter. The velocity of the upper limit of this linear region (Point B) is designed V_{max}.

Region 2 (Curve BCD) shows an initial decrease in friction loss (to Point C) after which the friction loss again increases. The intersection of the pulp friction loss curve and the water friction loss curve (Point D) is termed the onset of drag reduction. The velocity at this point is designated V_W.

Region 3 (Curve DE) shows the friction loss curve for pulp fiber suspensions below the water curve. This is due to a phenomenon called drag reduction. Reference 2 describes the mechanisms which occur in this region.

Regions 2 and 3 are separated by the friction loss curve for water, which is a straight line with a slope approximately equal to 2.

The friction loss curve for mechanical pulp, as illustrated in Figure 11, is divided into only two regions:

Regions 1 and 3. For this pulp type, the friction loss curve crosses the water curve at V_W and there is no true V_{max}.

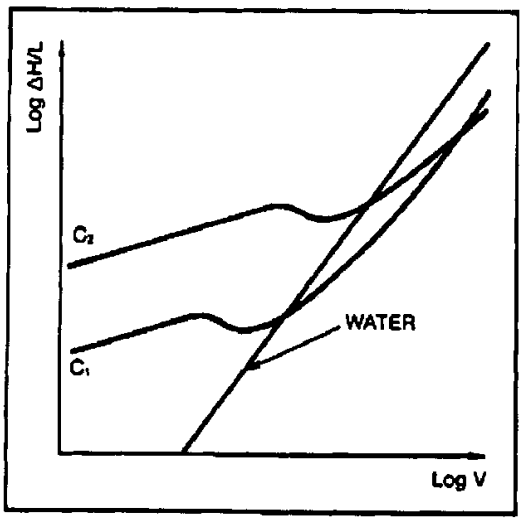


FIGURE 8 Friction loss curves for chemical pulp (Courtesy of ITT/Goulds Pumps, Inc.)

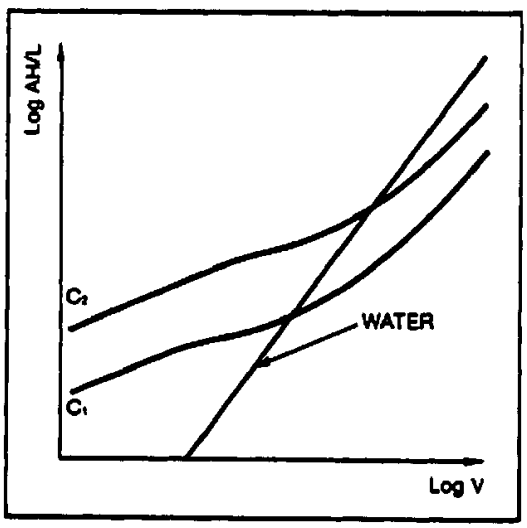


FIGURE 9 Friction loss curves for mechanical pulp (Courtesy of ITT/Goulds Pumps, Inc.)

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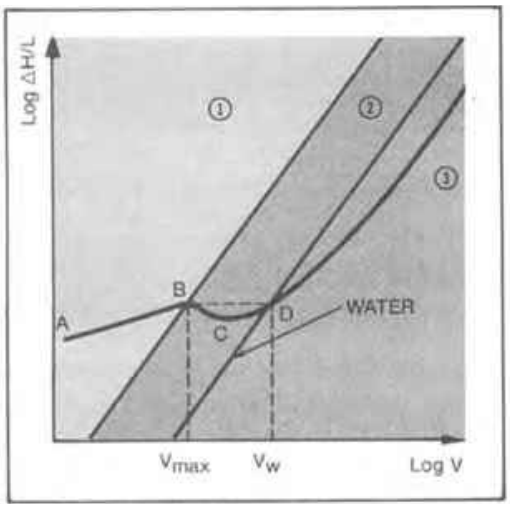


FIGURE 10 Friction loss curves for chemical pulp, shaded to show individual regions (Courtesy of ITT/Goulds Pumps, Inc.)

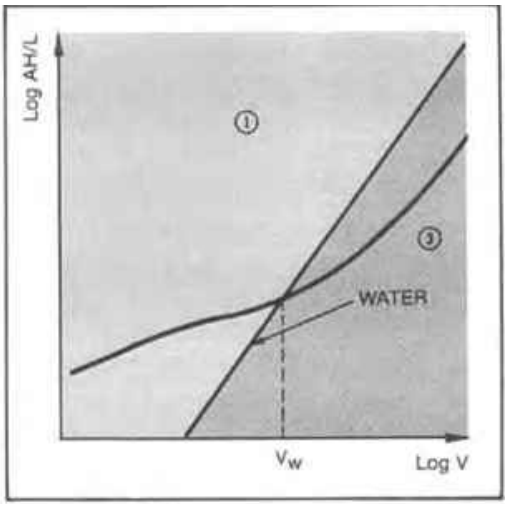


FIGURE 11 Friction loss curves for mechanical pulp, shaded to show individual regions (Courtesy of ITT/Goulds Pumps, Inc.)

To determine the pipe friction loss component for a specified design basis (usually daily mass flow rate), the following parameters must be defined:

a) **Pulp Type** Chemical or mechanical pulp, long or short fibered, never dried or dried and reslurried, and so on. This is required to choose the proper coefficients which define the pulp friction curve.

b) **Consistency, *C* (oven-dried)** Often a design constraint in an existing system. *Note*: If air-dried consistency is known, multiply by 0.9 to convert to oven-dried consistency.

c) **Internal pipe diameter, *D*** Lowering D reduces initial capital investment, but increases pump operating costs. Once the pipe diameter is selected, it fixes the velocity for a prespecified mass flow rate.

d) **Bulk velocity, *V*** Usually based on a prespecified daily mass flow rate. Note that both V and D are interdependent for a constant mass flow rate.

e) **Stock temperature, *T*** Required to adjust for the effect of changes in viscosity of water (the suspending medium) on pipe friction loss

f) **Freeness** Used to indicate the degree of refining or to define the pulp for comparison purposes

g) **Pipe material** Important to specify design correlations and compare design values

The bulk velocity (V) will depend on the daily mass flow rate and the pipe diameter (D) selected. The final value of V can be optimized to give the lowest capital investment and operating cost with due consideration of future demands or possible system expansion.

The bulk velocity will fall into one of the regions previously discussed. Once it has been determined in which region the design velocity will occur, the appropriate correlations for determining pipe friction loss value(s) may be selected. The following describes the procedure to be used for estimating pipe friction loss in each of the regions.

Region 1 The upper limit of Region 1 in Figure 10 (Point B) is designated V_{max}. The value of V_{max} is determined using Eq. 1 and data given in Table I or IA, Appendix B.

$$V_{max} = K'C^{\sigma}(\text{ft/s}), \tag{1}$$

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where K = numerical coefficient (constant for a given pulp is attained from Table 1 or IA)
C = consistency (oven-dried, expressed as a percentage, not decimally)
σ = exponent (constant for a given pulp), obtained from Table I or IA

If the proposed design velocity (V) is less than V_{max}, the value of flow resistance ($\Delta H/L$) may be calculated using Eq. 2 and data given in Table II or IIA, and the appendices.

$$H/L = FKV^{\alpha}C^{\beta}D^{y} \text{ (ft/100 ft)} \quad (2)$$

where F = factor to correct for temperature, pipe roughness, pulp type, freeness, or safety factor (refer to Appendix D)
K = numerical coefficient (constant for a given pulp), obtained from Table II or IIA
V = bulk velocity (ft/s)
C = consistency (ven-dried, expressed as a percentage, not decimally)
D = pipe inside diameter (in)
α, β, y = exponents (constant for a given pulp), obtained from Table II or IIA

For mechanical pumps, there is no true V_{max}. The upper limit of the correlation equation (Eq. 2) is also given by Eq. 1. In this case, the upper velocity is actually V_W.

Region 2 The lower limit of Region 2 in Figure 10 (Point B) is V_{max} and the upper limit (Point D) is V_W. The velocity of the stock at the onset of drag reduction is determined using Eq. 3.

$$V_W = 4.00C^{1.40} \text{ (ft/s)} \quad (3)$$

where C = consistency (oven-dried, expressed as a percentage, not decimally).

If V is between V_{max} and V_W, Eq. 2 may be used to determine $\Delta H/L$ at the maximum point (V_{max}). Because the system must cope with the worst flow condition, $\Delta H/L$ at the maximum point (V_{max}) can be used for all design velocities between V_{max} and V_W.

Region 3 A conservative estimate of friction loss is obtained by using the water curve. $(\Delta H/L)_W$ can be obtained from a Friction Factor vs. Reynolds Number plot (for example, Hydraulic Institute ANSI/HI 2000 Edition Pump Standards, Reference 3), or approximated from the following equation (based on the Blasius equation.)

$$(\Delta H/L)_W = 0.58V^{1.75}D^{-1.25} \text{ (f/100 ft)} \quad (4)$$

where V = bulk velocity (ft/s)
D = pipe diameter (in)

Previously published methods for calculating pipe friction loss of pulp suspensions gave a very conservative estimate of head loss. The method just described gives a more accurate estimate of head loss due to friction, and has been used successfully in systems in North America and world-wide.

Please refer to Appendix A for equivalent equations for use with metric (SI) units. Tables I and IA are located in Appendix B; Tables II and IIA are located in Appendix C. Pertinent equations, in addition to those herein presented, are located in Appendix D. Example problems are located in Appendix E.

The friction head loss of pulp suspensions in bands and fittings may be determined from the basic equation for head loss, Eq. 5.

$$H = KV_1^2/2g \text{ (ft)} \quad (5)$$

where K = loss coefficient for a given fitting
V_1 = inlet velocity (ft/s)
g = acceleration due to gravity (32.2 ft/s²)

Values of K for the flow of water through various types of bends and fittings are tabulated in numerous reference sources (for example, Hydraulic Institute ANSI/HI 2000 Edition Pump Standards, Reference 3). The loss coefficient for valves may be obtained from the valve manufacturer. See Section 11.1.

The loss coefficient for pulp suspensions in a given bend or fitting generally exceeds the loss coefficient for water in the same bend or fitting. As an approximate rule, the loss coefficient (K) increases 20% for each 1% increase in oven-dried stock consistency. Please note that this is an approximation; actual values of K may differ, depending on the type of bend or fitting under consideration (4).

Appendix A When metric (S/I) units are utilized, the following replace the corresponding equations in the main text.

$$V_{max} = K'C^{\sigma} \text{ (m/s)} \tag{1M}$$

where K = numerical coefficient (constant for a given pulp), obtained from Table I or IA
C = consistency (oven-dried, expressed as a percentage, not decimally)
σ = exponent (constant for a given pulp), obtained from Table I or IA

$$\Delta H/L = FKV^{\alpha}C^{\beta}D^{\gamma} \text{ (m/100 m)} \tag{2M}$$

where F = factor to correct for temperature, pipe roughness, pulp type, freeness, or safety factor (refer to Appendix D)
K = numerical coefficient (constant for a given pulp), obtained from Table II or IIA
V = bulk velocity (m/s)
C = consistency (oven-dried, expressed as a percentage, not decimally)
D = pipe inside diameter (mm)
α, β, γ = exponents (constant for a given pulp), obtained from Table II or IIA

$$V_W = 1.22C^{1.40} \text{ (m/s)} \tag{3M}$$

where C = consistency (oven-dried, expressed as a percentage, not decimally).

$$(\Delta H/L)_w = 264V^{1.75}D^{-1.25} \text{ (m/100m)} \tag{4M}$$

where V = bulk velocity (m/s)
D = pipe inside diameter (mm)

$$H - KV_1^{\,2}/2g \text{ (m)} \tag{5M}$$

where K = loss coefficient for a given fitting
V_1 = inlet velocity (m/s)
g = acceleration due to gravity (9.81 m/s²)

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Appendix B

TABLE I Data for use with Eq. 1 or Eq. 1M to determine velocity limit, V_{max}[1]

Pulp Type	Pipe Material	K'	σ
Unbeaten aspen sulfite never dried	Stainless Steel	0.85 (0.26)	1.6
Long fibered kraft never dried CSF = 725[6]	PVC	0.98 (0.3)	1.85
	Stainless Steel	0.89 (0.27)	1.5
Long fibered kraft never dried CSF = 650[6]	PVC	0.85 (0.26)	1.9
Long fibered kraft never dried CSF = 550[6]	PVC	0.75 (0.23)	1.65
Long fibered kraft never dried CSF = 260[6]	PVC	0.75 (0.23)	1.8
Bleached kraft never dried and reslurried[6]	PVC	0.79 (0.24)	1.5
	Stainless Steel	0.59 (0.18)	1.45
Long fibered kraft never dried and reslurried[6]	PVC	0.49 (0.15)	1.8
Kraft birch dried and reslurried[6]	PVC	0.69 (0.21)	1.3
Stone groundwood CSF = 114	PVC	4.0 (1.22)	1.40
Refiner groundwood CSF = 150	PVC	4.0 (1.22)	1.40
Newsprint broke CSF = 75	PVC	4.0 (1.22)	1.40
Refiner groundwood (hardboard)	PVC	4.0 (1.22)	1.40
Refiner groundwood (insulating board)	PVC	4.0 (1.22)	1.40
Hardwood NSSC CSF = 620	PVC	0.59 (0.18)	1.8

Notes:

1. When metric (SI) units are utilized, use the value of K' given in parentheses. When the metric values are used, diameter (D) must be in millimeters (mm) and velocity (V) in meters per second (m/s).
2. Original data obtained in stainless steel and PVC pipe. PVC is taken to be hydraulically smooth pipe.
3. Stainless steel may be hydraulically smooth although some manufacturing processes may destroy the surface and hydraulic smoothness is lost.
4. For cast iron and galvanized pipe, the K' values will be reduced. No systematic data are available for the effects of surface roughness.
5. If pulps are not identical to those shown, some engineering judgment is required.
6. Wood is New Zealand Kraft pulp.

TABLE IA Data (5, 6) for use with Eq. 1 or Eq. 1M to determine velocity limit, V_{max}

Pulp Type[5]	Pipe Material	K'	σ
Unbleached sulphite	Copper	0.98 (0.3)	1.2
Bleached sulphite	Copper	0.98 (0.3)	1.2
Kraft	Copper	0.98 (0.3)	1.2
Bleached straw	Copper	0.98 (0.3)	1.2
Unbleached straw	Copper	0.98 (0.3)	1.2

Estimates for othe pulps based on published literature.

Pulp Type[5, 6]	Pipe Material	K'	σ
Cooked groundwood	Copper	0.75 (0.23)	1.8
Soda	Steel	4.0 (1.22)	1.4

Note: When metric (SI) units are utilized, use the value of K' given in parentheses. When the metric values are used, diameter (D) must be in millimeters (mm) and velocity (V) in meters per second (m/s).

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Appendix C

TABLE II Data for use with Eq. 2 or Eq. 2M to determine head loss, $\Delta H/L$[1]

Pulp Type	K	α	β	y
Unbeaten aspen sulfite never dried	5.30 (235)	0.36	2.14	−1.04
Long fibered kraft never dried CSF = 725[5]	11.80 (1301)	0.31	1.81	−1.34
Long fibered kraft never dried CSF = 650[5]	11.30 (1246)	0.31	1.81	−1.34
Long fibered kraft never dried CSF = 550[5]	12.10 (1334)	0.31	1.81	−1.34
Long fibered kraft never dried CSF = 260[5]	17.00 (1874)	0.31	1.81	−1.34
Bleached kraft bleached and reslurried[5]	8.80 (970)	0.31	1.81	−1.34
Long fibered kraft dried and reslurried[5]	9.40 (1036)	0.31	1.81	−1.34
Kraft birch dried and reslurred[5]	5.20 (236)	0.27	1.78	−1.08
Stone groundwood CSF = 114	3.81 (82)	0.27	2.37	−0.85
Refiner groundwood CSF = 150	3.40 (143)	0.18	2.34	−1.09
Newspaper broke CSF = 75	5.19 (113)	0.36	1.91	−0.82
Refiner groundwood CSF (hardboard)	2.30 (196)	0.23	2.21	−1.29
Refiner groundwood CSF (insulating board)	1.40 (87)	0.32	2.19	−1.16
Hardwood NSSF CSF = 620	4.56 (369)	0.43	2.31	−1.20

Notes:

1. When metric (SI) units are utilized, use the value of K' given in parentheses. When the metric values are used, diameter (D) must be in millimeters (mm) and velocity (V) in meters per second (m/s).
2. Original data obtained in stainless steel and PVC pipe (7, 8, 9).
3. No safety factors are included in the above correlations.
4. The friction loss depends considerably on the condition of the inside of the pipe surface (10).
5. Wood is New Zealand Kraft pulp.

TABLE IIA Data[5,6] for use with Eq. 2 or Eq. 2M to determine head loss, $\Delta H/L$

Pulp Type[5]	K	α	β	y
Unbleached sulfite	12.69 (1438)	0.36	1.89	−1.33
Bleached sulfite	11.40 (1291)	0.36	1.89	−1.33
Kraft	11.40 (1291)	0.36	1.89	−1.33
Bleached straw	11.40 (1291)	0.36	1.89	−1.33
Unbleached straw	5.70 (646)	0.36	1.89	−1.33

Estimates for other pulps based on published literature.

Pulp Type[5,6]	K	α	β	y
Cooked groundwood	6.20 (501)	0.43	2.13	−1.20
Soda	6.50 (288)	0.36	1.85	−1.04

Note: When metric (SI) units are utilized, use the value of K' given in parentheses. When the metric values are used, diameter (D) must be in millimeters (mm) and velocity (V) in meters per second (m/s).

Appendix D

The following gives supplemental information to that where T.P.D. mill capacity (metric tons per day), provided in the main text.

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1. Capacity (flow), Q—

$$Q = \frac{16.65(\text{T.P.D.})}{C} \text{ (U.S. GPM)}, \quad \text{(i)}$$

where T.P.D. = mill capacity (short tons per day)
C = consistency (oven-dried, expressed as a percentage, *not* decimally)

If SI units are used, the following would apply:

$$Q = \frac{1.157(10^{-3})(\text{T.P.D.})}{C} \text{ (m}^3\text{/s)} \quad \text{(iM)}$$

where T.P.D. = mill capacity (metric tons per day)
C = consistency (oven-dried, expressed as a percentage, *not* decimally)

2. Bulk velocity, V—

$$V = \frac{0.321Q}{A} \text{ (ft/s)} \quad \text{(ii)}$$

or

$$V = \frac{0.4085Q}{D^2} \text{ (ft/s)} \quad \text{(ii)}$$

where Q = capacity (U.S. GPM)
A = inside area of pipe (in^2)
D = inside diameter of pipe (in)

The following would apply if SI units are used:

$$V = \frac{1(10^6)Q}{A} \text{ (m/s)} \quad \text{(iiM)}$$

or

$$V = \frac{1.273(10^6)Q}{D^2} \text{ (m/s)} \quad \text{(iiM)}$$

where Q = capacity (m^3/s)
A = inside area of pipe (mm^2)
D = inside diameter of pipe (mm)

3. Multiplification Factor, F (included in Eq. 2)—

$$F = F_1 \cdot F_2 \cdot F_3 \cdot F_4 \cdot F_5 \quad \text{(iv)}$$

where F_1 = correction factor for temperature. Friction loss calculations are normally based on a reference pulp temperature of 95°F (35°C). The flow resistance may be increased or decreased by 1% for each 1.8°F (1°C) below or above 95°F (35°C), respectively. This may be expressed as follows:

$$F_1 = 1.528 - 0.00556T_1 \quad \text{(v)}$$

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where T = pulp temperature (°F), or

$$F_1 = 1.35 - 0.01T_1 \qquad \text{(vM)}$$

where T = pulp temperature (°C)

F_2 = correction factor for pipe roughness. This factor may vary due to manufacturing processes of the piping, surface roughness, age, and so on. Typical values for PVC and stainless steel piping are listed below:

$F_2 = 1.0$ for PVC piping

$F_2 = 1.25$ for stainless steel piping

Please note that the previous values are typical values; experience and/or additional data may modify the factors.

F_3 = correction factor for pulp type. Typical values are listed below:

$F_3 = 1.0$ for pulps that have never been dried and reslurried

$F_3 = 0.8$ for pulps that have been dried and reslurried

Note: This factor has been incorporated in the numerical coefficient, K, for the pulps listed in Table II. When using Table II, F_3 should *not* be used.

F_4 = correction factor for beating. Data have shown that progressive beating causes, initially, a small decrease in friction loss, followed by a substantial increase. For a kraft pine pulp initially at 725 CSF and $F_4 = 1.0$, beating caused the freeness to decrease to 636 CSF and F_4 to decrease to 0.96. Progressive beating decreased the freeness to 300 CSF and increased F_4 to 1.37 (see K values in Table II). Some engineering jugment may be required.

F_5 = design safety factor. This is usually specified by company policy with consideration given to future requirements.

Appendix E

The following are three examples which illustrate the method for determination of pipe friction loss in each of the three regions shown in Figure 10.

EXAMPLE 1 Determine the friction loss (per 100 ft of pipe) for 1000 U.S. GPM of 4.5% oven-dried unbeaten aspen sulfite stock, never dried, in 8 in schedule 40 stainless steel pipe (pipe inside diameter = 7.981 in). Assume the pulp temperature to be 95°F.

Solution:

a. The bulk velocity, V, is

$$V = \frac{0.4085\,Q}{D^2} \qquad \text{(ii)}$$

and Q = flow = 1000 U.S. GPM

D = pipe inside diameter = 7.981 in

$$V = \frac{0.4085\,(1000)}{7.981^2} = 6.41 \text{ ft/s} \qquad \text{()}$$

b. It must be determined in which region (1, 2, or 3) this velocity falls. Therefore, the next step is to determine the velocity at the upper limit of the linear region, V_{max}.

$$V_{max} = K'C^{\sigma} \qquad (1)$$

and K' = numerical coefficient = 0.85 (from Appendix B, Table I)

C = consistency = 45%

σ = exponent = 1.6 (from Appendix B, Table I)

$V_{max} = 0.85(4.5^{1.6}) = 9.43$ ft/s

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c. Since V_{max} exceeds V, the friction loss, $\Delta H/L$, falls within the linear region, Region 1. The friction loss is given by the correlation:

$$\Delta H/L = FKV^{n}C^{\beta}D^{y} \tag{2}$$

and F = correction factor = $F_1 \cdot F_2 \cdot F_3 \cdot F_4 \cdot F_5$

F_1 = correction factor for pulp temperature. Since the pulp temperature is 95°F

$F_1 = 1.0$

F_2 = correction factor for pipe roughness. For stainless steel pipe,

$F_2 = 1.25$ (from Appendix D)

F_3 = correction factor for pulp type. Numerical coefficients for this pulp are contained in Appendix C, Table II, and have already incorporated this factor.

F_4 = correction factor for beating. No additional beating has taken place, therefore

$F_4 = 1.0$ (from Appendix D)

F_5 = design safety factor. This has been assumed to be unity.

$F_5 = 1.0$.

$F = (1.0)(1.25)(1.0)(1.0)(1.0) = 1.25$

K = numerical coefficient = 5.30 (from Appendix C, Table II)

σ, β, y = exponents = 0.36, 2.14, and −1.04, respectively (from Appendix C, Table II)

V, C, D have been evaluated previously.

$$\Delta H/L = (1.25)(5.30)(6.41^{0.36})(4.5^{2.14})(7.981^{-1.04})$$
$$= (1.25)(5.30)(1.952)(25.0)(0.1153)$$
$$= 37.28 \text{ ft head loss/100 ft of pipe}$$

This is a rather substantial head loss, but may be acceptable for short piping runs. In a large system, the economics of initial piping costs versus power costs should be weighed, however, before using piping which gives a friction loss of this magnitude.

EXAMPLE 2 Determine the friction loss (per 100 ft of pipe) of 2500 U.S. GPM of 3% oven-dried bleached kraft pine, dried and reslurried, in 12 in schedule 10 stainless steel pipe (pipe inside diameter = 12.39 in). Stock temperature is 1250°F.

Solution:

a. V, the bulk velocity, is

$$V = \frac{0.4085\,Q}{D^2} \tag{ii}$$
$$= \frac{0.4085(2500)}{12.39^2} = 6.65 \text{ ft/s}$$

b. The velocity at the upper limit of the linear region, V_{max}, is

$$V_{max} = K'C^{\sigma} \tag{1}$$

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and $K' = 0.59$ (from Appendix B, Table I)

$C = 3.0$ (from Appendix B, Table I)

$\sigma = 1.45$ (from Appendix B, Table 1)

$V_{max} = 0.59\,(3.0^{1.45}) = 2.90$ ft/s

c. Region 1 (the linear region) has been eliminated, since the bulk velocity, V, exceeds V_{max}. The next step requires calculation of V_W.

$$V_W = 4.00\,C^{1.40} \tag{3}$$

$$= 4.00(3.0^{1.40}) = 18.62 \text{ ft/s}$$

d. V exceeds V_{max}, but is less than V_W, indicating that it falls in Region 2. The friction loss in this region is calculated by substituting V_{max} into the equation for head loss, Eq. 2.

$$\Delta H/L = F\,K\,(V_{max})^{\alpha}\,C^{\beta}\,D^{\gamma}$$

and

$$F_1 \cdot F_2 \cdot F_3 \cdot F_4 \cdot F_5 \tag{iv}$$

$$F_1 = 1.528 - 0.00556T \tag{v}$$

and T = stock temperature = 125°F

$F_1 = 1.58 - 0.00556(125) = 0.833$

$F_2 = 1.25$ (from Appendix D)

$F_3 = F_4 = F_5 = 1.0$

$F = 0.833(1.25)(1.0) = 1.041$

$K = 8.80$ (from Appendix C, Table II)

$\alpha, \beta, \gamma = 0.31, 1.81$, and -1.34, respectively (from Appendix C, Table II)

V_{max}, C, and D have been defined previously.

$$\Delta H/L = 1.041(8.80)(2.90^{0.31})(3.0^{1.81})(12.39^{-1.34})$$

$$= 1.041(8.80)(1.391)(7.304)(0.03430)$$

$$= 3.19 \text{ ft head loss/100 ft of pipe}$$

EXAMPLE 3 Determine the friction loss (per 100 ft of pipe) for 2% oven-dried bleached kraft pine, dried and reslurried, through 6 in schedule 40 stainless steel pipe (inside diameter = 6.065 in). The pulp temperature is 90°F; the flow rate 1100 U.S. GPM.

Solution:

a. The bulk velocity is

$$V = \frac{0.4085\,Q}{D^2}$$

$$= \frac{0.4085(1100)}{6.065^2} = 12.22 \text{ ft/s} \tag{ii}$$

b. It must be determined in which region (1, 2, or 3) this velocity falls. To obtain an initial indication, determine V_{max}.

$$V_{max} = K'C^{\sigma} \tag{1}$$

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and $K = 0.59$ (from Appendix B, Table I)

$\sigma = 1.45$ (from Appendix B, Table I

$V_{max} = 0.59(2.0^{1.40}) = 1.61$ ft/s

c. Since V exceeds V_{max}, Region 1 (the linear region) is eliminated. To determine whether V lies in Region 2 or 3, the velocity at the onset of drag reduction, V_W, must be calculated.

$$V_W = 4.00\, C^{1.40}$$
$$= 4.00(2.0^{1.40}) = 10.56 \text{ ft/s}$$

d. V exceeds V_W, indicating that it falls in Region 3. The friction loss is calculated as that of water flowing at the same velocity.

$$(\Delta H/L)_w = 0.579\, V^{1.75}\, D^{1.25}$$
$$= 0.579(12.22^{1.75})(6.065^{-1.25})$$
$$= 4.85 \text{ ft head loss/100 ft of pipe} \quad (4)$$

This will be a conservative estimate, as the actual friction loss curve for pulp suspensions under these conditions will be below the water curve.

Digester Pump Applications Digester pump services encounter severe operating conditions and, therefore, require more consideration than some other pump services in the pulp mill. Two types of digesters are encountered in the pulping operation, continuous and batch. As both systems have similar pressure and temperature controls to produce pulp, the pump applications have similar characteristics.

Typical pressures encountered are the digester pressure that will be in the 250 to 300 psig range. The temperature range will be in the 200 to 350°F (93 to 177°C) range. The pumps that see these temperature and pressure ranges should be centerline-mounted design (Figure 12). The product pumped will include white liquor, black liquor, and pulp fiber depending on the location in the process.

The normal pump material is duplex stainless steel (such as ASTM A890 Grade 3A) due to the operating pressures encountered and the need for resistance to abrasion on many of the applications.

The operating conditions may result in severe pipe loading from thermal expansion. Some of the services may also encounter cyclical operations with the pump running throughout its curve thus creating high radial loads on the power end. These services may



FIGURE 12 Digester pumps with centerline mounting (Courtesy of ITT/Goulds Pumps, Inc.)

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also encounter some abrasives from sand carry over or the presence of pin chips and fiber from incomplete pulping operations. Another consideration on some of the digester pump applications is the NPSH available. Elevated operating temperatures and high flow rates involved in some circulation pump services coupled with limited suction head may create cavitation problems.

These operating characteristics may create high maintenance needs so pump selection should focus on operating at the slowest speed possible, properly sizing the pump for the duty conditions experienced and allow for a higher pressure and temperature operation with centerline mounting and enclosed impellers (Figure 13). Finally, considering variable speed drives to address large pump flow rate swings will enhance long-term pump life.

Medium Consistency Pump Services Paper stock pulp in the consistency range of 8 to 16% is referred to as medium consistency pulp. Pulp fibers become a very strong interlocked network and create high friction between the fiber and pipe wall. The fluid characteristics of pulp are clearly non-Newtonian. Traditional methods to pump pulp in the 8 to 16% consistency ranges began with positive displacement pumps and were extended to centrifugal pumps over the past 25 years.

The non-Newtonian fluid characteristics, both within the pump and with respect to friction loss characteristics in the pipe system, along with air content of the pulp, present specific challenges for either positive displacement or centrifugal pumps.

Perhaps the first question to address is—why a medium consistency centrifugal pump versus a positive displacement unit? Some of the advantages offered by centrifugal pumps over positive displacement pumps are:

- Lower installation cost
- Lower maintenance cost and significant ease of maintenance when required

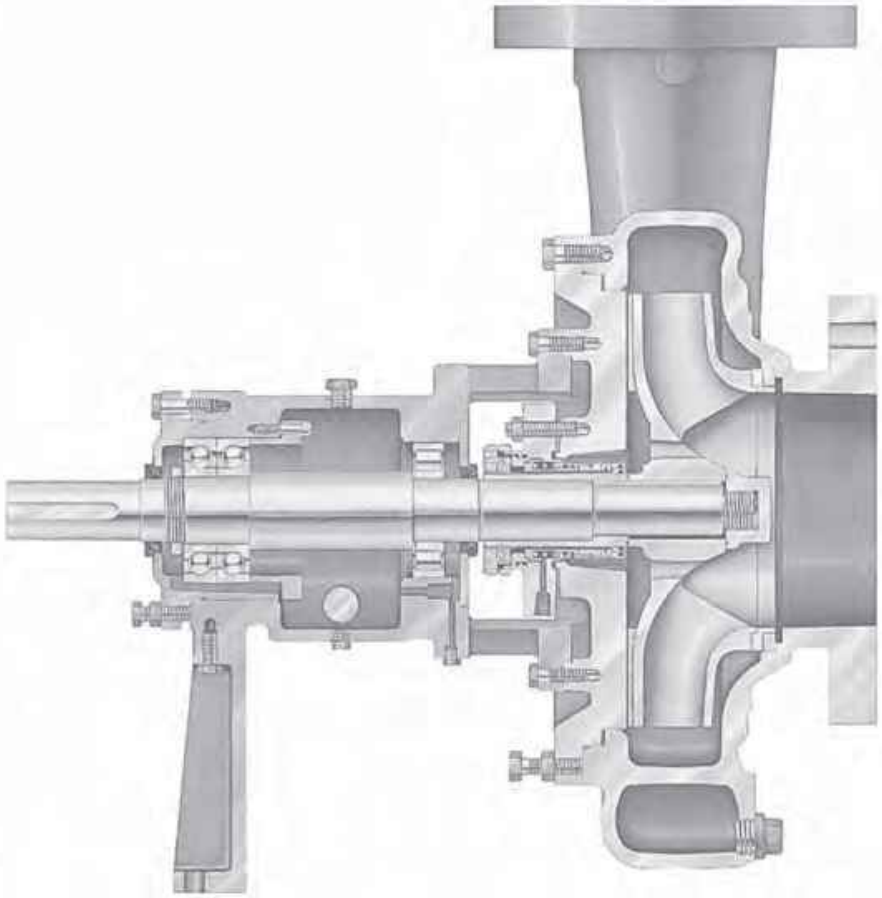


FIGURE 13 Digester pump with enclosed impeller (Courtesy of ITT/Goulds Pumps, Inc.)

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- Removal of air from pulp which can be detrimental in further unit operations such as chemical mixing in bleach plants
- Hydraulic capability to develop higher pressures often required in more modern pulping and bleaching systems

For effective centrifugal pumping of pulp suspensions in the 8 to 16% range there are three key areas to success. They are:

- Delivery system to the pump
- Air removal
- Simple control scheme

Delivery System to the Pump Figure 14 represents the most common configuration of medium consistency pump installation. The standpipe arrangement receives pulp from the stock source and provides a self-contained entity to accomplish delivery of pulp to the pump and the means to control the unit. Common sources of stock to the medium consistency pump can be a vacuum or pressure washer or disc filter.

The medium consistency pump is sized from a flow standpoint based on the required production rate of the process. Production rate changes translate into flow rate changes for the medium consistency pump. The table below illustrates the pump flow rate as a function of production rate with pulp consistency held constant.

The relationship of flow rate to production rate is:

$$Q = P \times 15/C$$



FIGURE 14 Medium consistency pump standpipe arrangement (Courtesy of ITT/Goulds Pumps, Inc.)

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where

Q = flow rate (GPM)

P = production rate (air dry short tons/day, adstpd)

C = pulp consistency (bone dry %)

Flow Rate at Different Production Rates

Production Rate, P (adstpd)	% C	Flow Rate, (GPM)
1000	12	1250
750	12	938
500	12	625

A last important aspect of the pump suction system relates to the pulp temperature. Typically, operation up to 170°F (76°C) can be accomplished in a normal 14 to 15 in tall standpipe. This will allow for 4′ to 6′ of suction head of pulp above the pump centerline. As pulp temperature increases, then the suction level above the pump must be increased to prevent flashing to vapor at the pump inducer or in the degas system. The increase in pulp operating level must be accompanied by a corresponding increase in standpipe height. On occasion in existing systems, there will be limitations which can present problems. For long-term, reliable operating performance this aspect should not be compromised.

Degas or Air Removal System This aspect of the pump system design is vital to successful operation of the pump. As many have experienced, the presence of even small amounts of air can reduce the effective performance of a conventional centrifugal pump. With medium consistency pulp suspensions between 8 and 16%, the air content encountered will be at least equal to the pulp consistency and often higher. Without air removal the pump operating control design will falter and flow rate reduction—or production rate tonnage from the operation viewpoint—will occur.

The air removal must be effective over a wide range of flow rates (production rates) and consistencies. For this to properly occur requires the functional integration of the design of the casing volute configuration, air removal holes, and air removal mechanism.

Control Scheme Figure 15 illustrates the control scheme normally used for a standpipe configured medium consistency pump. The importance aspect of the control scheme is the effective response to system changes. The system should be automatic and not require manual control or monitoring to make adjustments. The level control scheme is most common for

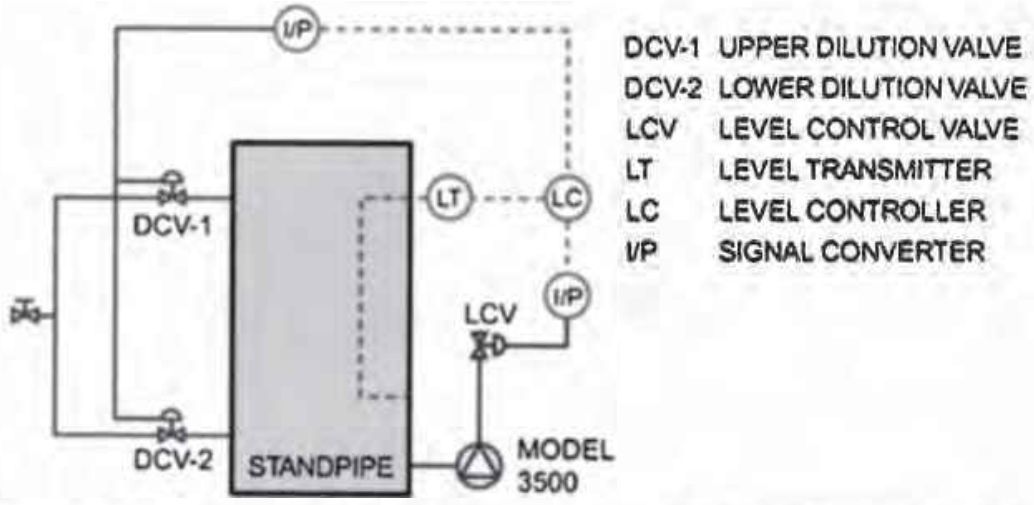


FIGURE 15 Medium consistency pump standpipe level/dilution system (Courtesy of ITT/Goulds Pumps, Inc.)

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standpipe design units. The unit operates to a set point level. Should the level deviate beyond the deadband range of the set point, control valve modulation will attempt to control to the set point. If the level change continues, then dilution valves open and the consistency is reduced. This allows the pump to regain control and return to the level set point. As the level returns to the set point, the dilution is reduced until it once again is off and the unit is controlled to the set point. Variable speed operation will differ somewhat for this method.

An example using a common situation that can arise would involve an increase in pulp consistency from the washer. Assume a unit is designed to handle 12% consistency pulp. If the consistency increases quickly to 14%, the system TDH would begin to increase dramatically. This is because consistency has, by far, the greatest impact on friction in the system. As the level moves rapidly outside the deadband (usually +/−10%), the pump moves back on its curve to develop sufficient TDH to deliver pulp through the system. Modulation of the level control valve when the production rate is near its design may not allow the unit to regain control. Sequencing of valves in the standpipe begins reduction of the pulp consistency to reduce system friction. This allows the pump to increase the flow rate. As the flow rate through the pump increases, the standpipe level begins to reduce and the standpipe level returns to its set point. As the consistency of the pulp being delivered to the pump returns to its design range, the system returns to the normal control mode and dilution is stopped.

Fan Pump Applications The importance of minimizing pressure pulsations in a headbox supply system has been emphasized with the inception of the hydraulic headbox to achieve higher paper machine speeds. This type of headbox can be used on twin wire or conventional fourdrinier paper machines, on paper grades ranging from corrugated mediums to tissue paper. Experience has shown that rotating equipment can be a source of pressure pulsations at the headbox, resulting in excessive machine direction basis weight variation (density variation) and thus poor quality paper as well as paper machine operation problems.

Pressure variations at the headbox will be either periodic or random. Operation with conventional airpadded headboxes was designed to control the periodic pressure variation. There was less emphasis on the random variation which often masked many periodic variations. Fan pump impeller specifications requiring impellers to have split, staggered, and skewed vanes evolved to increase the frequency, and reduce the magnitude of periodic pulsations at the vane pass frequency.

From a pulsation viewpoint, the fan pump is one component of the stock delivery system which deserves concern. However, other factors closely related to the installation of the fan pump also deserve careful consideration.

LOW PULSATION IMPELLERS Recent emphasis on minimizing pressure pulsations in fan pumps is placed on those factors which create pressure variations at low frequencies (i.e., 1 × rpm) rather than vane pass frequencies since these are above the range of concern for most hydraulic headboxes. One pressure pulse associated with fan pumps is the vane pass frequency pulse.

To reduce this pulsation level, pump manufacturers produce double suction impellers with split, staggered, and skewed vanes, thus reducing the magnitude of the pulse and increasing the frequency by having more vanes passing each cutwater per revolution. The vane pass frequency pulse is generally the largest magnitude pulse produced by the pump.

However, due to friction resistance within the system and the damping characteristics of the headbox, the vane pass frequency pulse rarely creates pressure pulsation problems. Due to its relatively long wave length, the low frequency (1 × rpm) pulse will not be dampened and can pass much more readily through the hydraulic headbox. This results in pressure variations at the headbox of a magnitude which results in unacceptable machine direction basis weight variations on the sheet. Control of the 1 × rpm pulse requires special attention to the design criteria and manufacture of the impeller. The types of nonuniformities which can create a 1 × rpm pulse include balance, singular surface irregularities, surface eccentricities, and surface finish.

Pressure pulsation requirements focus on the frequency range from 0 to approximately 25 Hz. The (1 × rpm) frequency of a fan pump is obtained by dividing the pump shaft rpm by 60 to obtain the frequency in hertz. A fan pump will only cause pressure pulsations at integer multiples of its shaft speed. Maximum pressure pulsation levels will depend on the headbox manufacturer, but they typically range from 0.15 to 0.25 psi (peak to peak), or greater.

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Other Design Considerations While the prime concern of the pump manufacturer is to minimize sources of the 1 × rpm pulse within the pump, external factors can influence the pump output or result in undesirable system effects when acting in conjunction with the headbox supply pump. To minimize potential pulsation problems, careful attention must be paid to the total system including the pump suction piping.

A common fan pump installation will utilize a suction transition piece between the wire pit silo and the pump suction. Two or more recirculation lines may enter this transition piece. These flows entering into the main body of the transition piece may be unequal. If there is a wide disparity between the flows from the recirculation lines, one side of the double suction impeller may be starved and the other side overloaded. In addition to increasing pressure pulsations at the pump discharge, such unequal loading of the impeller could cause pump operational problems.

Recirculation lines can be utilized for headbox overflow, primary and secondary cleaner accepts or rejects, rotary screen rejects, or thick stock addition for basis weight control. Generally, pressure pulsations can be transmitted along these lines to the pump suction. The fan pump may dampen these variations but will not completely eliminate them. It is also possible for pressure pulsations, caused by throttling valves in these lines, to pass directly back through the recirculation line and appear at the headbox since, as fluid-borne noise, pressure pulsations travel in all directions.

To minimize the pressure pulsations within a system, there should ideally be no recirculation lines entering the pump suction transition piece. All return lines should recycle directly into the wire pit silo which provides a large mass with a free surface to dampen out any pressure variations transmitted in the recirculation lines. A less desirable alternative would have the recirculation lines enter on the vertical centerline of the transition piece at a 45° angle. If thick stock is added, this line should be located closest to the pump. Recirculation line velocities should be a maximum of 1.1 to 1.5 times the velocity in the transition piece at the point of entry. The branch line nearest the pump should be no closer than a distance equal to twice the pump suction diameter.

DISCHARGE PIPING The factors which should be considered when designing the discharge piping of a fan pump include, but are not limited to, elbows located at or near the pump discharge, throttling valves, and piping support in the vicinity of the pump. Bends or 90° elbows close to the pump discharge can contribute to pressure pulsations and should be avoided.

A throttled valve can create pressure variations at frequencies similar to the fan pump frequency. In this situation, amplification of existing pressure pulsations can occur. Poorly supported discharge piping, which is susceptible to vibration, may amplify existing pressure pulsations in a stock delivery system.

PUMP DRIVE SYSTEMS Experience has shown that resonant system conditions often occur in stock delivery systems at, or near, normal fan pump operating speeds. It is, therefore, desirable to operate a pump at a different speed in order to move the pump frequency away from a system resonant frequency. Additionally, the existence of a system resonant condition can be ascertained by varying the pump speed and determining if the frequency of the pressure pulsations varies directly with pump speed (pump induced), or remains at a constant frequency regardless of the pump speed (system resonant condition). When constant speed drivers are used, the pump frequency cannot be varied; this makes it difficult to accurately determine sources of pressure pulsations as well as to avoid problems should they occur. For this reason, serious consideration should be given to a variable speed drive arrangement. An additional benefit of a variable speed drive system is the ability to closely match the system head with the fan pump performance curve. This allows high pump efficiency to be maintained over a wide range of operating conditions. Also, the discharge throttling valve can operate in a more desirable range and may even be eliminated.

OTHER SYSTEM COMPONENTS Pressure screens will often be used in a stock delivery system to remove foreign matter and provide uniformity to the stock. This is done by passing the stock stream through a screen at a low pressure (3 to 7 psi) differential. Materials

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which do not pass through the screen are removed by rotating foils or scraper blades. When these foils pass the screen discharge, a pressure pulse may be generated.

Consideration should be given to the relative operating frequencies of the pump and screens. It is possible that, if the screen frequency is close to other rotating equipment frequencies, reinforcement of existing pressure pulsations can occur. It is suggested that all pump operating frequencies be separated from the screen foil frequency by at least 20%.

Pressure Pulsation Testing To evaluate the pressure pulsation characteristics of a fan pump, pressure transducers and a spectrum analyzer are required. The pressure transducers are designed to measure variations in system pressure at the point of installation within the system. Transducers will normally be installed after the process equipment but may also be located near the inlet. The key to meaningful analysis of pressure pulsation data is the spectrum analyzer. This device performs a frequency analysis of the pulsation data, separating the signal from the transducer (amplitude vs. time) into its various components (amplitude vs. frequency). The vibration analyzer breaks down components of vibrations by individual frequencies or frequency bands.

This analysis can be conducted in both a factory and a mill environment. It can be expected that in a mill environment, additional pressure variations would be seen. Examples would be screen foil frequencies, valve cavitation, reflections from piping elbows, mechanical vibrations, and system resonances. A complete analysis using pressure transducers located throughout a headbox supply system would enable overall effects of different components to be determined, as well as demonstrate interaction of system components.

Recycle Service Applications Pumping recycled fiber presents several unique challenges that other paper services do not. This is the result of the shorter fibers that tend to dewater easier and the inherent contaminants contained in the bundles of recycled products used to start the process. The number one pump problem in the recycle paper process is plugging at the impeller eye. This plugging is due to large contaminants (strips of tape, wire, cans, and so on). The solids start to build up at the eye of the impeller in two significant ways, first, a general clog/plug between the vanes where the passageways are the narrowest. This is the result of simply too many solids trying to get through too small of a passage. Second, plastic straps or banding can be looped over or caught on the leading edge of a vane. If this banding doesn't get cut by the impeller or manage to slide off the vane's leading edge enabling the pump to clear itself, these contaminants will eventually rope together and create a plug in the suction of the pump.

The worst fibers are old corrugated containers (OCC). In addition to having a high content of contaminants, a high filler content (clays) is also present. The fillers sometime bind the contaminants together, making them larger. In addition, the course OCC fibers and the clays used as fillers are quite abrasive. As a result, OCC fibers and their inherent contaminants cause higher wear than most recycle fibers such as newsprint. The easiest fibers to pump are de-inked newsprint. There is no impeller plugging problems usually associated with these fibers.

Listed below are the names of some of the most difficult services:

	Pulper Pumps	Screen Pumps	Rejects Pumps
	Hydropulper	Primary Secondary Tertiary Slotted Hole Course Fine	Primary Secondary Tertiary
Comments:	Initial pump in system Sees all fibers/ contaminants from recycle source	Initial group of 6 to 10 pumps in system	Contaminants and fillers tend to concentrate in the rejects system

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When selecting pumps for a recycle fiber project it is always important to know:

1. What type and size of the contaminants the pump may see.
2. The service names of the pumps. They can give you an idea of the type of fluid and solids might be encountered in the application.

Note: Just because the data sheet may say 2%, there may be contaminants as well. Depending on the type and size of these contaminants, use of a recessed impeller pump or a modified paper stock pump may be appropriate. Modifications may include an opened impeller eye, increased casing/sideplate-to-impeller clearance, or a slotted sideplate.

RECESSED IMPELLER (VORTEX) PUMPS These pumps should be used on long stringy solids like tape, plastic, and wire (see Figure 16). They should also be used on the first pump in the system (usually called the hydropulper pump).

MODIFIED STOCK PUMPS A modified stock pump can be used on some of these applications. Typically, a three or 2/4 vane impeller is used in conjunction with a slotted sideplate (see Figure 17). The minimal number of impeller vanes allows a larger solid size to pass and the slotted sideplate will act as a shearing device, cutting the straps or banding should they get caught between the impeller and sideplate.

A two-vane impeller could also be used, but depending on the installation and application, but don't be surprised if you see vibration levels in excess of 0.3 in/s.

Another reliable alternative is the use of an impeller specially designed with long stringy solids in mind. These impellers have a wide open eye, a heavily raked vane leading edge, and a large impeller-to-casing/sideplate clearance (see Figure 18). These impellers are excellent at passing these long plastic scraps of banding commonly associated with recycled fiber. Any stringy solids caught on the leading edge of the vanes will slide off the impeller and into the process with being permanently caught on the impeller causing a roping plug.

Another modification that can be helpful is slotting the backplate of a dynamic seal. Recycled fiber has a tendency to clump up and dewater, especially in the dynamic seal area. The addition of the slots in the backplate has been proven through experience to create additional turbulence behind the repeller (see Figure 19). This additional turbulence

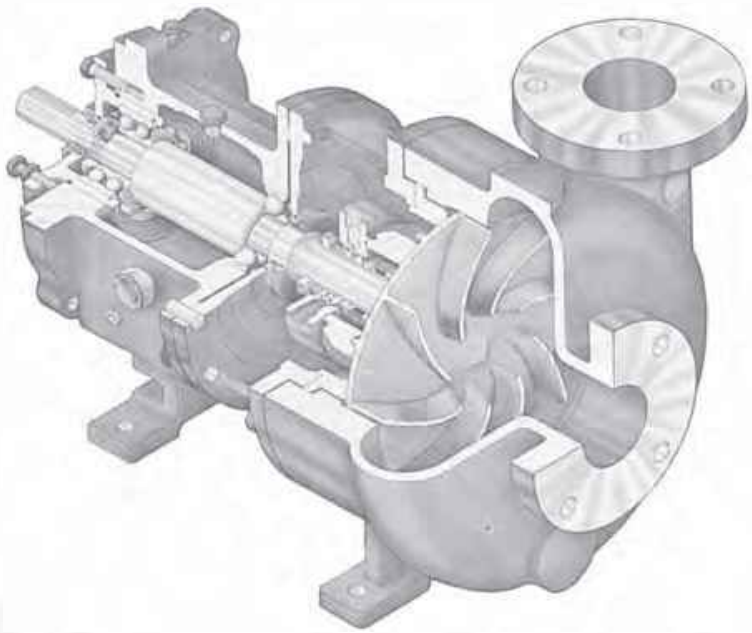


FIGURE 16 Recessed impeller (vortex) pump (Courtesy of ITT/Goulds Pumps, Inc.)

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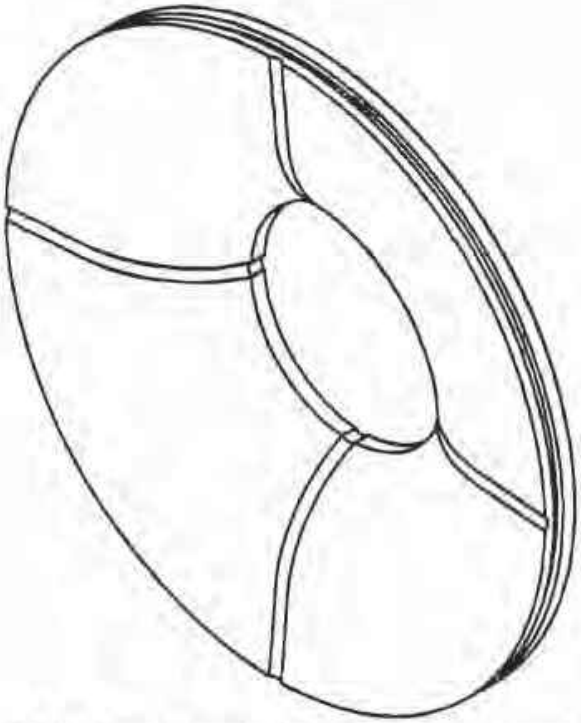


FIGURE 17 Slotted sideplate (Courtesy of ITT/Goulds Pumps, Inc.)

keeps the stock moving and prevents it from getting stuck between the repeller and backplate and quickly wearing the backplate.

Because of the high content of contaminants and fillers found in some recycle fibers the wear rate will be higher than on normal virgin fibers. To combat the wear and extend pump life, it is recommended that either 316 ss as a minimum, or preferably a form of duplex stainless steel be used for its increased hardness and greater resistance to wear.

Air Handling Applications Pump applications in many industrial processes involve handling liquid and gas mixtures. The entrained gas may be an essential part of an industrial process, or it may be unwanted. The pulp and paper industry, for example, injects

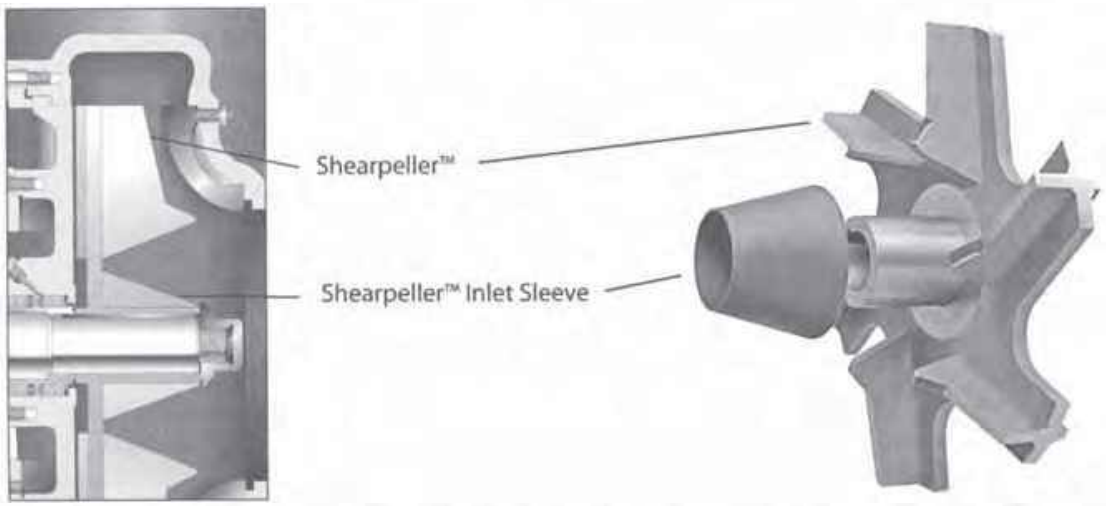


FIGURE 18 Impeller—sideplate design for long, stringy solids (Courtesy of ITT/Goulds Pumps, Inc.)

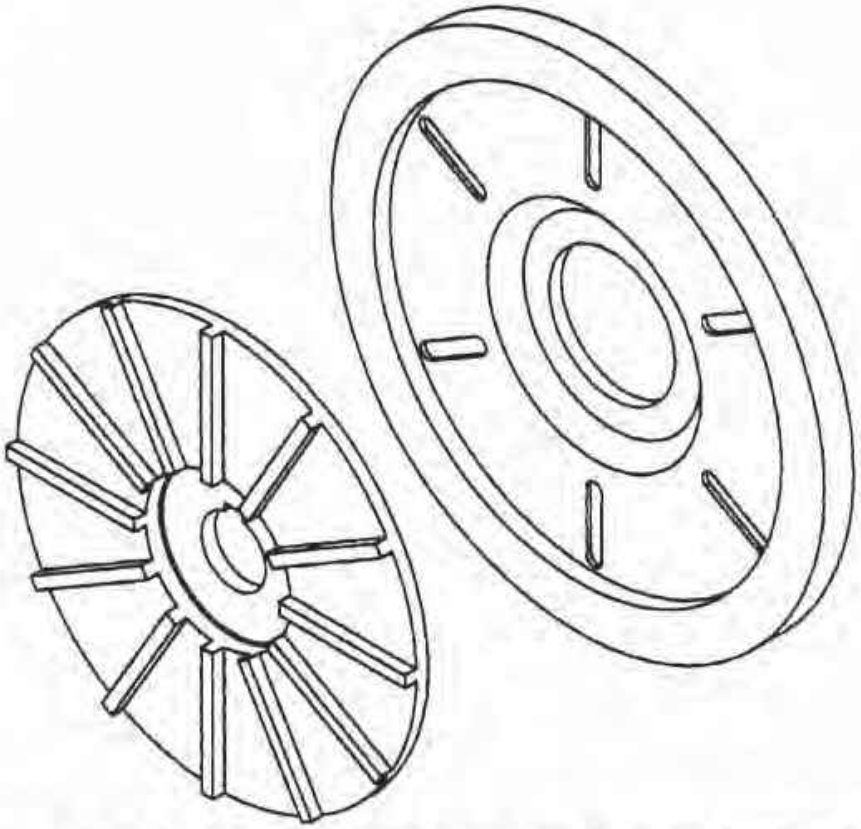


FIGURE 19 Slotted backplate provides turbulence to keep stock moving (Courtesy of ITT/Goulds Pumps, Inc.)

4 to 10% air into a dilute recycled pulp slurry as part of the ink removal process in a flotation cell. Many chemical and petrochemical processes also involve pumping a two-phase flow. Unwanted entrained gas can result from excess agitation or vortexing due to inadequate submergence on the suction of a pump.

The proper selection of a centrifugal pump for liquid and gas (two phase) mixtures is highly dependent on the amount of gas and the characteristics of the liquid. The presence of entrained gases will reduce the output of centrifugal pumps and can potentially cause loss of prime. Depending on the severity of the conditions, there are several possible solutions to this pumping challenge. This section will discuss these possible solutions, their limitations, and applications.

Ranges of Air Entrainment The first range is 0 to 4% entrained air. In this case, the size of the impeller must be increased to account for the effects of the air. This case would apply when designing a new system and the air content is known to be 0 to 4% or when trouble shooting an existing system.

The second range is for 5 to 10% entrained air. To account for the decrease in performance of the pump due to the air, the size of the impeller must be increased and may also require a larger size pump all together. This range also requires increased clearances between the impeller and casing/suction sideplate. Again, this is used in new systems or when solving existing field problems.

The third range is for 10 to 20% air. The best way to handle this case is to use a specially designed pump intended to handle the entrained gas. This would typically be a recessed impeller (or vortex) pump. An inducer may also be used on a modified conventional pump to boost its air handling capability above the normal 10% limit.

The final case is the use of an inducer. Typically, the inducer is thought of more as a problem solver for existing systems as opposed to part of a designed system. However, as inducers become more popular, it is becoming more common to see them used as part of an original system design.

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RANGE 0 TO 4% AIR Most centrifugal pumps have the ability to pump a small amount of entrained gas without major changes. However, impeller diameter corrections must be made to account for the loss of performance caused by the gas bubbles. The gas bubbles occupy space that would otherwise be occupied by the fluid. Consequently, the head output of the pump at a given flow is reduced. Increasing the impeller diameter will account for this loss of performance.

When air moves into the pump, the gas bubbles expand as they enter the eye of the impeller. This is because the fluid picks up a great deal of speed from the impeller. As the fluid speed increases, the pressure in that area of the pump decreases. With the decreasing pressure, the air bubbles are given a chance to expand. When the air concentration exceeds 4%, the air bubbles will accumulate in the low pressure, suction area of the pump; eventually the air will choke off the pump, and the pump will lose its prime.

Checking Figure 20, you can see that the head loss due to entrained gas concentrations of 2 to 4% can be as little as 5% of the pump performance for 2% below 60% of best efficiency point (BEP) up to 20% for 4% at BEP.

RANGE 5 TO 10% AIR As the gas concentration increases over 4%, the bubbles start to gather at the suction eye. Increased sideplate/impeller clearances will help break up the bubbles, moving them through the pump. The increased clearances create recirculation from the discharge side of the impeller to the suction eye, which retards the formation of large gas pockets. However, as a result of the increased clearances, the pump's efficiency is dramatically reduced. This compounds the effect of decreased performance from the entrained gas. Consequently, the pump selection will most likely increase in size with a considerable impeller diameter correction. The pump selection must be made such that the rated impeller diameter after correction is greater than or equal to 85% of the maximum diameter.

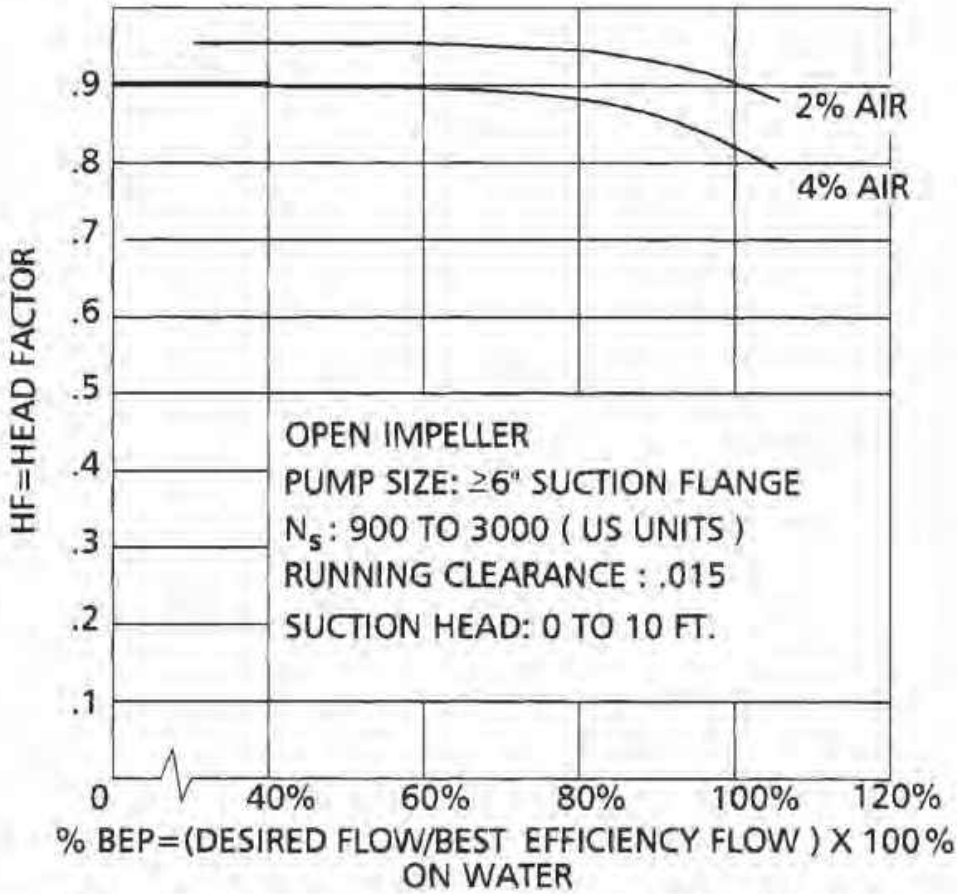


FIGURE 20 Head correction factors, 0–4% air (Courtesy of ITT/Goulds Pumps, Inc.)

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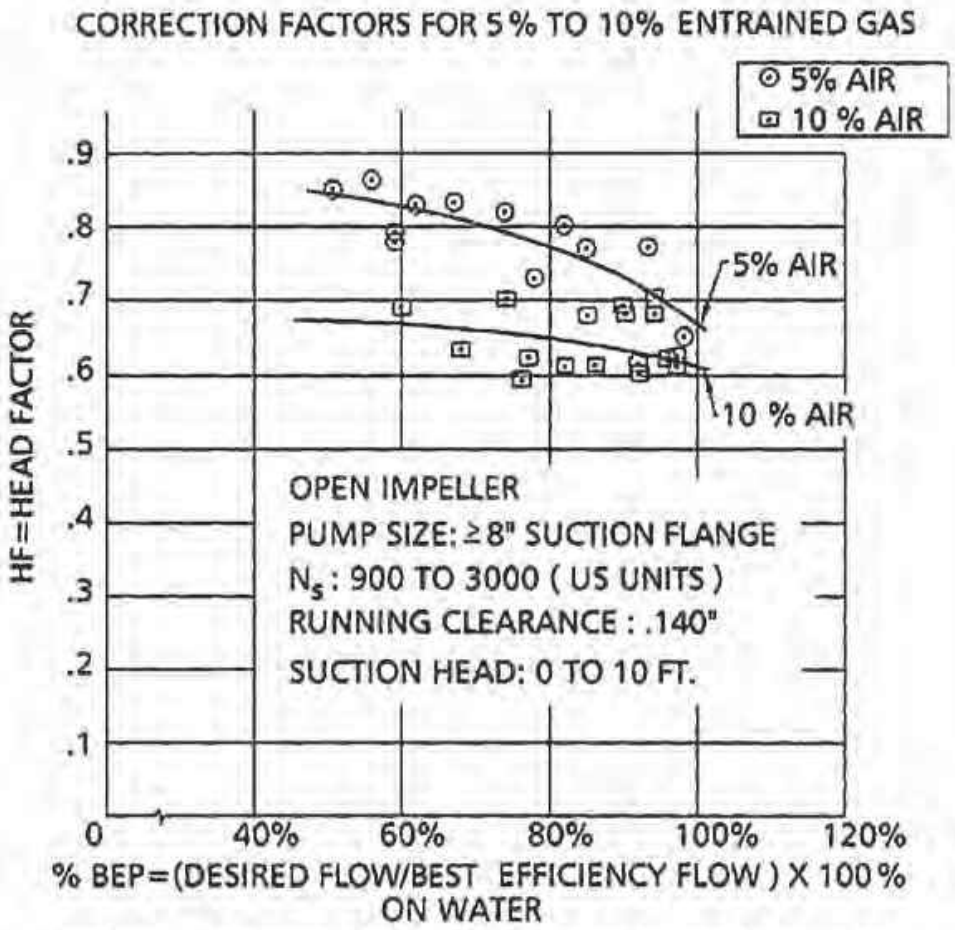


FIGURE 21 Head correction factors, 5–10% air (Courtesy of ITT/Goulds Pumps, Inc.)

Checking Figure 21, we see that the correction factors are in the 0.85 to 0.6 range. Meaning, with 5% gas at 50% BEP, you would experience a 15% loss in pump head and with 10% gas at BEP, you would see a 40% loss in pump head performance.

It is important to note that when using a conventional end-suction pump, whether with or without modification for air/gas handling, the rating/operating point should always be to the left of the BEP. The pump will not operate at or to the right of BEP because even with the increased impeller-to-casing/sideplate clearance, there will not be enough recirculation over the vanes to break up the collecting gas at the eye of the impeller.

RANGE 10 TO 15% AIR When gas concentrations exceed 10%, many times a recessed impeller pump can be used effectively. However, at these concentrations, the volume of air being pumped is rather large. Therefore, there is a very large head correction that needs to be applied for satisfactory operation.

Figure 22 shows the effect of entrained air/gas on a recessed impeller pump. As with the previous charts, the higher the air/gas concentration, and the higher the losses of the pump head performance. However, unlike the previous correction charts, you'll see here that pump performance does not worsen as the operating point approaches the BEP. To the contrary, performance actually improves. This is due to the pump design.

Inducers Inducers have several different uses and can be applied across all the ranges listed above. The primary function in an entrained air application is to agitate the air, keeping it in suspension and to increase the suction pressure into the pump. By agitating the air, it reentrains the air, increasing its pumpability and preventing the loss of prime due to the air.

A by-product of this action is the slight generation of head by the inducer. This head generation effectively increases the suction pressure into the pump. It has been shown in

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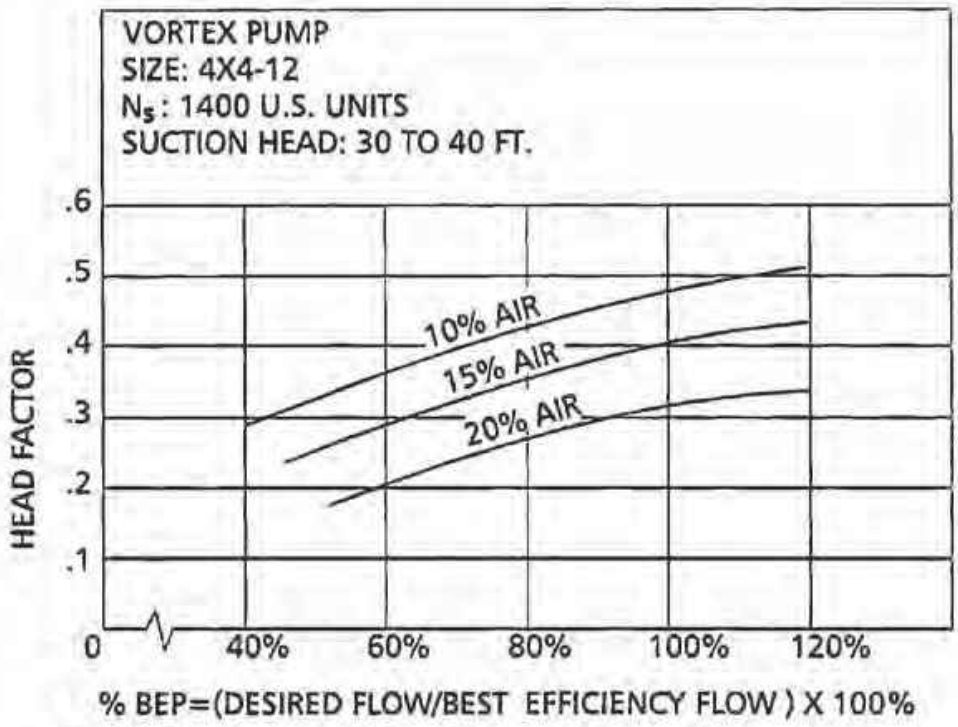


FIGURE 22 Head correction factors, 10–20% air (Courtesy of ITT/Goulds Pumps, Inc.)

testing that increased suction pressure greatly increases the pumpability of an air-entrained fluid. Because the inducer is generating head, albeit a small amount, it is doing work and, consequently, requires power input. All inducers will require some amount of horsepower input similar to a dynamic seal.

In all cases (0 to 4% air, 5 to 10% air and greater than 10% air), an inducer will reduce the correction factors used in sizing the pumps. With the decreased correction factors, at times smaller pumps can be used. The trade off for the benefit of the reduced corrections is increased power requirements and a somewhat reduced operating range. Instead of having the ability to run the pump from minimum flow to runout, it is now limited to a range of minimum flow to BEP.

There are two different designs that pump manufacturers commonly use. The first is the traditional style, Figure 23. The traditional inducer resembles a corkscrew. This design is used on clear liquids and is commonly referred to a flight inducer.

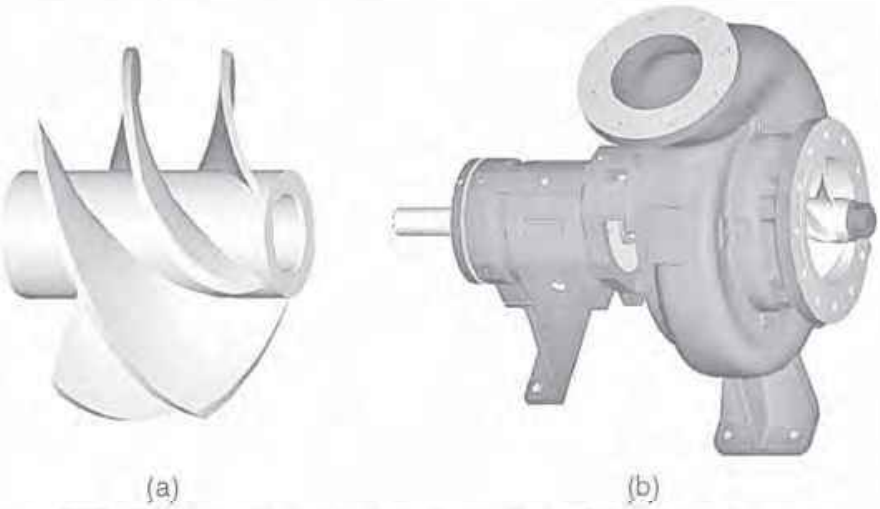


FIGURE 23 Traditional inducer design (Courtesy of ITT/Goulds Pumps, Inc.)

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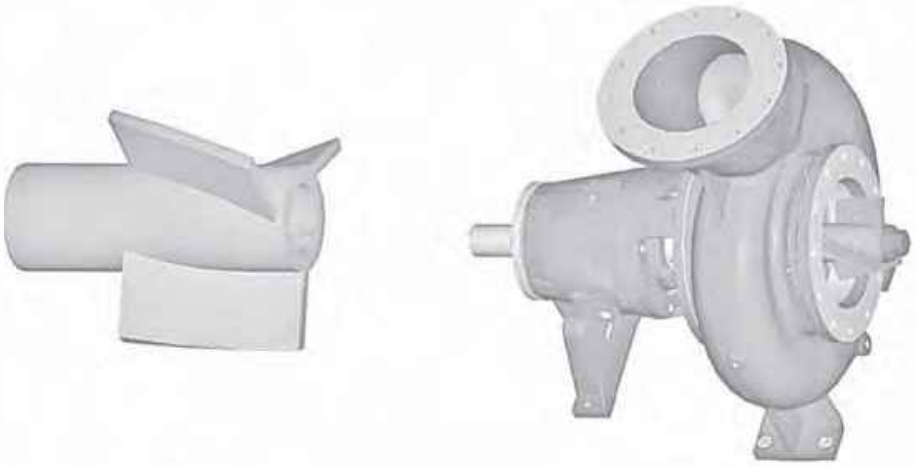


FIGURE 24 Feeder vane inducer style (Courtesy of ITT/Goulds Pumps, Inc.)

The second type is a feeder vane style, Figure 24. It has three blades or vanes that may appear as a mixer. The feeder vane inducer is used on pulp and nonhomogeneous liquids. The feeder vane inducer keeps the solids mixed and moving. It helps to retard plugging. Both styles require special design work for satisfactory performance.

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SECTION 12.10
FOOD AND BEVERAGE PUMPING

JAMES L. COSTIGAN
JOHN DRANE

INDUSTRY STANDARDS

Pumps for use in food, beverage, and pharmaceutical industries must meet various hygiene standards. These standards define the materials of construction and the "cleanability" of the pump. Unlike other industrial applications, pumps for the food and beverage industries must meet rigid sanitation codes, know in the industry as the "3-A" standards*. These standards were originally established for the dairy industry by the following organizations:

1. The International Association of Milk, Food, and Environment Sanitarians.
2. The U.S. Public Health Service.
3. The Dairy Industry Committee, composed of representatives from the following:
   - American Butter Institute
   - American Dry Milk Institute
   - Dairy and Food Industries Supply Association
   - Evaporated Milk Association
   - International Association of Ice Cream Manufacturers
   - Milk Industry Foundation
   - National Cheese Institute
   - National Creameries Association

*Available from the International Association of Milk, Food and Environmental Sanitariums, Box 437, Shelbyville, IN 45176.

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In the U.S., the 3A standards are widely enforced. They are recognized internationally, but are not enforced to the same degree. The 3A standards are a self-certifying standard, and they consider design and construction features only. They do not require microbiological testing for cleanability.

In Europe, the "Supply of Machinery (Safety) Regulations" (1992) defines the safety and hygienic requirements for equipment to be used on agri-foodstuffs. This is legislative and is not a voluntary standard. Compliance with these regulations enables the CE Mark to be attached to the machine. Noncompliance means machines cannot be sold within the European community (EC). The European Committee for Standardization (CEN) will produce a standard (TC 197) for pumps for food use, in support of the "Machinery Regulations" that will eventually form the controlling standard for all pumps designed for food use in the EC.

An independent group, formed mainly from users of sanitary equipment (European Hygienic Equipment Design Group—EHEDG), has produced guidelines for all types of equipment, including pumps. This group acts in an advisory capacity to CEN and other standardization bodies. These guidelines are generally very onerous, but they are often specified by end users of hygienic equipment.

The principle requirements of hygienic pumps are that the wetted parts should be compatible with the products being pumped, and that the pump can be easily cleaned, either by dismantling or by clean-in-place (C.I.P.) processes. Externally, the pumps should be smooth and free from crevices where dirt could lodge and bacteria or insects could flourish.

Emerging standards issued by the American Food and Drug Authority (AFDA) and European Directives based on these standards specify suitable materials for contact with foodstuffs. They also define any restrictions that may apply to them and test methods to prove compliance. These standards are recognized internationally and are widely used.

PUMP DUTIES

Pumps are used for filling, emptying, transferring, dosing, and mixing. They are also used to convey the process fluid through plant items with a high resistance to flow, such as membrane filters and heat exchangers. In general, pumping should not harm any solids in the liquid, and delicate shear-sensitive products must be handled gently. The choice of a particular pump depends on consideration of a number of factors. These include capacity, delivery pressure, and suction conditions. The calculation of system pressure losses is discussed in Section 11.1.

It is important to consider the viscosity of the liquid at the actual shear rate and temperature appropriate for the system. Newtonian liquids have a constant viscosity with shear rate (for example, glucose). Nonnewtonian liquids such as tomato ketchup have a viscosity that is dependent on shear rate. Some liquids become less viscous with increasing shear rate; these liquids are called *thixotropic*. Examples of a thixotropic liquid are starch and molasses. Others, less commonly, become more viscous with increasing shear rate; these liquids are called *dilatant*. An example of a dilatant liquid might be some candy compounds. Most liquids also become less viscous with increasing temperatures.

The fact that viscosity changes with shear rate and temperature means that a careful assessment must be made of all fluids that we intend to pump. This assessment would normally produce three values of viscosity that must be considered in the pump and system design.

1. Viscosity in the storage tank (low shear)
2. Viscosity in the pipe work (medium shear)
3. Viscosity in the pump (high shear)

A shear-sensitive material, which will degrade with work, would normally need a positive displacement pump with low shear characteristics, pump speed being selected accordingly. If solids in the product are not to be damaged, a pump with suitable-sized spaces and

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cavities must be selected. In some cases, high shear rates will be advantageous to the process (for example, mixing of emulsions) and the pump and system can be selected accordingly.

The chemistry of the liquid will determine the compatibility of the pump materials. The viscosity of the process fluid and its rheological properties affects both the speed at which the pump can run and also the friction losses to be expected within the system. The friction losses will affect the power required to run the pump. Generally, the more viscous a product, the slower the pump will need to run. If the product is very viscous, larger suction piping may be required, or in some cases, special enlarged inlets are used with auger assistance.

If the product is abrasive or carries hard solids in suspension, again pump speed will often need to be reduced to give economic life to pump components. Often, pumps with resilient components or those that do not have fixed clearances are better suited for this service. If, however, the solids in suspension are liable to settle at low speeds, speed must be increased to keep the solids in suspension to protect component life.

The product temperature may affect the suction performance capability of the pump. Also, for high fluid pumping temperatures, the pump and pumping elements must be selected to take account of differential expansion or contraction resulting from temperature variations. In some cases, pump performance is limited because of liquid properties.

PUMP TYPES

Food and beverage pumps are divided into two generic types: rotodynamic and positive displacement. Both types are widely used in the food and beverage industries.

Rotodynamic Pumps Rotodynamic (centrifugal) pumps are widely used in hygienic applications (Figure 1) because they are simple and inexpensive, they give a smooth delivery, and they are relatively easy to clean. They are, however, limited by viscosity that will greatly reduce pump efficiency above about 500 centipoise, generally regarded as the limiting viscosity. As the pumps rely on speed of rotation to generate head, there are fairly high shear rates within the pumping elements that affect shear-sensitive products. Also, because it is not self-priming, it has a limited ability to handle entrained gases and vapors. A positive feature of centrifugal pump designs is that the performance varies according to the system operating conditions to which it is subjected. Unlike a positive displacement pump, the discharge of a centrifugal pump can be partially or fully closed without over-pressurizing the pump.

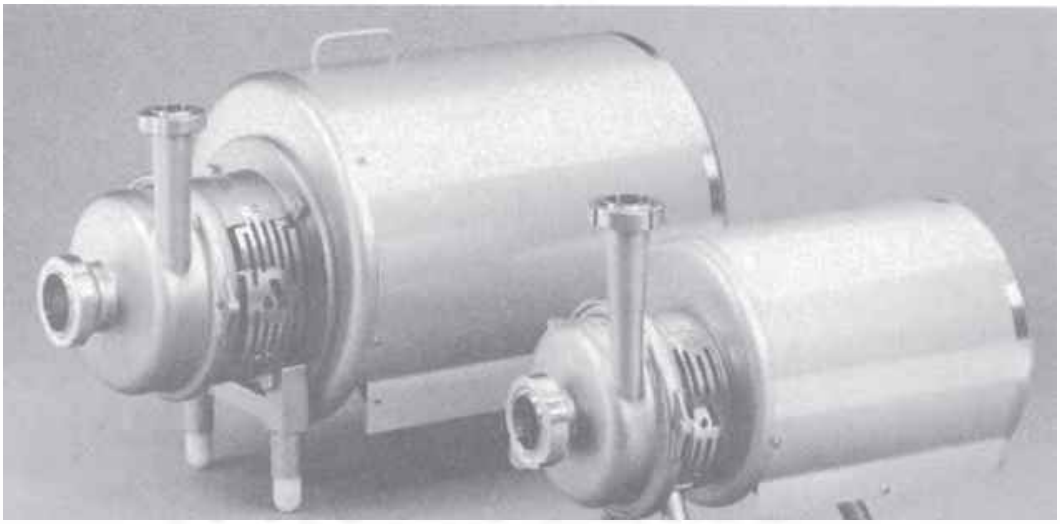


FIGURE 1 Hygienic centrifugal pump (APV Fluid Handling)

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ADVANTAGES These pumps are simple; they are of generally robust construction; pumps are easy to clean in place; pumps cannot be over-pressurized against a closed valve; they can be connected in series for increased pressure; and they can be connected in parallel for increased flow.

DISADVANTAGES These pumps typically are not self-priming; they are generally limited to about 500 centipoise viscosity; they are limited ability to handle fluids that are shear-sensitive; and they have a limited ability to handle high percentages of entrained gases.

Positive Displacement Pumps Positive displacement pumps are used in processes where viscosity limits the capabilities of a centrifugal pump or where the process needs to be versatile and the batches vary. They are normally self-priming and can handle gaseous products. However, all positive displacement pumps require pressure relief systems to prevent damage to the pump or system if a valve is closed or the outlet blocked.

Many types of positive displacement pumps are available and they are normally split into two groups—rotary and reciprocating, the reciprocating type needing a valving system to be able to operate. Valves increase shear rates and are susceptible to wear and blockage.

Gear Pumps

ADVANTAGES These pumps are self-priming; there is uniform discharge with little pulsing; the pumps are reversible; due to small clearances, little flow variation with change of viscosity or pressure.

DISADVANTAGES They can only pump clean fluids due to fine clearances and rubbing gear teeth; they cannot run dry; they need close tolerances to operate, so fits and alignment are critical.

Lobe Rotor Pumps (Figure 2)

ADVANTAGES These pumps can run dry for short periods as rotors have clearance, but run-dry time is limited by the seals; pumps can self-prime on low lifts; pumps can handle solids in suspension; there is little change of product velocity as it passes through the pump, so the pump has good net positive suction head (NPSH) performance, once primed; pumps can generally handle viscous materials; pumps are usually reversible.

DISADVANTAGES Fixed clearances can mean rapid performance degradation because of abrasive wear; pump has two shafts and associated shaft seals; suction lift limited at low speed/viscosity due to clearance of rotors.

Flexible Impeller Pumps (Figure 3)

ADVANTAGES Pumps are generally self-priming; pumps create relatively low pressure pulsations while pumping; pumps are reversible; pumps can handle gaseous fluids and fluids with some solids in suspension; pumps are easily cleaned.

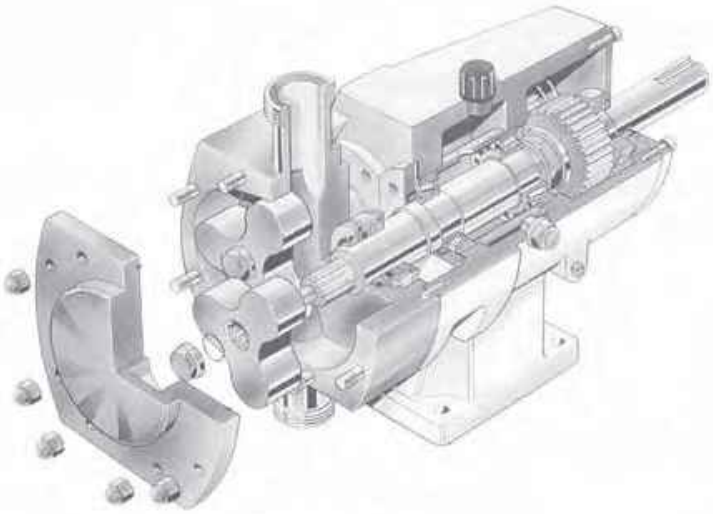
DISADVANTAGES Pumps cannot run dry; pumps are only suitable for low pressure applications; pumps are not suitable for abrasive services.

Progressive Cavity Pumps (Figures 4 and 7)

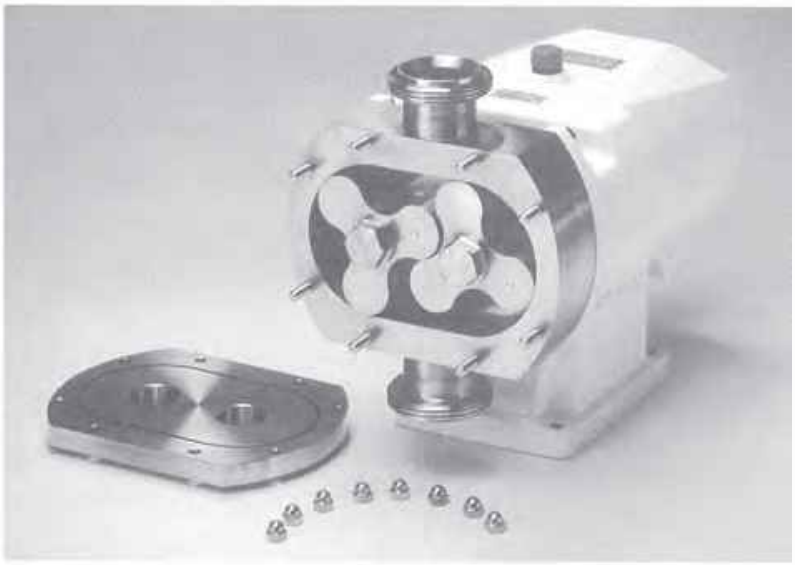
ADVANTAGES These pumps are self-priming; they exhibit uniform discharge with little pulsation; they are reversible and can handle gaseous fluids; they can handle solids in suspension and abrasives and can handle shear-sensitive fluids with little damage.

DISADVANTAGES The pumps cannot run dry.

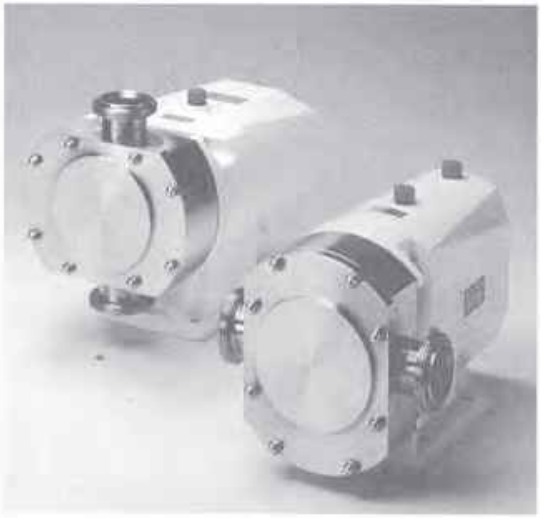
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A



B

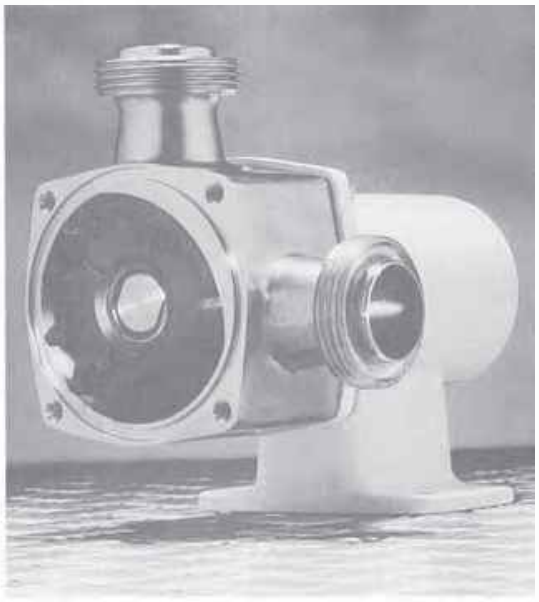


C

FIGURE 2A through C Rotary lobe pumps (Alfa Laval Pumps, Ltd.)

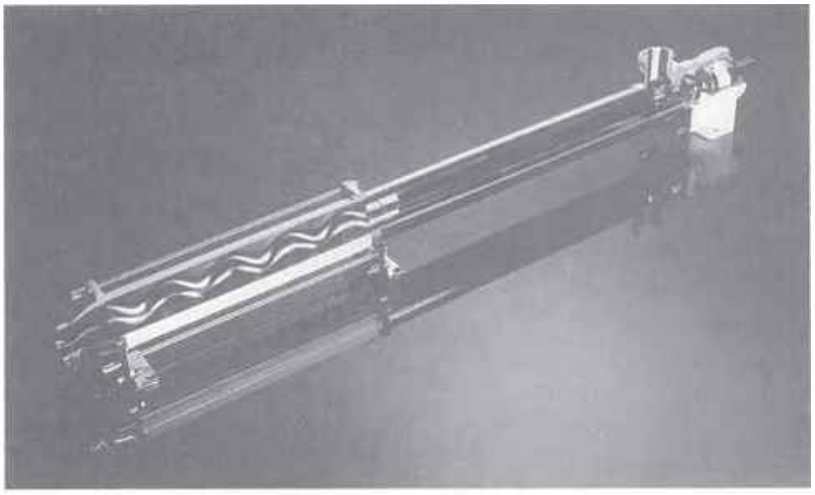


A

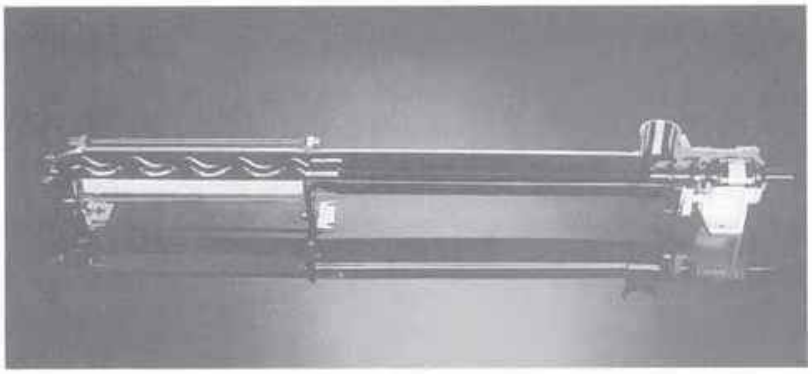


B

FIGURE 3A and B Flexible impeller pumps (ITT/Jabsco).



A



B

FIGURE 4A and B Hygienic progressive cavity pumps (Mono Pumps, Ltd.)

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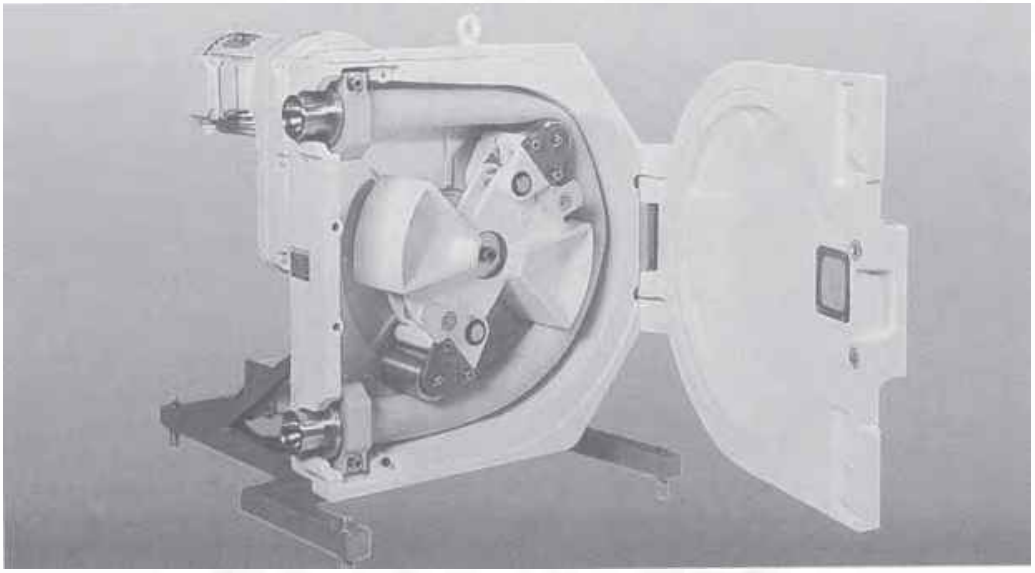


FIGURE 5 Hygienic peristaltic pump (Watson-Marlow, Ltd.)

Peristaltic Pumps (Figure 5)

ADVANTAGES Pumps are self-priming; they can run dry and are reversible; they can handle gaseous fluids and can pump abrasive material and solids in suspension; they have only one contact part with no seals.

DISADVANTAGES Tube failure means loss of product; at low rotational speeds, pulsation can be a problem.

Piston Pumps

ADVANTAGES Pumps may be used for very precise dosing; pumps can handle high pressures.

DISADVANTAGES Pumps are not reversible; they need valves to operate; and they generally have pulsing flow.

Diaphragm Pumps

ADVANTAGES Pumps can run dry for short periods; they are fairly easily cleaned; and they have no glands.

DISADVANTAGES Pumps are not reversible; they need valves to operate; and they have pulsing flow.

Mechanical Seals Pumps in the food and beverage industries are almost totally fitted with mechanical seals (see Figure 6). These seals must be simple in construction and designed in such a way as to facilitate cleaning. Cleaning in place (C.I.P.) is a frequent requirement, and mechanical seals for these pumps are specifically designed for this purpose.

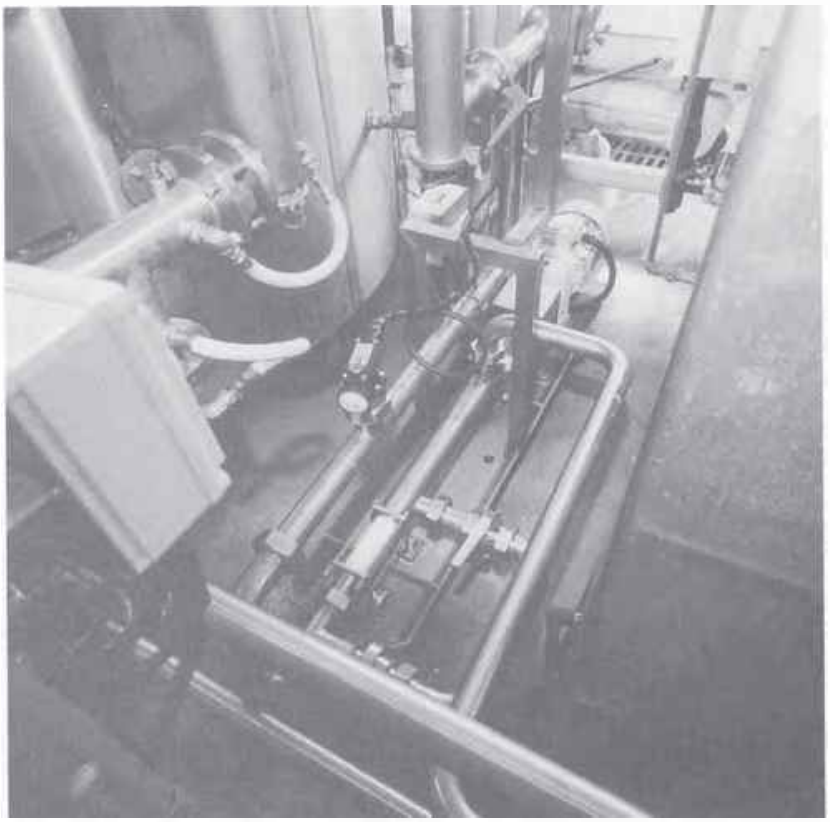
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FIGURE 6 A hygienic mechanical seal (Roplan, Ltd.)

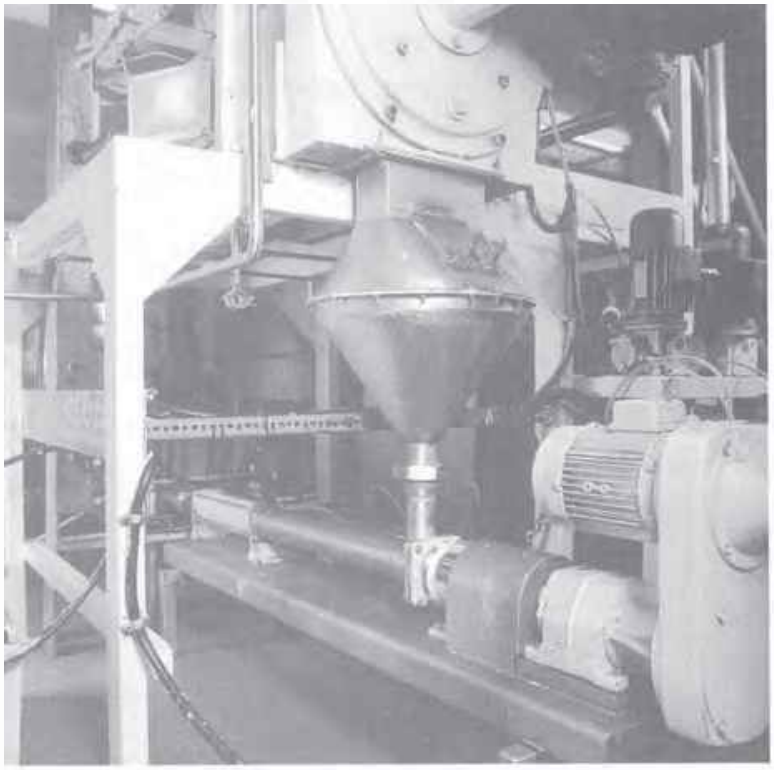
Pump Drives Standard electric motors, usually totally enclosed and fan-cooled, are used throughout the industry. A stainless steel casing is sometimes used to enclose the motor and simplify external cleanability. Pump units must be either grouted to the floor or mounted at least 2 in (50mm) above the floor to enable cleaning and eliminate any build-up of debris.

Figures 7A, B and C show typical installations of progressive cavity hygienic pumps.

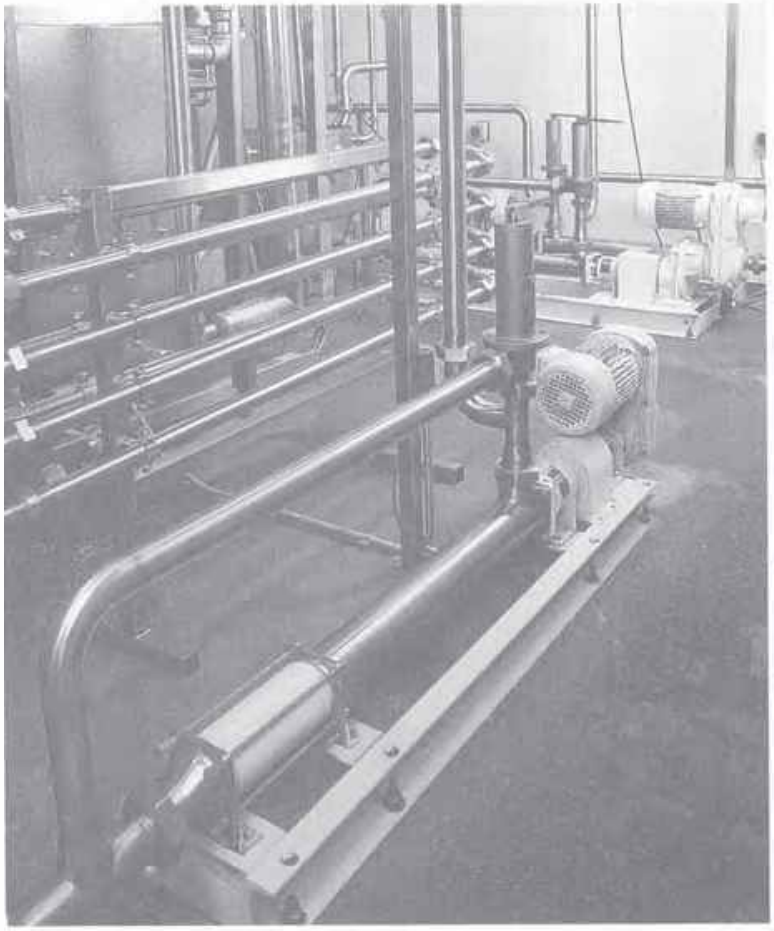


A

FIGURE 7A through C Typical installations of a progressive cavity hygienic pumps (Mono Pumps, Ltd.)



B



C

SECTION 12.11
MINING

W. D. HAENTJENS

It might be asked why pumps used in mining services receive a separate classification when the types involved do not differ in principle or in general appearance from those used in fresh-water service. The answer is the need for utmost reliability and an ability to withstand both corrosive and abrasive waters. Not all mine waters are corrosive, but almost all mine waters from active mines are abrasive because of the suspended solids from the mining operations. Much can be done to limit the amount of solids handled, but the solution generally requires a compromise based on economics. Removal of most coarse solids is usually justified but removal of fine solids is generally impractical. Thus, a heavy-duty pump has evolved that warrants the classification of mine pump.

PUMPING CONDITIONS AND PUMP TYPES

Pumping conditions in mining services can be determined from a consideration of the types of mines and the material being mined. For example, the broadest category would be a division between open-pit and underground or deep mines. Open pit mines seldom exceed 600 ft (183 m) in depth so the pumping heads generally are not much greater than this unless very long discharge lines are required. For many open-pit mines, the greatest pumping load does not result from groundwater but from rainfall. Unless a certain increase in water level can be tolerated at the bottom of the pit, the maximum rainfall rate and the drainage area involved will determine the required pumping capacity.

As mining progresses, the pumps generally must move with the operator. This suggests the use of barge-mounted vertical pumps. A practical cost-effective way of achieving dewatering under this scenario is the use of a pond system with barge-mounted vertical cantilevered shaft pumps. Barge mounting adds a great deal of flexibility. Simple barges, as in Figure 1, illustrate the simplicity of this mounting. Except for occasional greasing of the bearings, there is little maintenance, as these pumps do not require a stuffing box or

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FIGURE 1 A simple barge (Hazleton Pumps, Inc.)

mechanical seal. Almost any size can be barge mounted. Figure 2 shows one of three 1500 HP (1119 kW) vertical cantilevered shaft pumps that are installed on one barge. Each pump is rated at 14,000 gpm (3180 m^3/h).

Where there is significant rainfall, the flow rates are generally large and the combination of large capacities with moderate heads may suggest a double suction pump. The selection of a double suction pump may permit the use of a higher speed pump; however, the selection may not be based on hydraulic considerations alone. An examination of hydraulic design for a double suction pump may show it is ideal for a barge-mounted vertical pump, providing there is ample water level in the pit. On the other hand, minimum water levels may be required to facilitate mining operations. If these conditions exist, which commonly occurs, selection of a top inlet single-stage pump may be better. Top inlet barge mounted vertical pumps can pump the water level down without drawing the mud from the bottom. This is the conflict, as the desirable features of the double inlet pump are lost.

Comparative features can be readily seen from an examination of the net positive suction head (NPSH) requirements. High-capacity moderate head pumps have a specific speed (N_S) which may require a greater NPSH; that is, a greater submergence. This may alter the pump length and the ability to utilize a cantilevered shaft pump. Such pumps are ideal for barge mounting if the NPSH (required submergence) conditions exist.

Figures 3 and 4 are reproductions of charts published by the Hydraulic Institute. An example will highlight the available design choices. Calculations may show that some combinations of flow, head, and pump speed are not feasible. This is an important consideration before proceeding with design. Assuming we have a possible floating pump station with a pumping requirement of 7500 gpm (1700 m^3/h) and a calculated discharge head of 300 ft (91m), Table 1 shows the calculation of the required absolute suction conditions. Converting from absolute pressure to submergence from the water level to the impeller centerline, under standard atmospheric conditions, the table shows how much submergence is required for a given operating speed. The table shows that a single inlet pump at 1800 rpm is not practical, and a pump designed for 1200 rpm is barely acceptable. A double

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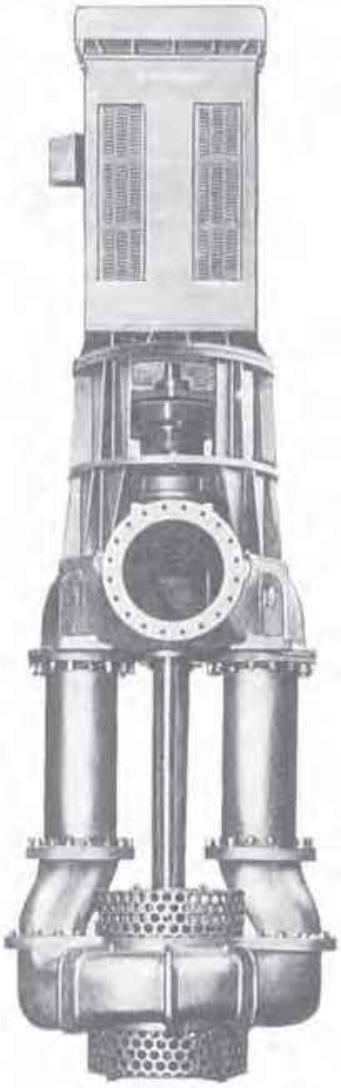


FIGURE 2 Vertical cantilevered shaft pumps used for barge mounting (Hazleton Pumps, Inc.)

suction pump at 1800 rpm is also barely acceptable if there is to be a cushion for low barometer days. A double suction pump at 1200 prm is fully acceptable. We are not locked into this selection, as a lower flow rate with more pumps may be fully acceptable at a higher speed.

The calculation for NPSH shows that the 1800 rpm double suction pump and the 1200 rpm single suction pump have about the same NPSH requirement. Although both are for the same hydraulic conditions, they are substantially different pumps. If the NPSH requirements can be met with a simple single inlet higher-speed barge mounted pump installation, a substantial reduction in cost and weight is possible.

Technically, other solutions are possible when the NPSH is marginal. If the NPSH requirement is only a few additional feet, it may be possible to lengthen the pump to increase the submerged depth. In comparing available NPSH and required NPSH, the altitude of the installation and the temperature of the liquid should be considered. Altitude correction can be simply applied as 1 ft (0.3 m) for each 1000 ft (305 m) of elevation in the calculation of NPSH available.

The development of large overhung vertical shaft pumps has been a boon to mining operations. Figure 5 shows the installation of four floats, each with a 400 HP (300 kW) top inlet cantilevered shaft pump. The type of mining operation frequently dictates the pump design and particularly the installation. Figure 6 shows float mounted pumps designed to

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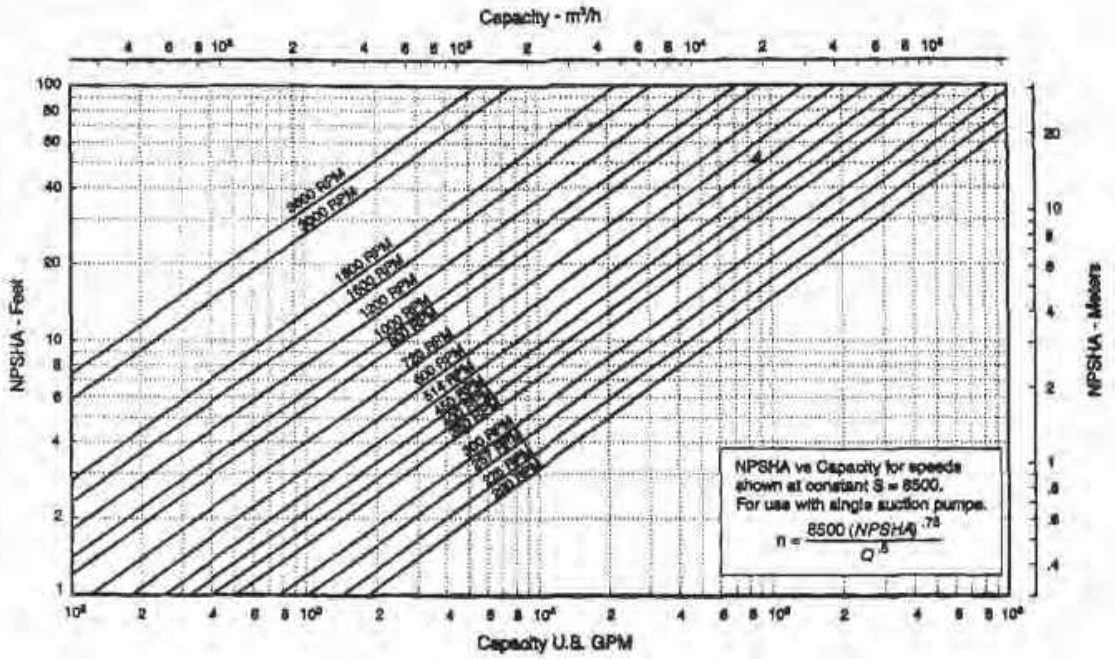


FIGURE 3 Recommended maximum operating speeds for single suction pumps (Hydraulic Institute ANSI/HI 2000 Edition Pump Standards, Reference 1)

skim the clean liquid in a settling pond. The float construction acts as a weir so only the cleanest liquid is returned to the plant. The float mounts 250 HP (185 kW) top inlet vertical cantilevered shaft pumps. An interesting feature is the hydraulic thrust balancing by the use of opposed flexible hose discharge lines. Note that these forces also exist in steel pipeline and must be adequately restrained.

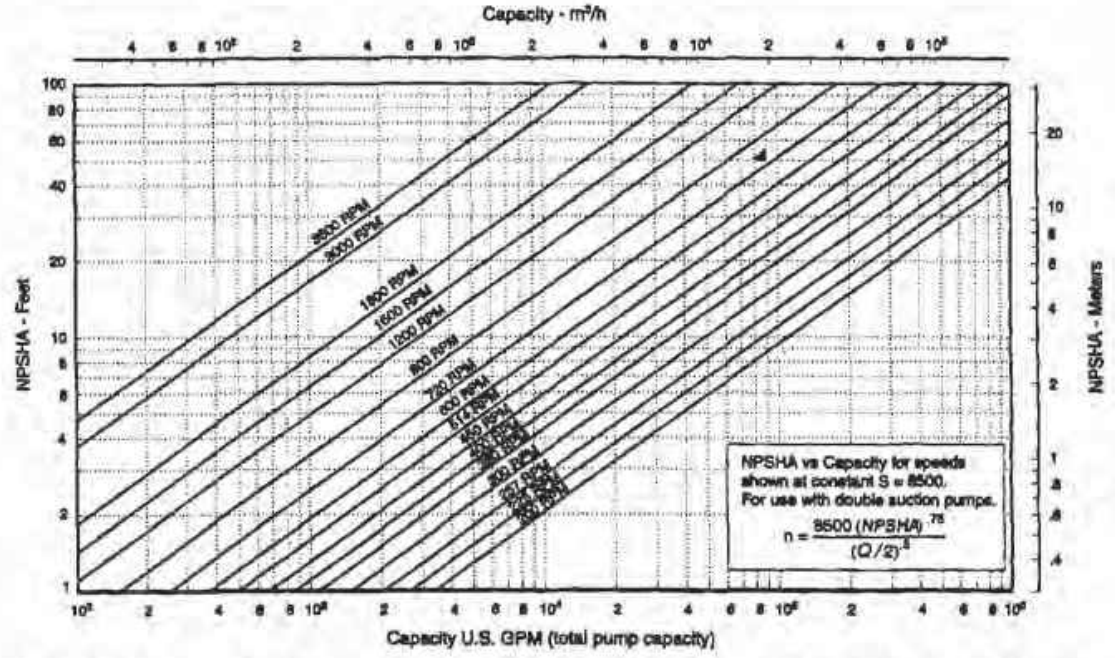


FIGURE 4 Recommended maximum operating speeds for double suction pumps (Hydraulic Institute ANSI/HI 2000 Edition Pump Standards, Reference 1)

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