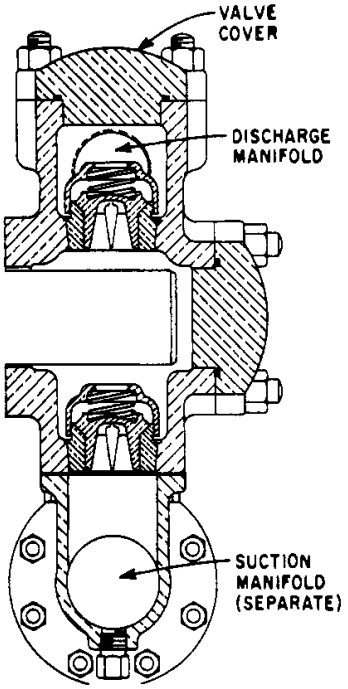


FIGURE 12 Separate suction with an integral discharge manifold (Gardner-Denver)

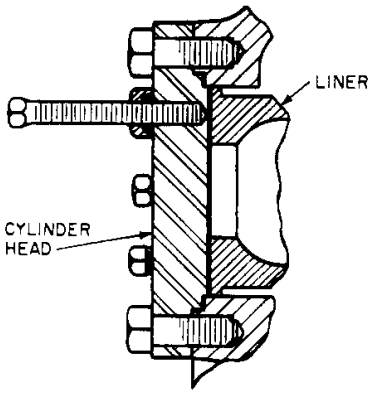


FIGURE 13 A plunger cover (FWI)

speeds, counterweights and flywheels are not used. Except for duplex pumps, the crankshaft always has an odd number of throws to obtain the best flow pulsation characteristics. The firing order in a revolution depends on the number of throws on the crankshaft (see Table 3).

The crankshaft main and rod-bearing surfaces are ground to a 16 rms surface finish or better. Large radii are used where journal diameters intersect with the adjacent cheeks to

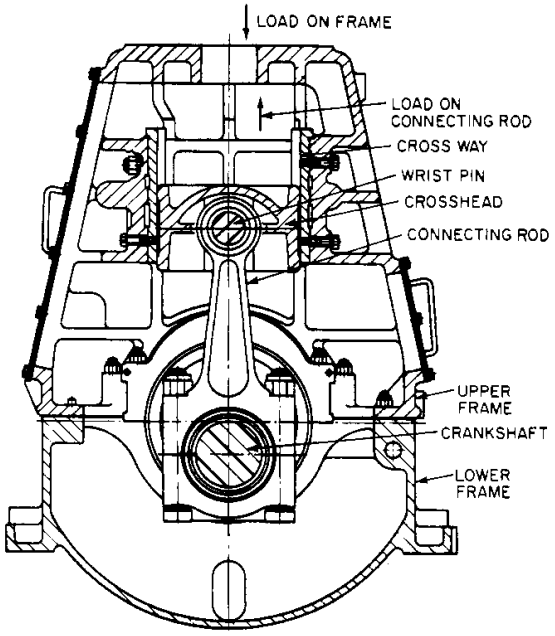


FIGURE 14 A vertical pump power end (Flowserve Corporation)

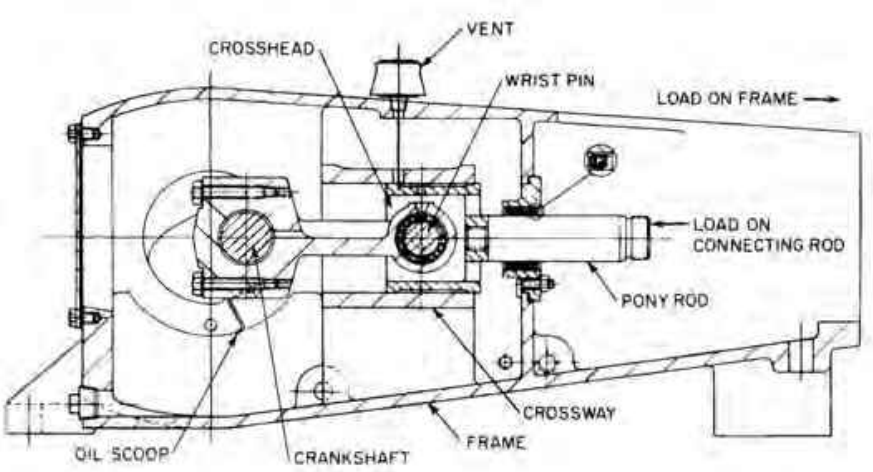


FIGURE 15 A horizontal pump power end (Flowserve Corporation)

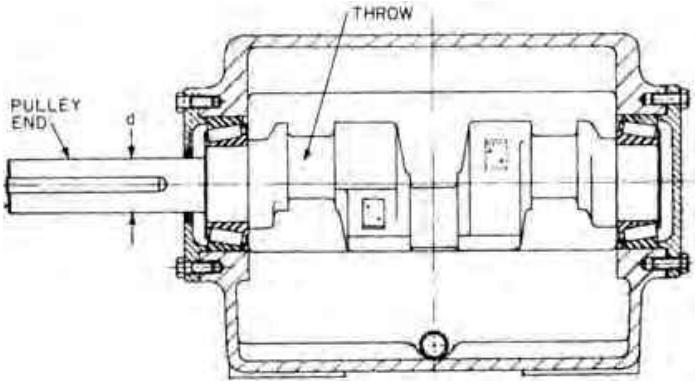


FIGURE 16 A cast crankshaft (Flowserve Corporation)

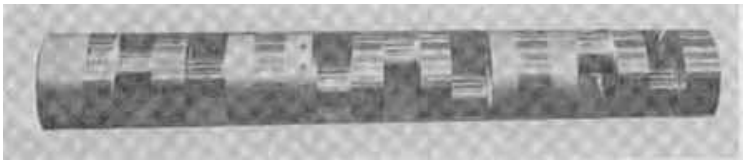


FIGURE 17 A crankshaft machined from a billet (Flowserve Corporation)



FIGURE 18 A cast crankshaft with integral gear (Continental-EMSCO)

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TABLE 3 Order of Pressure Build-up or Firing Order

Throw from pulley end	No. of plungers or pistons	Pressure build-up order								
		1	2	3	4	5	6	7	8	9
Duplex	2	1	2							
Triplex	3	1	3	2						
Quintaplex	5	1	3	5	2	4				
Septuplex	7	1	4	7	3	6	2	5		
Nonuplex	9	1	5	9	4	8	3	7	2	6

minimize stress concentrations in these fatigue-prone areas. Cross-drilling of the main to rod journals is often provided to furnish lubrication to the crankpin bearings.

Many horizontal pumps under 250 hp (186 kW) are driven by means of a sheave or pulley and belt arrangement. Increasing the diameter of the pulley for a given horsepower will increase the stress in the crankshaft. The maximum allowable crankshaft stress will often limit the maximum diameter of the pulley that can be used. Stress levels of 2000 to 3000 lb/in^2 (14 to 20.7 MPa) are common, where d is the pulley end diameter of the crankshaft in inches (mm) and n is the pump speed in revolutions per minute.

Connecting Rods and Eccentric Straps The connecting rods (see Figure 19) transfer the rotating force of the crank pin to an oscillating force on the wrist pin. Most connecting rods are split perpendicular to their centerline at the crankpin end for ease of assembly onto the crankshaft.

The cap and rod are aligned with close-tolerance bushings or body-bound bolts. The rods either are rifle-drilled or have cast passages to transfer oil from the wrist pin to the crank pin. A connecting rod with a tension load is made of forged steel, cast steel, or fabricated steel. Rods with a compressive loading are cast steel, nodular iron, or aluminum alloy. The connecting rod finish, where the bearings are mounted, is 32 to 63 rms.

The ratio of the distance between the centerlines of the wrist pin and crank pin bearings to half the length of the stroke is referred to as the L/R. The ratio directly affects the pressure pulsations, volumetric efficiency, size of the pulsation dampener, speed of liquid separation, acceleration head, moments of inertia forces, and size of the frame. Low L/R results in higher fluid pulsations. A high L/R reduces pulsations but may result in a large and uneconomical power frame. The common industrial L/R range is 4:1 to 6:1.

The eccentric strap (see Figure 20) performs the same function as a connecting rod, except that the former usually is not split. The eccentric strap is furnished with rolling element bearings, whereas connecting rods are furnished with sleeve bearings. Eccentric straps are applied to mud pumps and some slurry pumps, which are started against a full load without requiring a bypass line.

Wrist Pin Located in the crosshead, the wrist pin transforms the oscillating motion of the connecting rod to a reciprocating motion. The maximum stress in the wrist pin from deflection should not exceed 10,000 lb/in^2 (69 MPa):

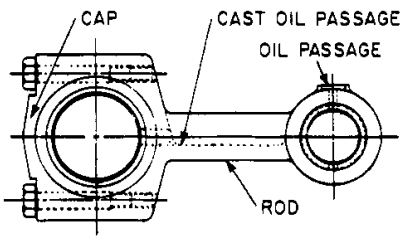


FIGURE 19 The connecting rod (Flowserve Corporation)

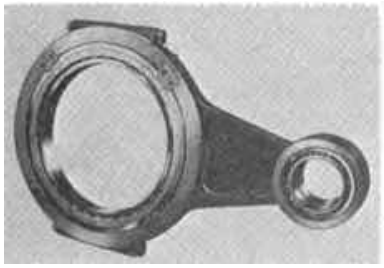


FIGURE 20 An eccentric strap (Gardner-Denver)

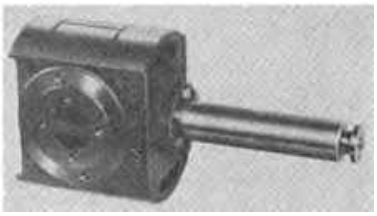


FIGURE 21 A crosshead (Gardner-Denver)

In USCS units,
$$S = \frac{PL \times \ell}{8 \times 0.098 d^3}$$

In SI units,
$$S = 1.25 \frac{PL \times \ell}{d^3}$$

where PL = plunger load, lb (N)
ℓ = length under load, in (mm)
d = diameter of pin, in (mm)

Large pins are hollow to reduce the oscillating mass and assembly weight. Depending on the design, pins can have a straight fit, taper fit, or loose fit in the crosshead. The pin is case-hardened and has a surface finish of 16 rms or better. When needle or roller bearings are used for the wrist pin bearing, the pin serves as the inner race.

Crosshead The crosshead (see Figure 21) moves in a reciprocating motion and transfers the plunger load to the wrist pin. The crosshead is designed to absorb the side or radial load from the plunger as the crosshead moves linearly on the crossway. The side load is approximately 25% of the plunger load. For cast-iron crossheads, the allowable bearing load is 80 to 125 lb/in^2 (551 to 862 kPa). Crossheads are grooved for oil lubrication and have a 63 rms surface finish. Crossheads are piston types (full round) or partial-contact types. The piston type should be open or vented to prevent air compression at the end of the stroke.

In vertical pumps, the pull rods, or side rods, go through the crosshead so that the crosshead is under compression when the plunger load is applied (refer to Figure 14).

Crosshead Guides (Crossways) The crossway (see Figure 22) is the surface on which the crosshead reciprocates. In horizontal pumps, it is cast integral with the frame (refer to Figure 15). In large frames, it is usually replaceable and is shimmed to effect proper running clearances (refer to Figures 14 and 22). The crossway finish is 63 rms.

Pony Rod (intermediate Rod or Extension Rod) The pony rod is an extension of the crosshead on horizontal pumps (refer to Figures 21 and 22). It is screwed or bolted to the

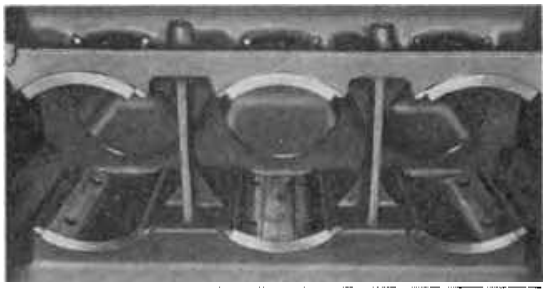


FIGURE 22 Crossways (Gardner-Denver)

crosshead and extends through the frame (refer to Figure 15). A seal on the frame and against the pony rod prevents oil from leaking out of the frame. A baffle is attached to the plunger end of the pony rod to prevent stuffing box leakage from entering the frame.

Pull Rod (Side Rod) In vertical pumps, two pull rods are connected to the crosshead. The rods are secured by a shoulder and nut so that the cast iron is in compression when the plunger load (rod load) is applied. The rods extend out through the top of the frame and fasten to a yoke (upper crosshead). The plunger is attached to the middle of the yoke with an aligning feature for the plunger (refer to Figure 2). The nuts fastening the pull rod to the yoke and crosshead must be torqued to a high enough level to prevent the bolted joint from seeing cyclic loading.

Bearings Both sleeve and rolling element bearings are used in power pumps. Some power ends use all sleeve, others use all rolling element, and others use a combination of both.

SLEEVE BEARINGS

When properly installed and lubricated, sleeve bearings are considered to have an infinite life. They are inexpensive and can usually be replaced without special tools and with minimum pump disassembly. The sleeve bearing is designed to operate within a certain speed range, and too high or too low a speed will upset the oil film lubrication. With standard sleeve-bearing designs, pumps cannot be operated at speeds below 20 rpm with the plunger fully loaded using standard lubrication. Below this speed, the oil film in the wrist pin bearing starts to break down and the increased friction causes excessive heat and leads to bearing failures. Recent developments of improved bearing materials, along with special grooving and other aids to lubrication, now allow for fully loaded pump operations at speeds as low as 10 rpm. The finish on the sleeve bearing is 16 rms. Clearances are approximately 0.001 in per in (0.001 mm per mm) of the diameter of the bearing.

Wrist Pin Bearings These bearings are normally the highest loaded bearings in the pump. They can only perform an oscillating motion and therefore cannot develop the dynamic oil film common to fully rotational bearings. In single-acting pumps with low suction pressure, adequate reversals in loading take place on the bearing to permit replenishment of the oil film. High suction pressure in a horizontal pump increases the reverse loading. In vertical pumps, high suction pressure can produce a condition of no reverse loading, and in this case, special bearing materials, grooving, and pump derating may be required. Increasing oil pressure by itself will not solve this problem. Allowable projected area loading is 1200 to 1500 lb/in^2 (8.27 to 10.3 MPa) with bearing bronze.

Crank Pin Bearings The crank pin bearing is a rotating split bearing and develops a better oil film than the wrist pin bearing. It is clamped between the connecting rod and cap. The bearing is a bronze-backed babbitt metal or a tri-metal automotive type bearing. The allowable projected area load is 1200 to 1600 lb/in^2 (8.27 to 11.08 MPa).

Main Bearings The main bearings absorb the plunger load and gear load. The total plunger load varies during each revolution of the crankshaft. The triplex main bearings receive the greatest variations in loading because the crankshaft has the greatest relative span between bearings. Sleeve bearings are flanged to locate them in an axial position, and the flange absorbs residual axial thrust. In long-stroke vertical pumps, a main bearing can be found between every connecting rod. Split bearings are a bronze-backed babbitt metal with an allowable loaded area of 750 lb/in^2 (5.1 MPa).

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ROLLING ELEMENT BEARINGS

A pump with rolling element bearings can be started under a full plunger load without a bypass line. Rolling element bearings enable the pump to operate continuously at low speeds with full plunger loads. They are selected for 30,000 to 50,000 hours of L_{10h} life. Slurry pipeline applications may require 100,000 hours. Eccentric straps are used in place of connecting rods.

Wrist Pin Bearings These are needle or roller bearings. The outer race is a tight fit into the strap, and the wrist pin is used as the inner race. This reduces the size of the bearing and strap. The wrist pin is held in the crosshead with a taper or keeper plate (see Figure 23).

Main Bearing The main bearings used in conjunction with sleeve wrist pin bearings and sleeve crank pin bearings are usually tapered roller bearings, as is the case with a horizontal triplex plunger pump. The main bearings are usually mounted directly into the frame (see Figure 24).

The main bearings used with full rolling element bearing designs are self-aligning spherical roller bearings. These bearings compensate for axial and radial movements of the crankshaft and are usually mounted in bearing holders, which in turn are mounted on the frame (refer to Figure 23). This type of design is mostly used in mud pumps and in some slurry pumps.

LUBRICATION

Non-detergent SAE 30 to 50 oil is used for bearing lubrication. In splash lubrication, the cheeks of the crankshaft or oil scoops (refer to Figure 15) throw oil by centrifugal force against the frame wall. This oil is then distributed by gravity to the crosshead, wrist pin, and crank pin. At low speeds, there is not enough centrifugal force for proper oil distribution, so a partial force feed is then employed to maintain proper lubrication. Force-feed lubrication requires 0.5 to 1 gpm (1.9 to 3.8 l/m) per bearing with an oil pressure of 28 to 40 lb/in^2 (1.7 to 2.8 bar).

APPLICATIONS

Power pumps are used in a variety of applications where high-pressure and low-flow requirements exist. They are also widely used when a high-volumetric efficiency is

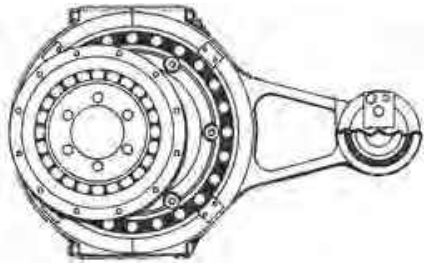


FIGURE 23 The main bearing mounted in a bearing holder (Gardner-Denver)

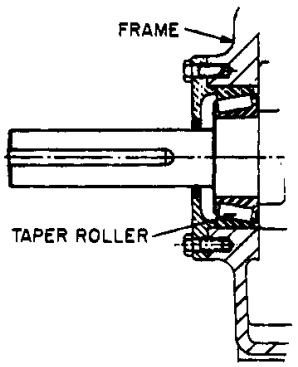


FIGURE 24 The main bearing mounted directly in the frame (Flowserve Corporation)

required. For these applications, other types of pumps have design or operating deficiencies that make their lower initial cost or lower maintenance costs less attractive.

Some of the common applications for power pumps include the following:

Ammonia service	Liquid petroleum gas
Carbamate service	Liquid pipeline
Chemicals	Power oil
Crude-oil pipeline	Power press
Cryogenic service	Salt-water injection for water flood
Fertilizer plants	Slurry pipeline (up to 70% by weight)
High-pressure water cutting	Slush ash service
Hydro forming	Steel mill descaling
Hydrostatic testing	Water blast service

The following applications may require power pumps with special modifications, and the exact requirements should be reviewed in detail with the pump supplier:

Cryogenic service	Low-speed operation
Highly compressible liquids	Slurry pipeline
Liquids over 250°F (121°C)	Special fluid-end materials
Liquids with entrained gas	Viscosities over 500 SSU

Duty Service To make an economical pump selection, the type of full-load service that you need should be considered. For instance, a small, lighter weight pump operating at higher speeds may be suitable for an intermittent duty application such as hydrostatic testing. The different types of duty services are as follows:

- *Continuous*: 8 to 24 hours per day
- *Light*: 3 to 8 hours per day
- *Intermittent*: Up to 3 hours per day
- *Cyclic*: 30 seconds loaded out of every 3 minutes

RELIEF VALVES

Relief valves should be rated for the full flow of the pump. The *set pressure* or *cracking pressure* must be sufficiently above the operating pressure to prevent normal pressure pulsations from activating the relief valve. The following values can be used as guidelines, but the final selection should be made only in agreement with the pump supplier and system designer:

- Duplex double-acting: 25%
- Triplex and above: 10%

Bypass Relief Starting a power pump under a full load requires a high starting torque. Also, a pump with sleeve-bearing construction when starting under a full load may not have adequate bearing lubrication to reach the operating speed.

The starting torque on the driver and plunger load on the bearings may be reduced by performing one of two steps:

- Install a bypass line from the discharge back to the source or to a drain (see Figure 25). This line is located between the pump discharge and a check valve before the discharge piping system. The bypass valve is operated manually or automatically. The bypass line

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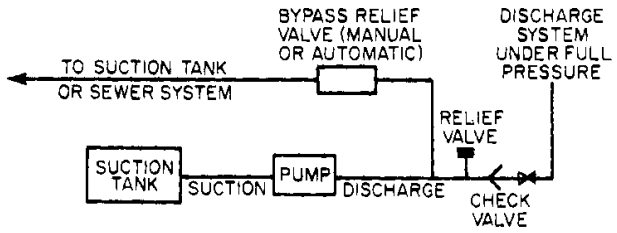


FIGURE 25 A bypass relief system (Flowserve Corporation)

reduces the starting torque from the mechanical losses and the liquid inertia of the suction and bypass systems. After the pump is up to speed, the bypass is closed and the pump goes into normal operation.

- Use suction valve unloaders. The unloader is a mechanism that mechanically holds the suction valves off of their seats before and during startup, which stops the pumping action. Only mechanical losses must be overcome. After the pump is up to speed, the suction valves are closed in unison. In large-flow pumps, an electronic distributor is used to synchronize actuation of the suction valves with the plunger pressure build-up order (or firing order).

PULSATION DAMPENERS

The performance of a power pump, or its interaction with the suction and discharge piping systems, can often be improved by adding suction and discharge dampeners. Dampeners are used to reduce suction and discharge pressure pulsations. A properly sized and located suction dampener can reduce the system pulsations to an equivalent pipe length of 5 to 15 pipe diameters. If the dampener is the flow-through type, it should be located on the same side as the pipe, not at the dead end of the manifold.

The most commonly used dampeners are the gas-bladder types. These units are normally charged with nitrogen gas at 50 to 66% of the system pressure. Although these units are relatively inexpensive, they can attenuate up to 90% of the pressure pulsations, below 50 Hz, in the interconnecting piping systems.

SECTION 3.3
STEAM PUMPS

ROBERT M. FREEBOROUGH

BASIC THEORY

A reciprocating positive displacement pump is one in which a plunger or piston displaces a given volume of fluid for each stroke. The basic principle of a reciprocating pump is that a solid will displace an equal volume of liquid. For example, when an ice cube is dropped into a glass of water, the volume of water that spills out of the glass is equal to the submerged volume of the ice cube.

In Figure 1, a cylindrical solid, a plunger, has displaced its volume from the large container to the small container. The volume of the displaced fluid (B) is equal to the plunger volume (A). The volume of the displaced fluid equals the product of the cross-sectional area of the plunger and the depth of submergence.

All reciprocating pumps have a fluid-handling portion, commonly called the *liquid end,* that has

1. A displacing solid called a *plunger* or *piston*
2. A container to hold the liquid, called the *liquid cylinder*
3. A suction check valve to admit fluid from the suction pipe into the liquid cylinder
4. A discharge check valve to admit flow from the liquid cylinder into the discharge pipe
5. Packing to seal the joint between the plunger and the liquid cylinder tightly to prevent liquid from leaking out of the cylinder and air from leaking into the cylinder

These basic components are identified on the rudimentary liquid cylinder illustrated in Figure 2. To pump the liquid through the liquid end, the plunger must be moved. When the plunger is moved out of the liquid cylinder as shown in Figure 2, the pressure of the fluid in the cylinder is reduced. When the pressure becomes less than that in the suction pipe, the suction check valve opens and liquid flows into the cylinder to fill the volume being

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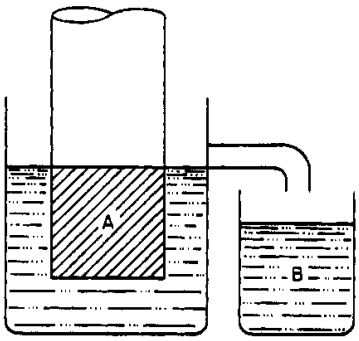


FIGURE 1 The volume of liquid displaced by a solid equals the volume of the solid

vacated by withdrawal of the plunger. During this phase of operation, the discharge check valve is held closed by the higher pressure in the discharged pipe. This portion of the pumping action of a reciprocating positive displacement pump is called the *suction stroke.*

The withdrawal movement must be stopped before the end of the plunger gets to the packing. The plunger movement is then reversed, and the *discharge stroke* portion of the pumping action is started, as illustrated in Figure 3.

Movement of the plunger into the cylinder causes an increase in pressure of the liquid contained therein. This pressure immediately becomes higher than suction pipe pressure and causes the suction valve to close. With further plunger movement, the liquid pressure continues to rise. When the liquid pressure in the cylinder reaches that in the discharge pipe, the discharge check valve is forced open and liquid flows into the discharge pipe. The volume forced into the discharge pipe is equal to the plunger displacement less very small losses. The plunger displacement is the product of its cross-sectional area and the length of stroke. The plunger must be stopped before it hits the bottom of the cylinder. The motion is then reversed, and the plunger again goes on suction stroke as previously described.

The pumping cycle just described is that of a *single-acting* reciprocating pump. It is called single-acting because it makes only one suction stroke and only one discharge stroke in one reciprocating cycle.

Many reciprocating pumps are *double-acting;* that is, they make two suction and two discharge strokes for one complete reciprocating cycle (Figure 4). Most double-acting pumps use as the displacing solid a piston that is sealed to a bore in the liquid cylinder or

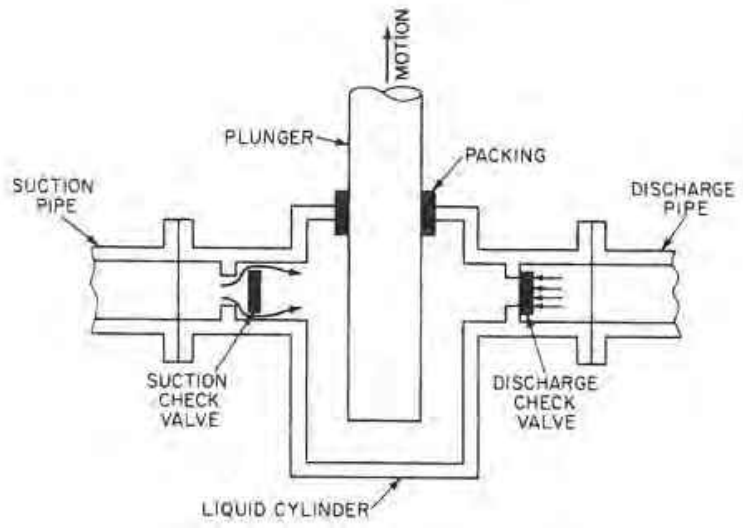


FIGURE 2 Schematic of the liquid end of reciprocating pump during the suction stroke

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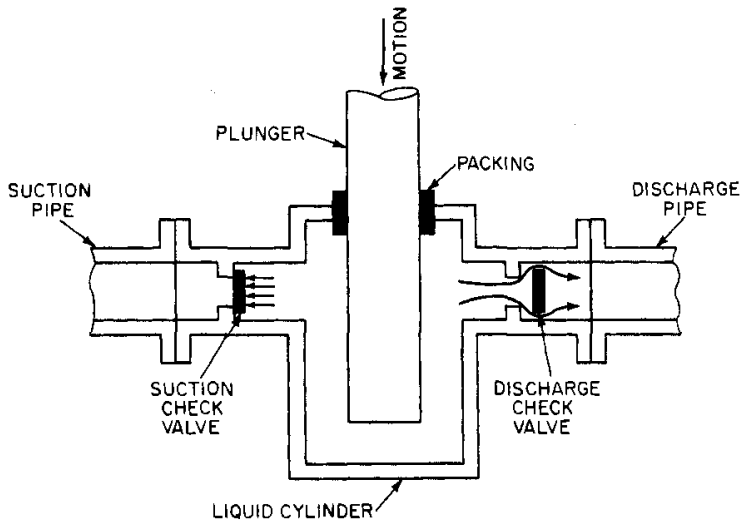


FIGURE 3 Schematic of the liquid end of a reciprocating pump during the discharge stroke

to a liquid cylinder liner by pistons with packing. It has two suction and two discharge valves, one of each on each side of the piston. The piston is moved by a piston rod. The piston rod packing prevents liquid from leaking out of the cylinder. When the piston rod and piston are moved in the direction shown, the right side of the piston is on a *discharge stroke* and the left side of the piston is simultaneously on a *suction stroke.* The piston packing must seal tightly to the cylinder liner to prevent leakage of liquid from the high-pressure right side to the low-pressure left side.

The piston must be stopped before it hits the right side of the cylinder. The motion of the piston is then reversed so the left side of the piston begins its discharge stroke and the right side begins its suction stroke.

A reciprocating pump is not complete with a liquid end only; it must also have a driving mechanism to provide motion and force to the plunger or piston. The two most common driving mechanisms are a reciprocating steam engine and a crank-and-throw device. Those pumps using the steam engine are called *direct-acting steam pumps.* Those pumps using the crank-and-throw device are called *power pumps.* Power pumps must be connected to an external rotating driving force, such as an electric motor, steam turbine, or internal combustion engine.

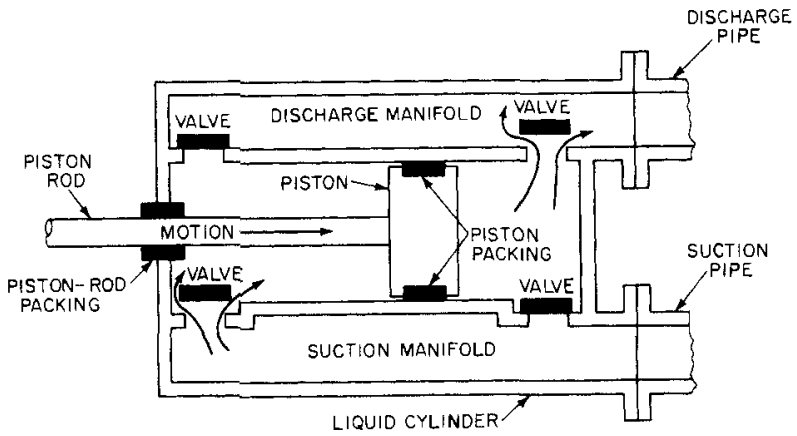


FIGURE 4 Schematic of a double-acting liquid end

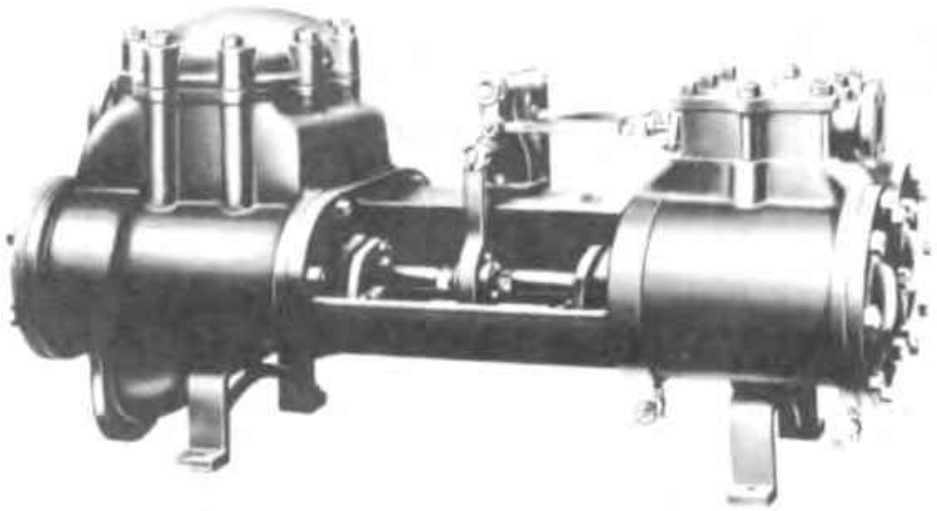


FIGURE 5 Duplex steam pump (Flowserve Corporation)

DIRECT-ACTING STEAM PUMPS

Direct-acting steam pumps are mainly classified by the number of working combinations of cylinders. For example, a duplex pump (Figure 5) has two steam and two liquid cylinders mounted side by side, and a simplex pump (Figure 6) has one steam and one liquid cylinder.

Additionally, simplex and duplex pumps may be further defined by (1) cylinder arrangement, whether horizontal or vertical; (2) number of steam expansions in the power end; (3) liquid end arrangement, whether piston or plunger; and (4) valve arrangement, that is, cap and valve plate, side pot, turret type, and so on.

Although this section will refer to steam as the driving medium, compressed gases such as air or natural gas can be used to drive a steam pump. These gases should have oil or mist added to them prior to entering the pump to prevent wear of the steam end parts.

STEAM END CONSTRUCTION AND OPERATION

The driving mechanism, or *steam end,* of a direct-acting steam pump includes the following components, as illustrated in Figure 7:

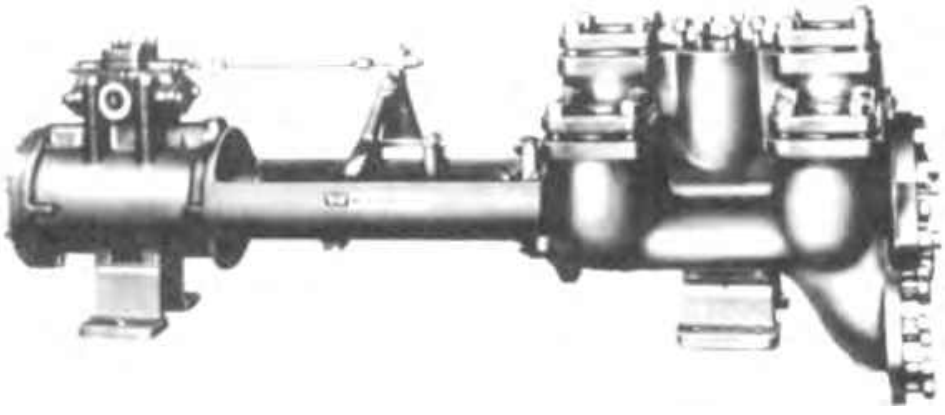


FIGURE 6 Simplex steam pump (Flowserve Corporation)

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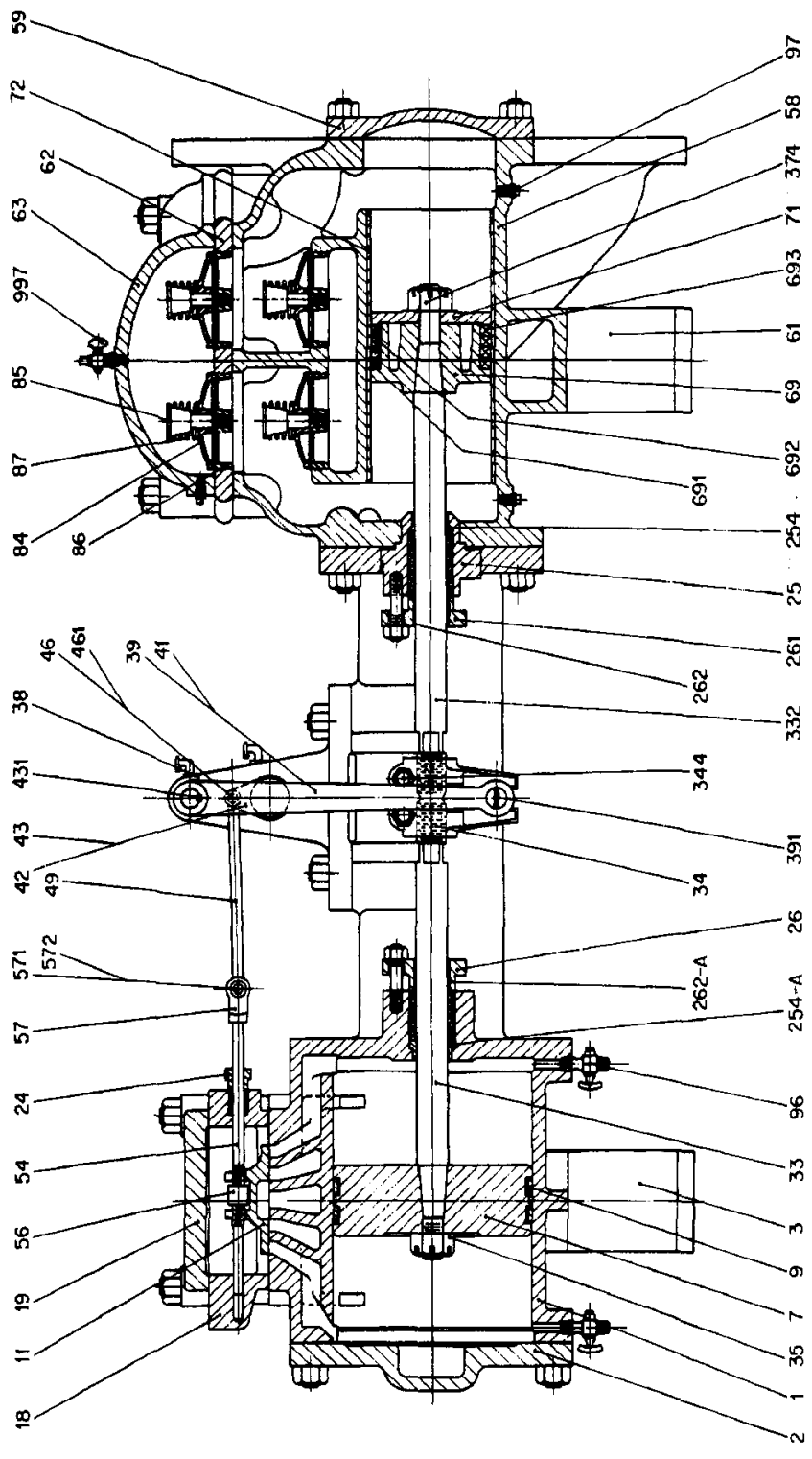


FIGURE 7 Typical section of duplex steam pump: (1) Steam cylinder with cradle, (2) steam cylinder head, (3) steam cylinder foot, (7) steam piston, (9) steam piston rings, (11) slide valve, (18) steam chest, (19) steam chest cover, (24) valve rod stuffing box gland, (25) piston rod stuffing box, liquid, (26) piston rod stuffing box gland, steam, (33) steam piston rod, (34) steam piston spool, (35) steam piston nut, (38) cross stand, (39) long lever, (41) short lever, (42) upper rock shaft, long crank, (43) lower rock shaft, short crank, (46) crank pin, (49) valve rod link, (54) valve rod, (56) valve rod nut, (57) valve rod head, (58) liquid cylinder, (59) liquid cylinder head, (61) liquid cylinder foot, (62) valve plate, (63) force chamber, (69) liquid piston body, (71) liquid piston follower, (72) liquid cylinder lining, (84) metal valve, (85) valve guard, (86) valve seat, (87) valve spring, (96) drain valve for steam end, (97) drain plug for liquid end, (254) liquid piston rod stuffing box bushing, (332) liquid piston rod, (344) piston rod spool bolt, (374) liquid piston rod nut, (391) lever pin, (431) lever key, (461) crank pin nut, (571) valve rod head pin, (572) valve rod head pin nut, (691) liquid piston snap rings, (692) liquid piston bull rings, (693) liquid piston fibrous packing rings, (997) air cock, (251) liquid piston rod stuffing box, (254A) steam piston rod stuffing box bushing, (261) piston rod stuffing box gland, liquid, (262) piston rod stuffing box gland lining, liquid, (262A) piston rod stuffing box gland lining, steam (Flowserve Corporation)

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1. One or more steam cylinders with suitable steam inlet and exhaust connections
2. Steam piston with rings
3. Steam piston rods directly connected to liquid piston rods
4. Steam valves that direct steam into and exhaust steam from the steam cylinder
5. A steam valve actuating mechanism that moves the steam valve in proper sequence to produce reciprocating motion

The operation of a steam pump is quite simple. The motion of the piston is obtained by admitting steam of sufficient pressure to one side of the steam piston while simultaneously exhausting steam from the other side of the piston. There is very little expansion of the steam because it is admitted at a constant rate throughout the stroke. The moving parts, that is, the steam piston, the liquid piston, and the piston rod or rods, are cushioned and brought to rest by exhaust steam trapped in the end of the steam cylinder at the end of each stroke. After a brief pause at the end of the stroke, steam is admitted to the opposite side of the piston and the pump strokes in the opposite direction.

Steam Valves Because the steam valve and its actuating mechanism control the reciprocating motion, any detailed description of the construction of the direct-acting steam pump should rightfully begin with a discussion of steam valve types, operation, and construction.

Duplex Steam Valves The steam valves in a duplex steam pump are less complicated than those in a simplex pump and will be described first. As previously stated, the duplex steam pump can be considered as two simplex pumps arranged side by side and combined to operate as a single unit. The piston rod of one pump, in making its stroke, actuates the steam valve and thereby controls the admission or exhaust of steam in the second pump. A valve gear cross stand assembly is shown in Figure 8. The wishbone-shaped piston rod lever of one side is connected by a shaft to the valve rod crank of the opposite side. The steam valve is connected to the valve rod crank by the steam valve rod and steam valve link. Through this assembly, the piston rod of one side moves the steam valve of the opposite side in the same direction. When the first pump has completed its stroke, it must pause until its own steam valve is actuated by the movement of the second pump before it can make its return stroke. Because one or the other steam cylinder port is always open, there is no "dead center" condition; hence, the pump is always ready to start when steam

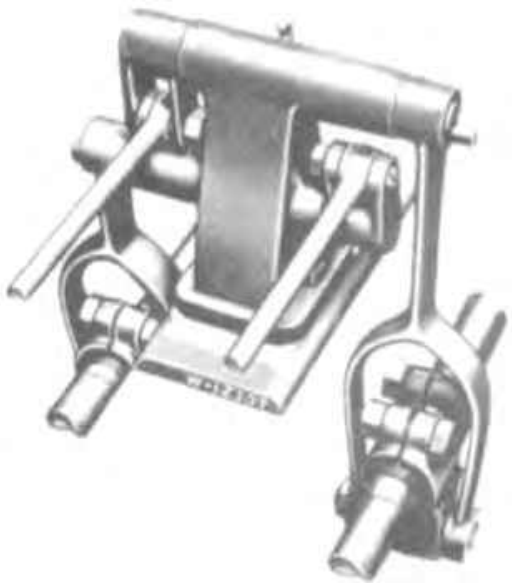


FIGURE 8 Steam valve actuating mechanism or a duplex pump (Flowserve Corporation)

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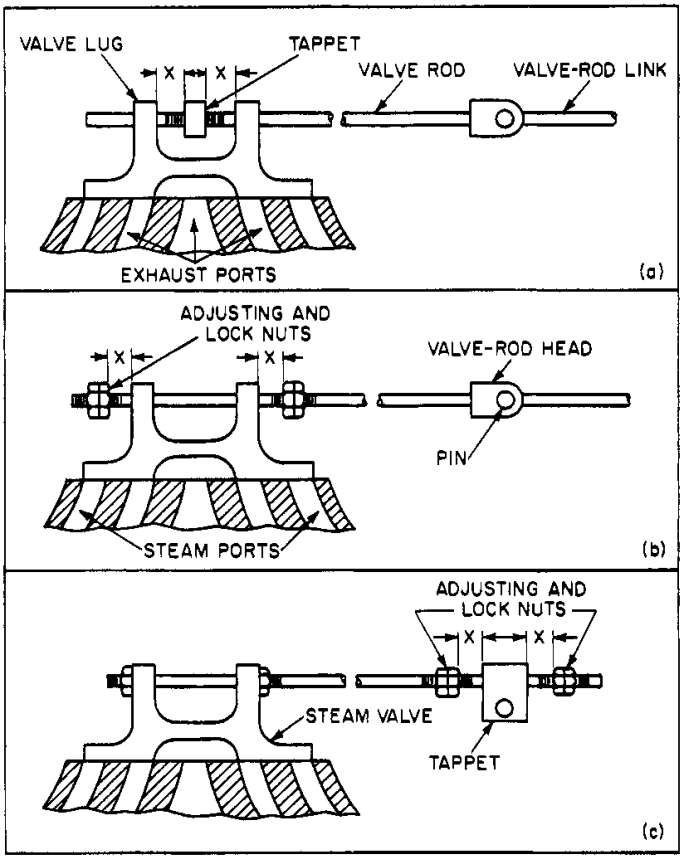


FIGURE 9A through C Duplex pump steam valve lost-motion arrangements

is admitted to the steam chest. The movements of both pistons are synchronized to produce a well-regulated flow of liquid free of excessive pulsations and interruptions.

Flat Slide Steam Valves Steam enters the pump from the steam pipe into the steam chest on top of the steam cylinder. Exhaust steam leaves the pump through the center port of five ports, as shown in Figure 9. Most duplex pumps use a flat slide valve that is held against its seat by steam pressure acting upon its entire top area; this is called an *unbalanced* valve. The flat or D valve, as it is often called, is satisfactory for steam pressures up to approximately 250 lb/in^2 (17 bar*) and has a reasonable service life, particularly where steam end lubrication is permissible. On large pumps, the force required to move an unbalanced valve is considerable, and so a balanced piston valve, which will be discussed later, is used.

The slide valve shown in Figure 9 is positioned on dead center over the five valve ports. A movement of the valve to the right uncovers the left-side steam port and the right-side exhaust port, which is connected through the slide valve to the center exhaust port. The main steam piston will be moved from left to right by the admitted steam. Movement of the slide valve from dead center to the left would, of course, cause opposite movement of the steam piston.

The steam valves of a duplex pump are mechanically operated, and their movements are dependent upon the motion of the piston rod and the linkage of the valve gear. In order to ensure that one piston will always be in motion when the other piston is reversing at the end of its stroke, lost motion is introduced into the valve gear. Lost motion is a means by which the piston can move during a portion of its stroke without moving the steam valve. Several lost motion arrangements are shown in Figure 9.

*1 bar = 10^5 Pa.

If the steam valves are out of adjustment, the pump will have a tendency not to operate through its designed stroke. Increasing the lost motion lengthens the stroke; if this is excessive, the piston will strike the cylinder head. Reducing the lost motion shortens the stroke; if this is excessive, the pump will short-stroke, with a resulting loss in capacity.

The first step in adjusting the valves is to have both steam pistons in a central position in the cylinder. To accomplish this, the piston is moved toward the steam end until the piston strikes the cylinder head. With the piston rod in this position, a mark is made on the rod flush with the steam end stuffing box gland. Next, the piston rod is moved toward the liquid end until the piston strikes, and then another mark is placed on the rod halfway between the first mark and the steam end stuffing box gland. After this, the piston rod is returned toward the steam end until the second mark is flush with the stuffing box gland. The steam piston is now in central position. This procedure is repeated for the opposite piston rod assembly.

The next step is to see that both steam valves are in a central position with equal amounts of lost motion on each side, indicated by distance X in Figure 9.

Most small steam pumps are fitted with a fixed amount of lost motion, as shown in Figure 9a. With the slide valve centered over the valve ports, a properly adjusted pump will have the tappet exactly centered in the space between the valve lugs. The lost motion (X) on each side of the tappet will be equal.

Larger pumps are fitted with adjustable lost motions, such as are shown in Figure 9b. The amount of lost motion (X) can be changed by moving the locknuts. Manufacturers provide specific instructions for setting proper lost motion. However, one rule of thumb is to allow half the width of the steam port on each side for lost motion. A method to provide equality of lost motion is to move the valve each way until it strikes the nut and then note if both port openings are the same.

In some cases, it is desirable to be able to adjust the steam valves while the pump is in motion. With the arrangements previously mentioned, this cannot be done because the steam chest head must be removed. In a pump equipped with a lost-motion mechanism such as that shown in Figure 9c, all adjustments are external and can therefore be made while the pump is in operation.

Balanced Piston Steam Valve Thebalanced piston steam valve (Figure 10) is used on duplex steam pumps when the slide-type valve cannot be used because of size. The balanced piston valve can also be used without lubrication at pressure above 250 lb/in^2 (17 bar) and temperatures above 500°F (260°C). At higher pressures, wire drawing or steam cutting can occur as the piston slowly crosses the steam ports. To prevent wear and permanent damage to the steam chest and piston, piston rings are used on the steam valve and a steam chest liner is pressed into the steam chest to protect it.

Cushion Valves Steam cushion valves are usually furnished on larger pumps to act as an added control to prevent the steam piston from striking the cylinder heads when the pump operates at high speeds. As previously shown, the steam end has five ports, the outside ports are for steam admission, and the inside ports are for steam exhaust. As the steam piston approaches the end of the cylinder, it covers the exhaust port, trapping a

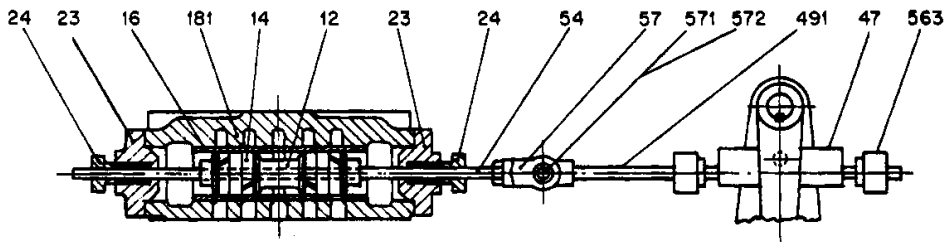


FIGURE 10 Balanced piston steam valve: (12) piston valve, (181) steam chest, (23) valve rod stuffing box, (24) valve rod stuffing box gland, (54) valve rod, complete, (57) valve rod head, (14) piston valve ring, (16) piston valve lining, (47) lost-motion block tappet, (491) valve rod link, (563) valve rod collar (Flowserve Corporation)

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volume of steam in the end of the cylinder. This steam acts as a cushion and prevents the piston from striking the cylinder head. The cushion valve is simply a bypass valve between the steam and exhaust ports; by opening or closing this valve, the amount of cushion steam can be controlled.

If the pump is running at low speed or working under heavy load, the cushion valve should be opened as much as possible without allowing the piston to strike the cylinder head. If the pump is running at high speed or working under light load, the cushion valve should be closed. The amount of steam cushion and, consequently, the length of stroke can be properly regulated for different operating conditions by the adjustment of this valve.

Simplex Steam Valves The simplex pump steam valve is steam-operated, not mechanically operated as duplex steam valves are. The reason for this is that the piston rod assembly must operate its own steam valve. Consequently the travel of the valve cannot be controlled directly by means of the piston rod motion. Instead, the piston rod operates a pilot valve by means of a linkage similar to that used with a duplex pump. This controls the flow of steam to each end of the main valve, shuttling the steam back and forth. The arrangement illustrated in Figure 11 is one of the designs available to produce this motion.

With the pilot valve in the position shown in Figure 11, steam from the live steam space flows through the pilot valve steam port into the steam space at the left-hand end of the main valve (balanced piston type). Simultaneously, the D section of the pilot valve connects the steam space at the right-hand end of the main valve with the exhaust port, thereby releasing the trapped steam. The main valve has moved completely across to the right end of the chest. The main valve in this position permits steam to flow from the chest to the left steam cylinder port and, at the same time, connects the right steam cylinder port with the exhaust port.

The steam piston now moves to the right, and after the lost motion is taken up in the valve gear, the pilot valve moves to the left. In this position, the cycle previously described now takes place at the opposite end of the steam chest. Because the main valve is steam-operated, it can be in only two positions, either at the left-hand or at the right-hand end of the chest. Hence, it is impossible to have it at dead center. In other words, steam can always flow either to one side or to the other of the steam piston, regardless of the position of the steam piston.

For the valve to operate smoothly and quietly, an arrangement must be provided to create a cushioning effect on the valve travel. The steam piston, as it approaches the end of its travel, cuts off the exhaust port and traps a certain amount of steam, which acts as a cushion and stops the steam piston.

All valve adjustments are outside of the steam chest, and so it is possible to adjust the valve while the pump is in operation. The effect of decreasing or increasing the lost motion is the same as that described for duplex pumps. The lost-motion arrangement is the same as that shown in Figure 9c.

Steam End Materials For most services, cast iron is an excellent material for the steam cylinder and it is the major element of the steam end. It is readily cast in the complicated shape required to provide the steam porting. It possesses good wearing qualities, largely because of its free graphite content. This is required in the piston bores, which are continuously being rubbed by the piston rings. At high steam temperatures and pressures, ductile iron or steel is used. In the latter case, however, cast iron steam cylinder liners are frequently used because of their better wear resistance.

Counterbores are provided at each end of a steam cylinder so the leading piston ring can override, for a part of its width, the end of the cylinder bore to prevent the wearing of a shoulder on the bore.

The cylinder heads and steam pistons are also usually made of cast iron. The cylinder head has a pocket cast in it to receive the piston rod nut at the end of the stroke. Most steam pistons are made in one piece, usually with two piston ring slots machined into the outside circumference.

The relatively wide piston rings are usually made from hammered iron. They are split so they can be expanded to fit over the piston and snapped into the grooves in the piston.

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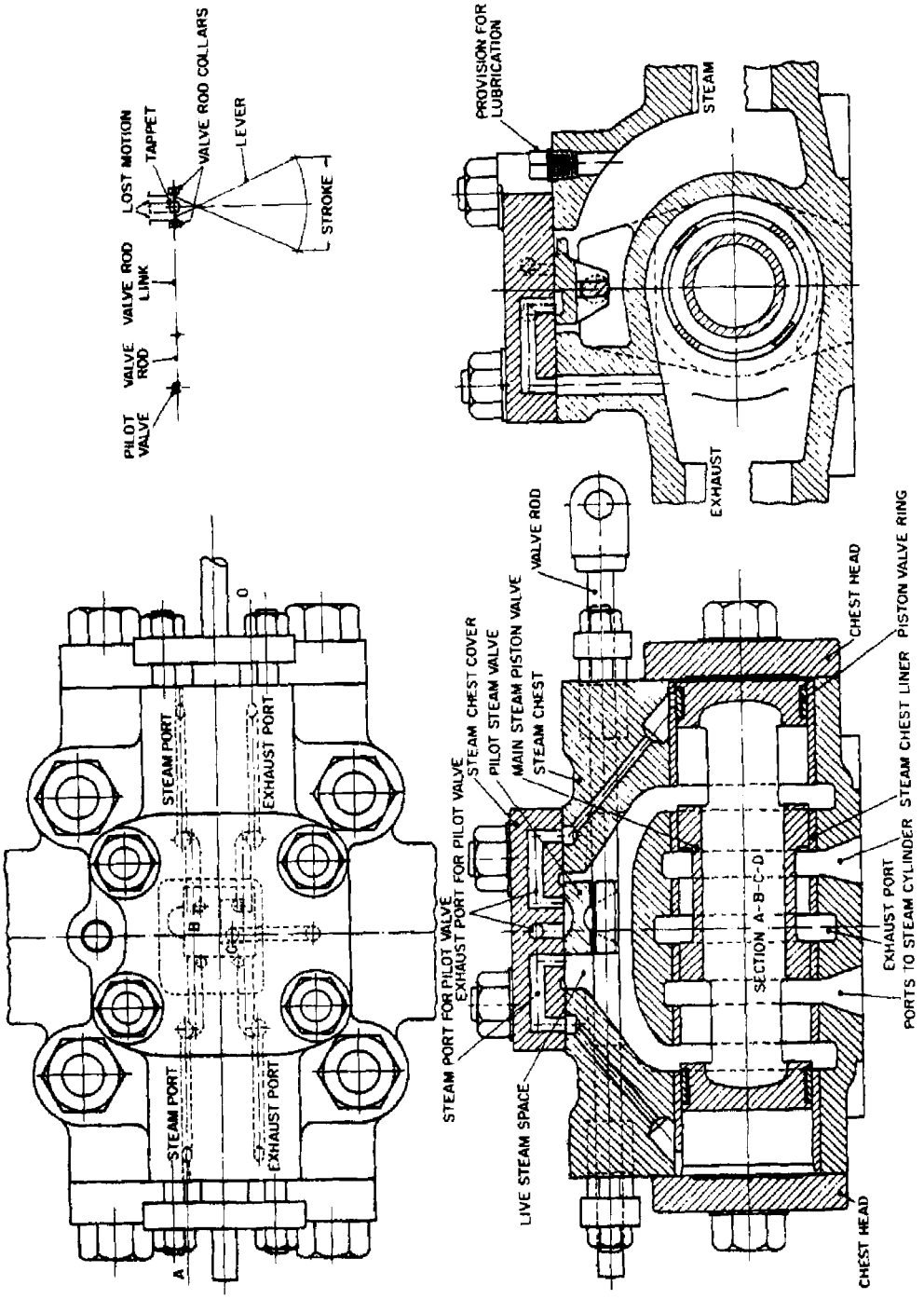


FIGURE 11 Simplex-type steam valve (Flowserve Corporation)

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They must be compressed slightly to fit into the bores in the cylinder. This ensures a tight seal with the cylinder bores even as the rings wear during operation. In services where steam cylinder lubrication is not permissible, a combination two-piece ring of iron and bronze is used to obtain longer life than is obtained with the hammered iron rings.

The piston and valve rods are generally made from steel, but stainless steel and Monel are also commonly used. Packing for the rods is usually a braided graphited asbestos.

Drain cocks and valves are always provided to permit drainage of condensation, which forms in the cylinder when a pump is stopped and cools down. On each start-up these must be cracked open until all liquid is drained and only steam comes out; they are then closed.

The steam end and liquid end are joined by a cradle. On most small pumps, the cradle is cast integrally with the steam end. On large pumps, it is a separate casting or fabricated weldment.

LIQUID END CONSTRUCTION

Steam pumps are equipped with many types of liquid ends, each being designed for a particular service condition. However, they can all be classified into two basic types, the piston, or inside-packed, type and the plunger, or outside-packed, type.

The piston pump (Figure 7) is generally used for low and moderate pressures. Because the piston packing is located internally, the operator cannot see the leakage past it or make adjustments that could make the difference between good operation and packing failure. Generally, piston pumps can be used at higher pressures with noncorrosive liquids having good lubricating properties, such as oil, than with corrosive liquids, such as water.

Plunger pumps, illustrated in Figure 12, are usually favored for high-pressure and heavy-duty service. Plunger pumps have stuffing box packing and glands of the same type as those on the piston rods of piston pumps. All packing leakage is external, where it is a guide to adjustments that control the leakage and extend packing and plunger life. During operation, lubrication can be supplied to the external plunger packing to extend its life. Lubrication cannot be supplied to the piston packing rings on a piston pump.

Piston-Type Liquid Ends The most generally used piston pump is the cap-and-valve plate design, illustrated in Figure 7. This is usually built for low pressures and temperatures, although some designs are used at up to 350 lb/in^2 (246 bar) of discharge pressure and 350°F (177°C). The discharge valve units are mounted on a plate separate from the cylinder and have a port leading to the discharge connection. A dome-shaped cap, subject to discharge pressure, covers the discharge valve plate. The suction valve units are mounted in the cylinder directly below their respective discharge valves. A passage in the liquid cylinder leads from below the suction valves down between the cylinders of a duplex pump to the suction connection.

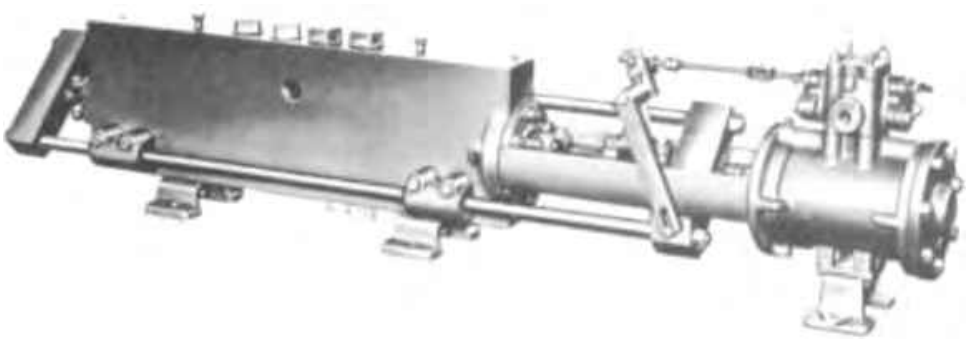
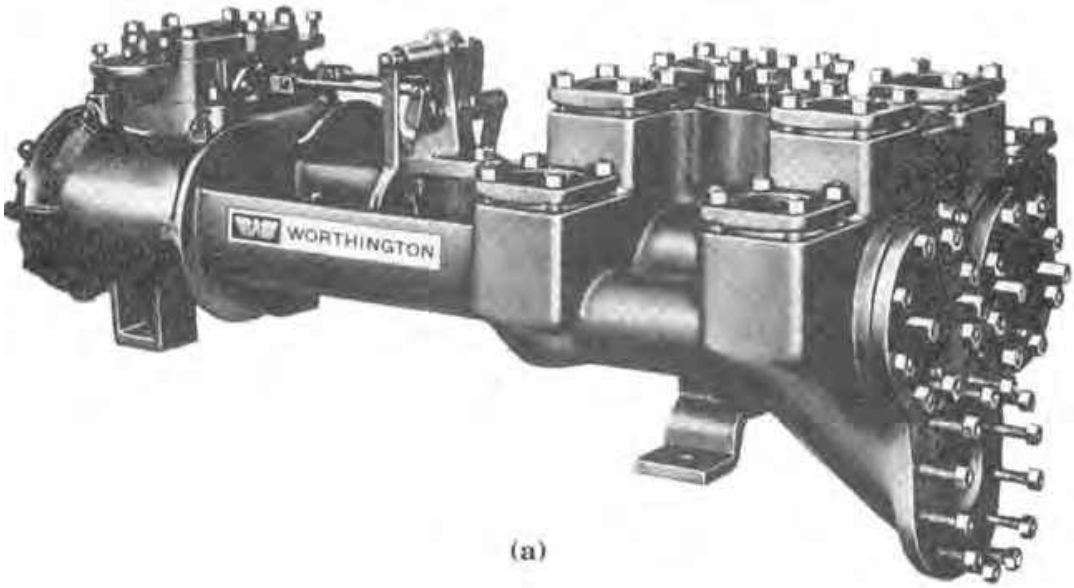
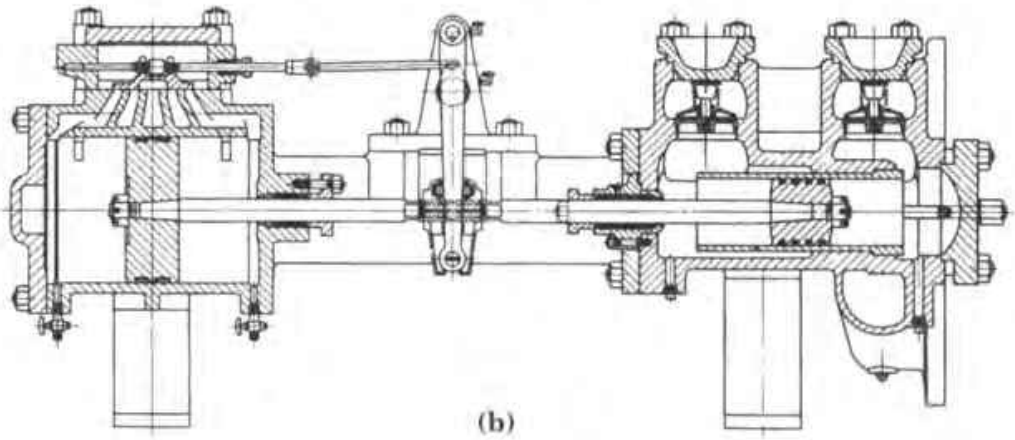


FIGURE 12 Simplex-type plunger pump (Flowserve Corporation)

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(a)



(b)

FIGURE 13A and B Side-pot piston pump (Flowserve Corporation)

Side-pot liquid ends are used where the operating pressures are beyond the limitations of the cap-and-valve-plate pump. Figure 13 illustrates this design. Suction valves are placed in individual pots on the side of the cylinders and discharge valves in the pots above the cylinders. Each valve can be serviced individually by removing its cover. The small area of the valve covers exposed to discharge pressure makes the sealing much simpler than is the case in the cap-and-valve design. Side-pot liquid ends are widely used in refinery and oil field applications. This design is commonly employed to the maximum pressure practicable for a piston pump.

There are several specially designed piston-type liquid ends that have been developed for specific applications. One of these is the close-clearance design illustrated in Figures 14 and 15. This pump can handle volatile liquids, such as propane or butane, or a liquid that may contain entrained vapors.

The close-clearance cylinder is designed to minimize the dead space when the piston is at each end of its stroke. The liquid valves are placed as close as possible to the pump chamber to keep clearance to a minimum. The suction valves are positioned below the cylinder at the highest points in the suction manifold to ensure that all the gases are passed into the pump chamber. Although these pumps are of close-clearance design, they are not compressors and can vapor-bind; that is, a large amount of gas trapped below the

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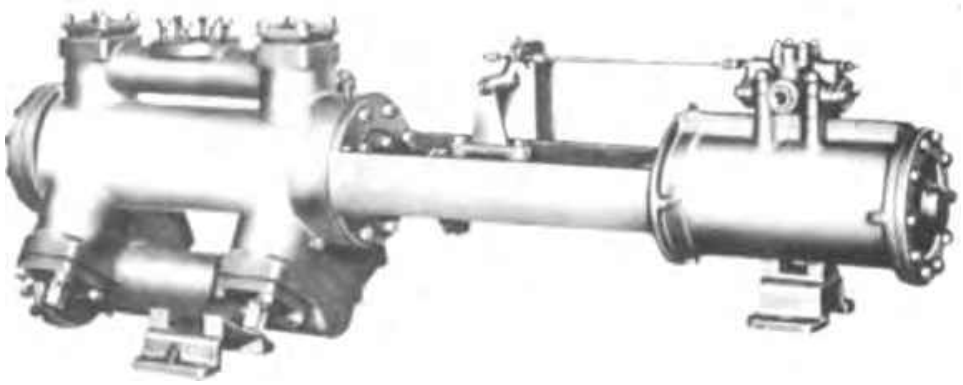


FIGURE 14 Close-clearance liquid end pump (Flowserve Corporation)

discharge valve will compress and absorb the entire displacement of the pump. When this occurs, the discharge valve will not open and this will cause a loss of flow. Hand-operated bypass or priming valves are provided to bypass the discharge valve and permit the trapped gases to escape to the discharge manifold. When the pump is free of vapors, the valves are closed.

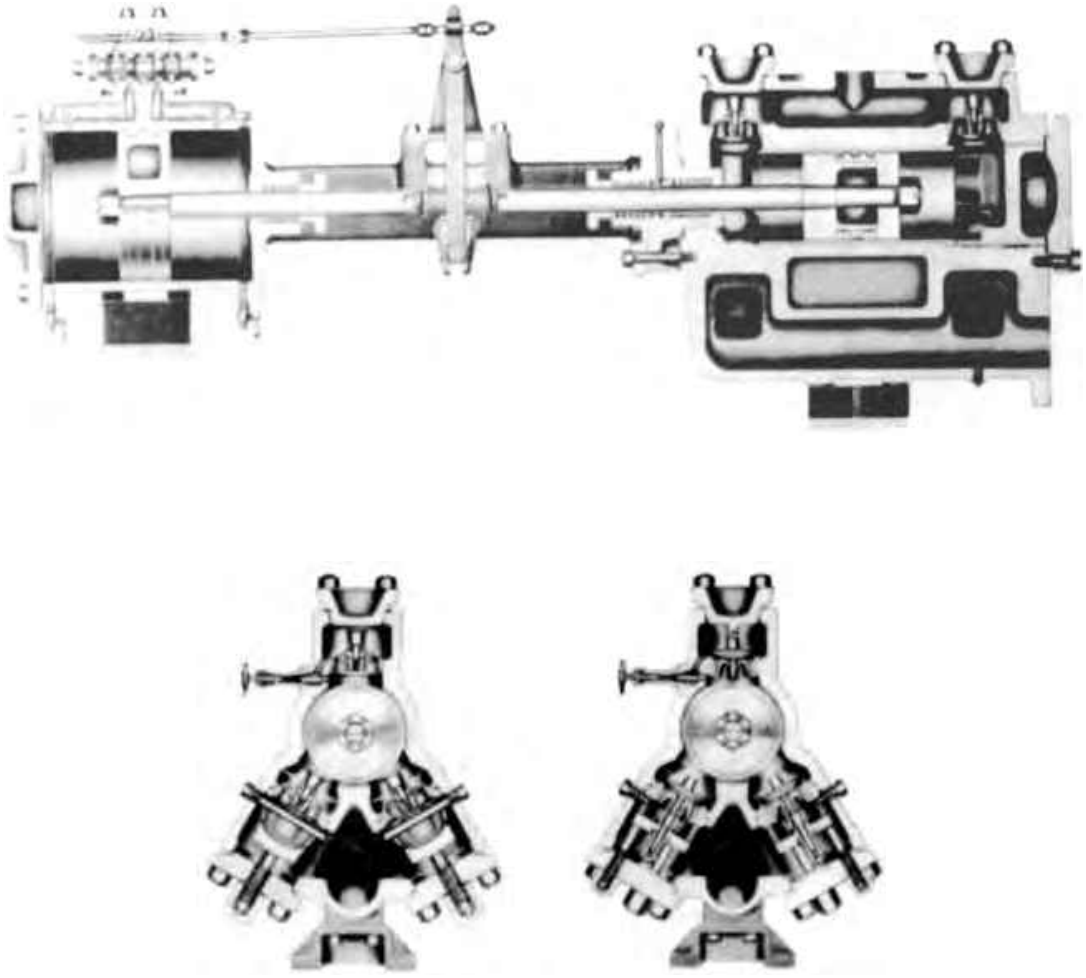


FIGURE 15 End views of close-clearance liquid end pump, showing disk valve assembly and wing valve assembly (Flowserve Corporation)

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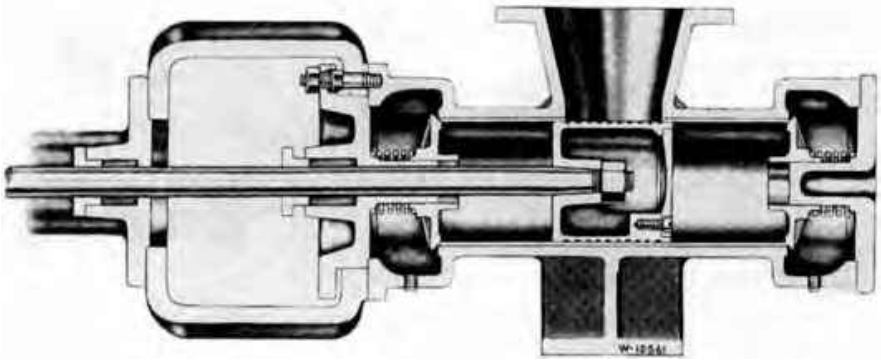


FIGURE 16 A pump for handling viscous liquid (Flowserve Corporation)

There are a number of other special piston pump designs for certain services in addition to the most common types just described. One of these special designs is the wet vacuum pump, which features tight-sealing rubber valves that permit the pump to handle liquid and air or non-condensable vapors. Another special design is made of hard, wear-resistant materials to pump cement grout on construction projects. Another design, shown in Figure 16, has no suction valves and is made for handling viscous products such as sugarcane pulp, soap, white lead, printer's ink, and tar. The liquid flows into the cylinder from above through a suction port that is cut off as the piston moves back and forth.

Piston Pump Liquid End Materials and Construction The materials used for piston pump liquid ends vary widely with the liquids handled. Most of the services to which these pumps are applied use one of the common material combinations listed in Table 1.

TABLE 1 Material and service specifications for pump liquid ends

Part	Regular Fitted (RF)	Bronze Fitted (BF)	Fully Bronze Fitted (FBF)	All Iron Fitted (AIF)	All Bronze (AB)
Cylinders	Cast iron	Cast iron	Cast iron	Cast iron	Bronze
Cylinder liners	Bronze	Bronze	Bronze	Cast iron	Bronze
Piston	Cast iron	Cast iron	Bronze	Cast iron	Bronze
Piston packing	Fibrous	Fibrous	Fibrous	Cast-iron, 3-ring	Fibrous
Stuffing box	Cast iron, bronze bushed	Cast iron, bronze bushed	Cast iron, bronze bushed	Cast iron	Bronze
Piston rod	Steel	Bronze	Bronze	Steel	Bronze
Valve service	Bronze	Bronze	Bronze	Steel	Bronze
Services for which most often used	Cold water; other cold liquids not corrosive to iron and bronze	Same as for RF, with reduced maintenance; continuous hot water	Boiler feed; intermittent hot-water; sodium chloride, brines	Oils and other hydrocarbons not corrosive to iron or steel; caustic solutions	Mild acids that would attack iron cylinders but not acid-resisting bronze

The liquid cylinder, the largest liquid end component, is most frequently made from cast iron or bronze. However, other materials are also used. Cast steel cylinders are used in refineries and chemical plants for high-pressure and high-temperature applications. Nickel cast steels are used for low-temperature services. Ni-Resist cast iron, chrome-alloy steels, and stainless steels are occasionally used for certain corrosive and abrasive applications, but tend to make pump cost very high. The liquid cylinder heads and valve covers are usually made from the same material as the liquid cylinder.

As was the case in the steam end, a liquid cylinder liner is used to prevent wear and permanent damage to the liquid cylinder. Liners must be replaced periodically when worn by the piston packing to the point that too much fluid leaks from one side of the piston to the other. The liners may be either of a driven-in (or pressed-in) type or of a removable type, which is bolted or clamped in position in the cylinder bore.

The pressed-in type (Figure 7) derives its entire support from the drive fit in the cylinder bore. As a rule, such a liner is relatively thin and is commonly made from a centrifugal casting or a cold-drawn brass tube. After a driven liner is worn to the point where it must be replaced, it is usually removed by chipping a narrow groove along its entire length. This groove is cut as closely as possible through the liner without damaging the wall of the cylinder bore. A flange on the liner fits into a recess at the beginning of the cylinder bore. This flange is held in contact with a shoulder by jack bolts or a spacer between the cylinder head and the end of the liner. Sometimes a packing ring is used between the flange and shoulder for a positive seal. Removable liners are heavier than pressed-in ones.

There are several designs of pistons and piston packings used for various applications. The three most common are as follows:

1. The body-and-follower type of piston with soft fibrous packing or hard-formed composition rings (Figure 17). The packing is installed in the packing space on the piston with a clearance in both length and depth. This clearance permits fluid pressure to act on one end and the inside of the packing to hold and seal it against the other end of the packing space and the cylinder liner bore.
2. The solid piston or, as shown in Figure 18, a body and follower with rings of cast iron or other materials. This type is commonly used in pumps handling oil or other hydrocarbons. The metal rings are split with an angle or step-cut joint. Their natural tension keeps them in contact with the cylinder liner, assisted by fluid pressure under the ring.
3. The cup piston (Figure 19), which consists of a body-and-follower type of piston with molded cups of materials such as rubber reinforced with fabric. Fluid pressure on the inside of the cup presses the lip out against the cylinder bore, forming a tight seal.

The piston rod stuffing boxes are usually made separate from, but of the same material as, the liquid cylinder. When handling liquids with good lubrication properties, the stuffing boxes are usually packed full with a soft, square, braided packing that is compatible with the liquid. When the liquid has poor lubricating properties, a lantern ring is

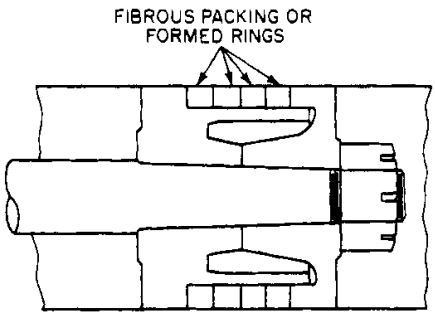


FIGURE 17 Body-and-follower piston

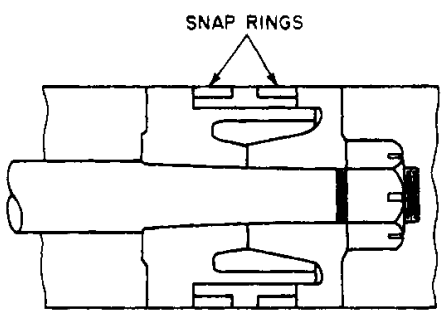


FIGURE 18 Body-and-follower piston with snap rings

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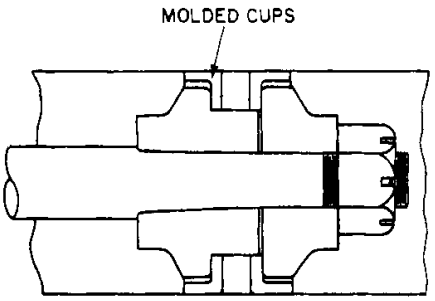


FIGURE 19 Body-and-follower piston with molded cups

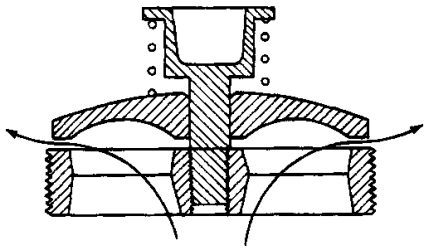


FIGURE 20 Stem-guided disk valve

installed in the center of the stuffing box with packing rings on both sides of it. A drilled hole is provided through which a lubricant, grease or oil, can be injected into the lantern ring from the outside of the stuffing box. At higher temperatures, approximately 500°F (260°C) or higher, a cooling water jacket is added to the outside of the stuffing box or as a spacer between the stuffing box and the liquid cylinder. The purpose of the cooling water jacket is to extend packing life by keeping the packing cool.

The liquid end valves of all direct-acting steam pumps are self-acting, in contrast to the mechanically operated slide valves in the steam end. The liquid end valves act like check valves; they are opened by the liquid passing through and are closed by a spring plus their weight.

Liquid end valves are roughly divided into three types: the disk valve for general service and thin liquids, the wing-guided valves for high pressures, and either the ball or the semispherical valve for abrasive and viscous liquids.

The valve shown in Figure 20 is typical of the disk type. This stem-guided design is commonly used in the cap-and-valve plate design. For hot-water boiler-feed and general service, the disk, seat, and stem are usually made of bronze, although other alloys may also be used. For lower temperatures and pressures, the disk may be made of rubber, which has the advantage of always forming a tight seal with the valve seat.

The wing-guided valve shown in Figure 21 is typical of the design use for high pressures. It derives its name from the wings on the bottom of the valve, which guide it in its seat. The beveled seating surfaces on the valve and seat tend to form a tighter seal than the flat seating surfaces on a disk valve. There is also less danger that a solid foreign particle in the liquid will be trapped between the seat and the valve. This type of valve is commonly made from a heat-treated chrome-alloy-steel forging, although a cast hard bronze and other materials may be used.

The ball valve, as its name suggests, is a ball that acts like a check valve. It is usually not spring-loaded, but guides and lift stops are provided as necessary to control its operation. The ball may be made of rubber, bronze, stainless steel, or other materials as service conditions require. The semispherical valve (Figure 22) is spring-loaded and can therefore be operated at higher speeds than the ball valve. Both the ball and the semispherical type

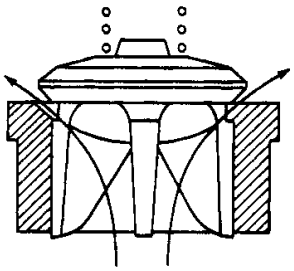


FIGURE 21 Wing-guided valve

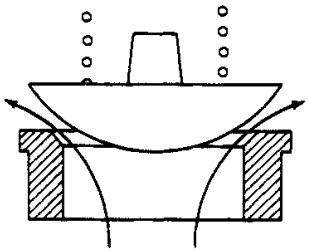


FIGURE 22 Semispherical valve

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have the advantage of having no obstructions to flow in the valve seat (the disk valve seat has ribs and the wing-guided valve has vanes which obstruct the flow). The one large opening in the seat and the smooth spherical surface of ball and semispherical valves minimize the resistance to flow of viscous liquids. These types are also used for liquids with suspended solids because their rolling seating action prevents trapping of the solids between the seat and valve.

Plunger-Type Liquid Ends As mentioned previously, plunger-type pumps are used where dependability is of prime importance, even when the pump is operated continuously for long periods and where the pressure is very high. Cast liquid end plunger pumps are used for low and moderate pressures. Forged liquid end pumps (Figure 23), which are the most common plunger types, are used for high pressures and have been built to handle pressures in excess of 10,000 lb/in^2 (69 MPa).

Most of these designs have opposed plungers; that is, one plunger operating into the inboard end of the liquid cylinder and one into the outboard end. The plungers are solidly secured to inboard and outboard plunger crossheads. The inboard and outboard plunger crossheads are joined by side rods positioned on each side of the cylinder. With this arrangement, each plunger is single-acting; that is, it makes only one pressure stroke for each complete reciprocating cycle. The pump, however, is double-acting because the plungers are connected by the side rods.

PLUNGER PUMP LIQUID END MATERIALS The liquid cylinder of a forged liquid end plunger-type pump is most commonly made from forged steel, although bronze, Monel, chrome alloy, and stainless steels are also used. The stuffing boxes and valve chambers are usually integral with the cylinder (Figure 23), which is desirable for higher temperatures and pressures because high-temperature joints are minimized.

The liquid plungers may be made of a number of materials. The plungers must be as hard and smooth as possible to reduce friction and to resist wear by the plunger packing. Hardened chrome-alloy steels and steel coated with hard-metal alloys or ceramics are most commonly used.

The stuffing box packing used will vary widely depending upon service conditions. A soft, square packing cut to size may be used. However, solid molded rings of square, V-lip, or U-lip design are commonly used at higher pressures. Oil or grease is frequently injected into a lantern ring in the center of the stuffing box to reduce friction and reduce packing and plunger wear.

The liquid valves may be of any of the types or materials described above. However, the wing-guided valve with beveled seating surfaces is the most common because it is most suitable for high pressures.

DIRECT-ACTING STEAM PUMP PERFORMANCE

The direct-acting steam pump is a very flexible machine. It can operate at any point of pressure and flow within the limitations of the particular design. The speed of, and therefore the flow from, the pump can be controlled from stop to maximum by throttling the steam supply. This can be done by either a manual or an automatically operated valve in the steam supply line. The maximum speed of a particular design is primarily limited by the frequency with which the liquid valves will open and close smoothly. The pump will operate against any pressure imposed upon it by the system it is serving, from zero to its maximum pressure rating. The maximum pressure rating of a particular design is determined by the strength of the liquid end. In a particular application, the maximum liquid pressure developed may be limited by the available steam pressure and by the ratio of the steam piston and liquid piston areas.

STEAM PUMP CAPACITY

The flow to the discharge system is termed the pump *capacity*. The capacity, usually expressed in U.S. gallons per minute (cubic meters per minute), is somewhat less than the

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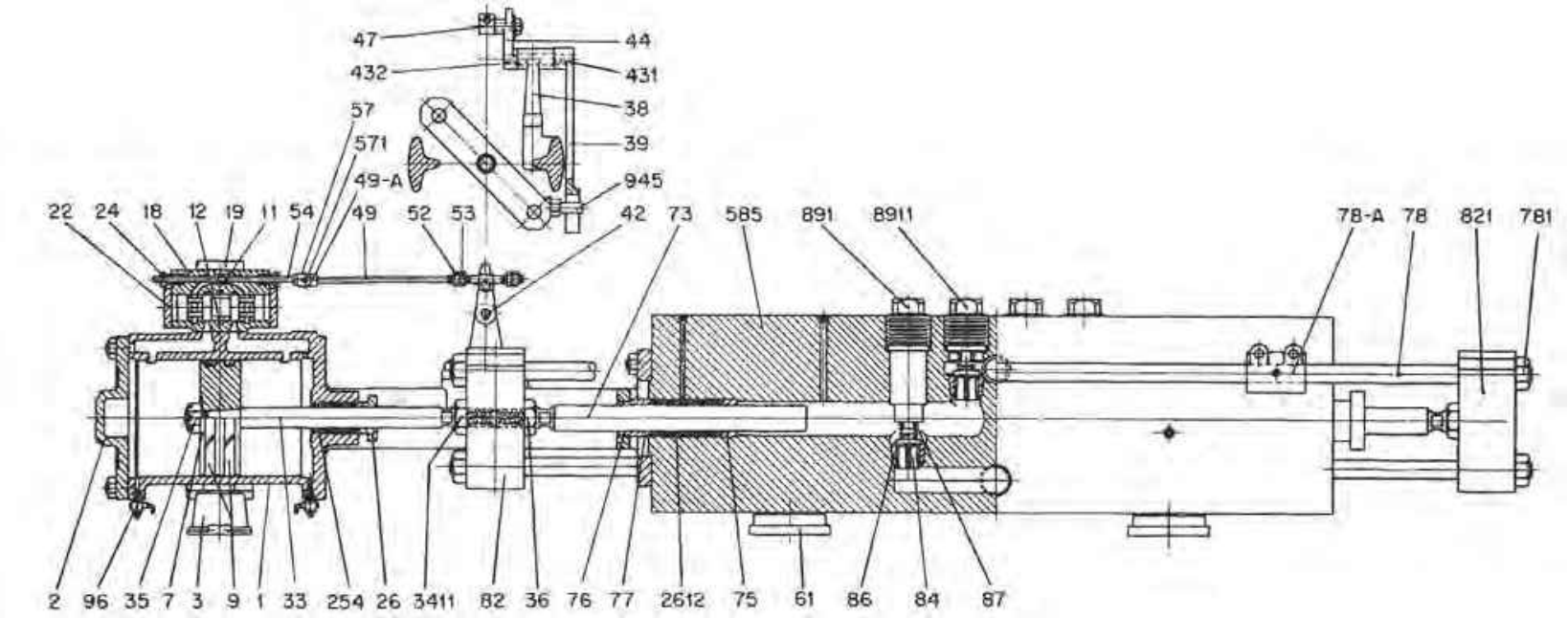


FIGURE 23 Simplex plunger pump with forged steel cylinder: (1) steam cylinder with cradle, (2) steam cylinder head, (3) steam cylinder foot, (7) steam piston, (9) steam piston rings, (11) slide valve, (12) piston valve, (18) steam chest, (19) steam chest cover, (22) steam chest head, (24) valve rod stuffing box gland, (26) steam piston rod stuffing box gland, (33) steam piston rod, (35) steam piston rod nut, (36) plunger nut, (38) cross stand, (39) lever, (42) fulcrum pin, (44) crank, (47) tappet, (49) valve rod link, (49A) valve rod link head, (52) lost-motion adjusting nut, (53) lost-motion locknut, (54) valve rod, (57) valve rod head, (61) liquid cylinder foot, (73) plunger, (75) plunger lining, (76) plunger gland flange, (77) plunger gland lining, (78) side rod, (78A) side rod guide, (82) plunger crosshead, front; (84) metal valve, (86) valve seat, (87) valve spring, (96) steam cylinder drain valve, (254) piston rod stuffing box bushing, (431) lever key, (432) crank key, (571) valve rod head pin, (585) liquid cylinder, (781) side rod nut, (821) plunger crosshead, rear, (891) suction valve plug, (945) crosshead pin, (2612) lantern gland, (8911) discharge valve plug, (3411) steam piston rod jam nut (Flowserve Corporation)

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theoretical *displacement* of the pump. The difference between displacement and capacity is called *slip*. The displacement is a function of the area of the liquid piston and the speed at which the piston is moving.

The displacement of a single double-acting piston can be calculated from the formula

in USCS units $$D = \frac{12 \times AS}{231} \text{ or } 0.0408\, d^2 S$$

in SI units $$D = 1 \times 10^{-6} \times AS$$

where D = displacement, gpm (m^3/min)
A = area of piston or plunger, in^2 (mm^2)
S = piston speed, ft/min (m/min)
d = diameter of the liquid piston or plunger, in (mm)

For a duplex double-acting pump, D is multiplied by 2. This formula neglects the area of the piston rod. For very accurate calculations, it is necessary to deduct the rod area from the piston area. This is normally not done, and the resultant loss is usually considered part of the slip.

The slip also includes losses due to leakage from the stuffing boxes, leakage across the piston on packed-piston pumps, and leakage back into the cylinder from the discharge side while the discharge valves are closing. Slip for a given pump is determined by a test. For a properly packed pump, slip is usually 3 to 5%. As a pump wears, slip will increase, but this can be compensated for by increasing the pump speed to maintain the desired capacity.

PISTON SPEED

Although piston speed in feet per minute (meters per minute) is the accepted term used to express steam pump speed, it cannot easily be measured directly and is usually calculated by measuring the revolutions per minute of the pump and converting this to piston speed. One revolution of a steam pump is defined as one complete forward and reverse stroke of the piston. The relationship between piston speed and rpm is

in USCS units $$S = \frac{\text{rpm} \times \text{stroke}}{6}$$

in SI units $$S = 0.002 \times \text{rpm} \times \text{stroke}$$

where S = piston speed, ft/min (m/min)
rpm = revolutions per minute
stroke = stroke of pump, in (mm)

A steam pump must fill with liquid from the suction supply on each stroke, or it will not perform properly. If the pump runs too fast, the liquid cannot flow through the suction line, pump passageways, and valves fast enough to follow the piston. On the basis of experience and hydraulic formulas, maximum piston speeds that vary with the length of stroke and the liquid handled can be established.

Table 2 shows general averages of maximum speed ratings for pumps of specified stroke handling various liquids. Some pumps, by reason of exceptionally large valve areas or other design features, may be perfectly suitable for speeds higher than shown. From the table, it should be noted that piston speed should be reduced for viscous liquids. Unless the net positive suction head is proportionately high, viscous liquids will not follow the piston at high speeds because frictional resistance in suction lines and in the pump increases with viscosity and rate of flow. Pumps handling hot water are run more slowly to prevent boiling of the liquid as it flows into the low-pressure area behind the piston.

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TABLE 2 Average maximum speed ratings

Stroke Length, in*	Piston speed, ft/min*: Cold Water; Oil to 250 SSU	Oil, 250–500 SSU	Oil, 500–1000 SSU	Oil, 1000–2500 SSU	Oil, 2500–5000 SSU	Boiler Feed 212°F (100°C)
3	37	35	33	29	24	22
4	47	45	42	36	31	28
5	53	51	47	41	35	32
6	60	57	53	46	39	36
7	64	61	57	49	42	39
8	68	65	61	53	45	41
10	75	72	67	58	49	45
12	80	77	71	62	52	48
15	90	86	80	69	57	54
18	95	91	85	73	62	57
24	105	100	94	81	68	63
36	120	115	107	92	78	72

*SI conversion factors: in × 25.4 = mm; ft/min × 0.3048 = m/min.

SIZE OF LIQUID END

The size of a steam pump is always designated as follows:

Steam piston diameter × liquid piston diameter × stroke

For example, a $\frac{1}{2} \times 5 \times 6$ steam pump has a $7\frac{1}{2}$-in (191-mm) diameter steam piston, a 5-in (127-mm) diameter liquid piston, and a 6-in (152-mm) stroke.

To determine the liquid piston diameter for a specified capacity, the following procedure is used. First, a reasonable stroke length is assumed and the maximum piston speed for this stroke and the type of liquid pumped is selected from Table 2. Then the desired capacity is increased by 3 to 5% to account for slip. The result is the desired displacement. Then for either a simplex or duplex pump, the liquid piston diameter can be calculated as follows:

In USCS units, for simplex pumps: $$d_l = 4.95\left(\frac{D}{S}\right)^{1/2}$$

For duplex pumps: $$d_l = 3.5\left(\frac{D}{S}\right)^{1/2}$$

In SI units, for simplex pumps: $$d_l = 1128.4\left(\frac{D}{S}\right)^{1/2}$$

For duplex pumps: $$d_l = 797.9\left(\frac{D}{S}\right)^{1/2}$$

where d_l = liquid piston diameter, in (mm)
D = displacement, gpm (m^3/min)
S = piston speed, ft/min (m/min)

Using the resultant liquid piston diameter, the next larger standard piston size is selected.

SIZE OF STEAM END

To calculate the size of the steam end required for a specific application, the basic principle of steam pump operation should be considered. A simple schematic of a steam pump is shown in Figure 24.

In order for the pump to move, the force exerted on the steam piston must exceed the force on the liquid piston that is opposing it. The force on the steam piston is the product of the net steam pressure and the steam piston area. The *net* steam pressure is the steam inlet pressure minus the exhaust pressure. The force acting on the liquid piston is the product of the net liquid pressure and the liquid piston area. The *net* liquid pressure is the pump discharge pressure minus the suction pressure or *plus* the suction lift. This may be expressed algebraically as follows:

$$P_s A_s > P_l A_l$$

where P_s = net steam pressure, lb/in^2 (bar)
A_s = steam piston area, in^2 (mm^2)
P_l = net liquid pressure, lb/in^2 (bar)
A_l = liquid piston area, in^2 (mm^2)

Because the pistons are circular, the squares of their diameters are directly proportional to their areas, and the above formula can be rewritten as

$$P_s d_s^2 > P_l d_l^2$$

where d_s = steam piston diameter, in (mm)
d_l = liquid piston diameter, in (mm)

In practice, it is necessary for the force on the steam piston to exceed the force opposing it on the liquid piston by a considerable amount. This is because of mechanical losses, which include stuffing box friction, friction between piston rings and cylinder of both liquid and steam ends, and the operation of the valve gear. These losses are determined by testing and are accounted for in size calculations by introduction of a mechanical efficiency figure. Mechanical efficiencies are expressed as a percentage, with 100% being a perfect balance of forces acting on the steam and liquid pistons as expressed in the previous formula. Because the efficiencies of two identical pumps may vary with stuffing box and piston ring packing tightness, the efficiencies published by manufacturers tend to be conservative.

With the mechanical efficiency factor inserted, the formula of forces becomes

$$P_s d_s^2 E_m = P_l d_l^2$$

where P_s = net steam pressure, lb/in^2 (bar)
d_s = steam piston diameter, in (mm)

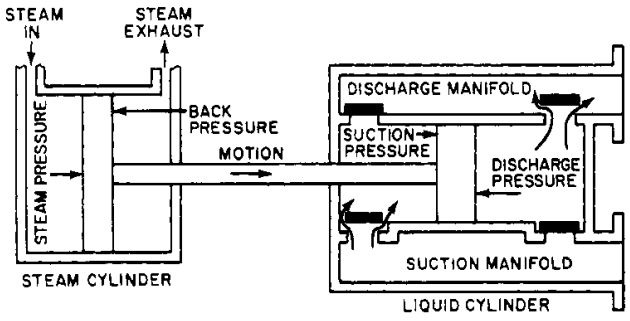


FIGURE 24 Schematic showing direction of forces on pistons

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TABLE 3 Typical mechanical efficiency values

	Mechanical efficiency, %	
Stroke length, in (mm)	Piston Pump	Plunger pump
3 (76)	50	47
4 (102)	55	52
5 (127)	60	57
6 (152)	65	61
8 (201)	65	61
10 (254)	70	66
12 (305)	70	66
18 (457)	73	69
24 (610)	75	71

P_l = net liquid pressure, lb/in^2 (bar)
d_l = liquid piston diameter, in (mm)
E_m = mechanical efficiency, expressed as a decimal

This formula is commonly used to determine the *minimum* size of steam piston required when the liquid piston size has already been selected and the net steam and net liquid pressures are known. For this calculation, the formula is rearranged to the form

$$d_s = d_l\left(\frac{P_l}{P_s E_m}\right)^{1/2}$$

The efficiency of a long-stroke pump is greater than that of a short-stroke pump. Although mechanical efficiency varies with stroke length, any two pumps of the same size are capable of the same efficiency.

Table 3 shows typical mechanical efficiencies that can be used to determine required steam end size.

STEAM CONSUMPTION AND WATER HORSEPOWER

After determining the proper size of other pump types, the next concern usually is to calculate the maximum brake horsepower so the proper size of driver can be selected. With a steam pump, the next step is usually to determine the steam consumption. This must be known to ensure that the boiler generating the steam is large enough to supply the steam required by the pump as well as that required for all its other services.

To determine the steam consumption, it is necessary first to calculate the water horsepower as follows:

In USCS units $$whp = \frac{Q \times P_l}{1715}$$

In SI units $$\text{kW} = \frac{Q \times P_l}{36}$$

where whp = water horsepower
Q = pump capacity, gpm (m^3/h)
P_l = net liquid pressure, lb/in^2 (bar)

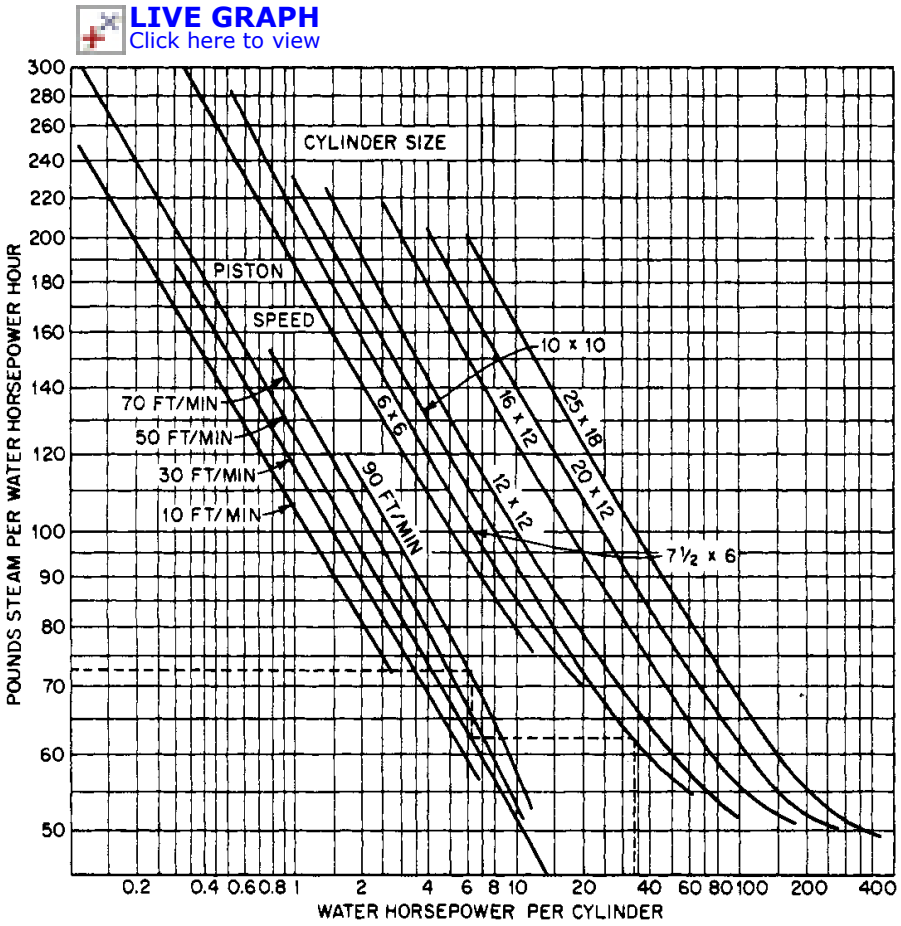


FIGURE 25 Approximate steam consumption for steam pumps (pounds of steam per water horsepower-hour = 1.644 kg of steam per water kilowatt-hour; water horsepower per cylinder = 1.341 wkW per cylinder; ft/s = 0.3047 m/s; in = 25.44 mm) (Hydraulic Institute Standards, 12th Edition, 1969—out of print)

A steam consumption chart (Figure 25) affords a means of quickly obtaining an approximate figure for the steam rate of direct-acting steam pumps. For duplex pumps, divide the water horsepower by 2 before applying it to the curves. These curves were made up on the basis of water horsepower per cylinder; if the above procedure is not followed, the results will be inaccurate.

Starting with the water horsepower per cylinder:

1. Move vertically to the curve for steam cylinder size.
2. Move horizontally to the curve for 50-ft/min (15.15-m/min) piston speed. This is the basic curve from which the other curves were plotted.
3. Move vertically to the piston speed at which the pump will run.
4. Move horizontally to the steam rate scale and read it in pounds per water horsepower-hour (kilograms per kilowatt-hour).
5. Multiply the result by total water horsepower to obtain the steam rate in pounds per hour (kilograms per hour).

For steam cylinders with diameters as shown, but with longer stroke, deduct 1% from the steam rate for each 20% of additional stroke. Thus, a 12 × 24 steam end will have a steam consumption about 5% less than a 12 × 12 steam end. For $5\frac{1}{4} \times 5$ and $4\frac{1}{2} \times 4$ steam ends, the 6 × 6 curve will give approximate figures. For cylinders of intermediate diameters, interpolate between the curves.

To correct for superheated steam, deduct 1% for each 10° of superheat. To correct for back pressure, multiply the steam rate by a correction factor equal to

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$$\left(\frac{P + BP}{P}\right)^{1/2}$$

where P = net steam pressure to drive pump, lb/in^2 (bar)

BP = back pressure, lb/in^2 (bar)

Direct-acting steam pumps have inherently high steam consumption. This is not necessarily a disadvantage, however, when the exhaust steam can be used for heating the boiler-feed water or for building heating or process work. Because these pumps can operate with a considerable range of back pressure, it is possible to recover nearly all the heat in the steam required to operate them. Because they do not use steam expansively, they are actually metering devices rather than heat engines and as such consume heat from the steam only as the heat is lost via radiation from the steam end of the pump. These pumps act, in effect, like a reducing valve to deliver lower pressure steam that contains nearly all its initial heat.

SUCTION SYSTEMS AND NET POSITIVE SUCTION HEAD

A majority of pump engineers agree that most operating problems with pumps of all types are caused by failure to supply adequate suction pressure to fill the pump properly.

The steam pump industry uses the term *net positive suction head required* (NPSHR) to define the head, or pressure, required by the pump over its datum, usually the discharge valve level. This pressure is needed to (1) overcome frictional losses in the pump, (2) overcome the weight and spring loadings of the suction valves, and (3) create the desired velocity in the suction opening and through the suction valves. The NPSHR of a steam pump will increase as the piston speed and capacity are increased. The average steam pump will have valves designed to limit the NPSHR to 5 lb/in^2 (0.3 bar) (34.5 MPa) or less at maximum piston speed.

If the absolute pressure in the suction system minus the vapor pressure is inadequate to meet or exceed the NPSHR, the pump will cavitate. Cavitation is the change of a portion of the liquid to vapor, and it causes a reduction in delivered capacity, erratic discharge pressure, and noisy operation. Even minor cavitation will require frequent refacing of the valves, and severe cavitation can cause cracked cylinders or pistons or failure of other major parts.

FLOW CHARACTERISTICS

The flow characteristics of duplex and simplex pumps are illustrated in Figure 26. The flow from a simplex pump is fairly constant except when the pump is at rest. However,

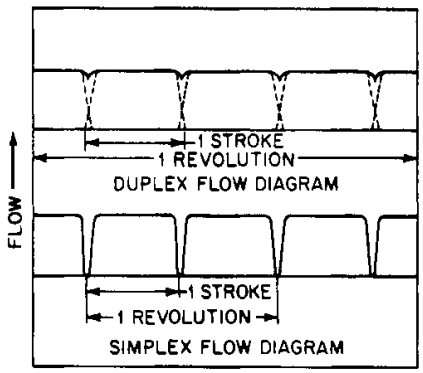


FIGURE 26 Flow characteristics of simplex and duplex pumps

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because the flow must stop for the valves to close and for the forces on both sides of the steam and liquid pistons to reverse, there is uneven and pulsating flow. This can be compensated for, in part, by installing a pulsation-dampening device on the discharge side of the pump or in the discharge line.

In a duplex pump, one piston starts up just before the other piston completes its stroke, and the overlapping of the two strokes eliminates the sharp capacity drop.

FURTHER READING

Grobholz, R. K. "Get the Right Steam Pump to Handle the Job." *Mill and Factory,* 1955.

American National Standard for Direct Acting (Steam) Pumps for Nomenclature, Definitions, Application and Operation, ANSI/HI 8.1-8.5-2000, Hydraulic Institute, Parsippany, NJ www.pumps.org.

Schaub, D. G. "Reciprocating Steam Pumps." *South. Power and Ind.,* 1955.

Wright, E. F. "Direct-Acting Steam Pumps." In *Standard Handbook for Mechanical Engineers* (T. Baumeister and L. Marks, eds.), 7th ed., McGraw-Hill, New York, 1967, pp. 14–9 to 14–14.

Wright, E. F. In *Pump Questions and Answers.* (R. Carter, I. J. Karassik, and E. F. Wright, eds.) McGraw-Hill, New York, 1949.

SECTION 3.4
DISPLACEMENT PUMP PERFORMANCE, INSTRUMENTATION, AND DIAGNOSTICS

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FRED R. SZENASI

The most common operational and reliability problems in reciprocating positive displacement pump systems are characterized by

- Low net positive suction head (NPSH), pulsations, pressure surge, cavitation, waterhammer
- Vibrations of pump or piping
- Mechanical failures, wear, erosion, alignment
- High horsepower requirements, high motor current, torsional oscillations
- Temperature extremes, thermal cycling
- Harsh liquids: corrosive, caustic, colloidal suspensions, precipitates (slurries)

Although any component in a pump system may be defective, most operational problems are caused by liquid transient interaction of the piping system and pump system or by purely mechanical interaction of the pump, drive system, foundation, and so on. This section discusses hydraulic and mechanical problems and suggests measurement and diagnostic procedures for determining the sources of these problems.

HYDRAULIC AND MECHANICAL PUMP PROBLEMS

Inadequate NPSH Net positive suction head available (NPSHA) is the static head plus atmospheric head minus lift loss, frictional loss, vapor pressure, and acceleration head available at the suction connection centerline.

Acceleration head can be the highest factor of NPSHA. In some cases, it is 10 times the total of all the other losses. Data from both the pump and the suction system are required

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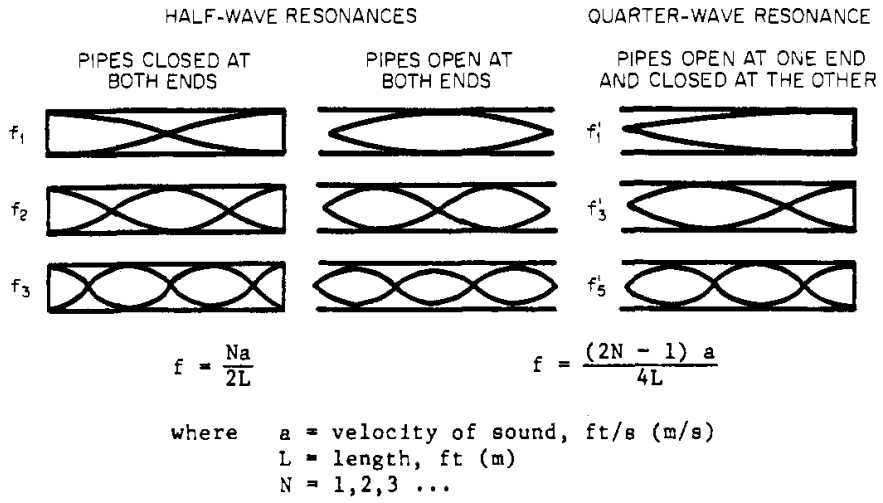


FIGURE 1 Organ pipe resonant mode shapes

to determine acceleration head; its value cannot be calculated until these data have been established. Inadequate NPSH can cause cavitation, the rapid collapse of vapor bubbles, which can result in a variety of pump problems, including noise, vibration, loss of head and capacity, and severe erosion of the valves and surfaces in the adjacent inlet areas. To avoid cavitation of liquid in the pump or piping, the absolute liquid static pressure at pumping temperatures must always exceed the vapor pressure of the liquid. The pressure at the pump suction should include sufficient margin to allow for the presence of pulsations as well as pressure losses due to flow.

Positive Displacement Pump Pulsations The intermittent flow of a liquid through pump internal valves generates liquid pulsations at integral multiples of the pump operating speed. For example, a 120-rpm triplex pump generates pulsations at all multiples of pump speed (2 Hz, 4 Hz, and so on); however, the most significant components will usually be multiples of the number of plungers (6 Hz, 12 Hz, 18 Hz, and so on). Resultant pulsation pressures in the piping system are determined by the interaction of the generated pulsation spectrum from the pump and the acoustic length resonances of liquid in the piping. For variable-speed units, the discrete frequency components change in frequency as a function of operating speed and the measured amplitude of any pulsation harmonic can vary substantially with changes in the location of the measurement point relative to the pressure nodes and antinodes of the standing wave pattern.

Piping System Pulsation Response Because acoustic liquid resonances occur in piping systems of finite length, these resonances will selectively amplify some pulsation frequencies and attenuate others. Resonances of individual piping segments can be described from organ pipe acoustic theory. The resonant frequencies of standing pressure waves depend upon the velocity of sound in the liquid being pumped, pipe length, and end conditions. The equations for calculating these frequencies are shown in Figure 1. All of the integral multiples (N) of a resonance can occur, and it is desirable to mismatch the excitation frequencies from any acoustical resonances. A 2:1 diameter increase or greater would represent an open end for the smaller pipe. Closed valves, pumps, or a 2:1 diameter reduction represent closed ends. For example, a 2-in (51-mm) diameter pipe that connects radially into two 8-in (203-mm) diameter volumes would respond acoustically as an open-end pipe.

Complex piping system responses depend upon the termination impedances and interaction of acoustical resonances and cannot be handled with simplified equations. An electroacoustic analog[1] or digital computer can be used for the more complex systems.

Velocity of Sound in Liquid Piping Systems The acoustic velocity of liquids can be determined by the following equation:

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$$a = C_1\sqrt{\frac{K_s}{\text{sp. gr.}}} \tag{1}$$

where a = velocity of sound, ft/s (m/s)
C_1 = 8.615 for USCS units, 1.0 for SI units
K_s = isentropic bulk modulus, lb/in² (kPa)
sp. gr. = specific gravity

In liquid piping systems, the acoustic velocity can be significantly affected by pipe wall flexibility. The acoustic velocity can be adjusted by the following equation:

$$a_{\text{adjusted}} = a\sqrt{\frac{1}{1 + \frac{DK_s}{tE}}} \tag{2}$$

where D = pipe diameter, in (mm)
t = pipe wall thickness, in (mm)
E = elastic modulus of pipe material, lb/in² (kPa)

Pipe wall radial compliance can reduce the velocity of sound in liquid in a pipe as shown in Figure 2.[2]

The bulk modulus of water can be calculated with the following equation[3] for temperatures from 0 to 212°F (0 to 100°C) and pressures from 0 to 4.4×10^4 lb/in² (0 to 3 kbar*):

$$K_s = K_0 + 3.4P \tag{3}$$

where K_s = isentropic bulk modulus, kbar
K_0 = constant from Table 1, kbar
P = pressure, kbar
(1 kbar = 10^5 kPa = 14,700 lb/in²)

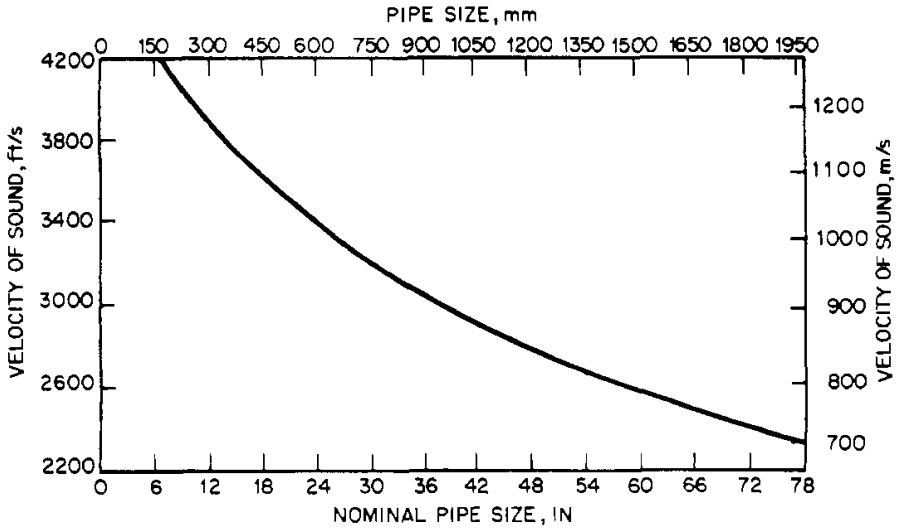


FIGURE 2 Velocity of sound in water at 14.6 lb/in² (1 bar), 60°F (15.6°C) versus nominal pipe size with 0.25-in (6.35-mm) wall thickness

*1 bar = 10^5 Pa.

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TABLE 1 Constant K_0 for evaluation of isentropic bulk modulus of water from 0 to 3 kbar

Temperature, °C (°F)		Isentropic constant K_0, kbar[a]
0	(32)	19.7
10	(50)	21.0
20	(68)	22.0
30	(86)	22.7
40	(104)	23.2
50	(122)	23.5
60	(140)	23.7
70	(158)	23.7
80	(176)	23.5
90	(194)	23.3
100	(212)	22.9

[a]1 kbar = 14,700 lb/in^2.
Source: Reference 3.

The calculation of the isentropic bulk modulus of water is accurate to ±0.5% at 68°F (20°C) and lower pressures.[4] At elevated pressures (greater than 3 kbar) and temperatures (greater than 100°C), the error should not exceed ±3%.

The bulk modulus for petroleum oils (hydraulic fluids) can be obtained at various temperatures and pressures by using Figures 3 and 4, which were developed by the American

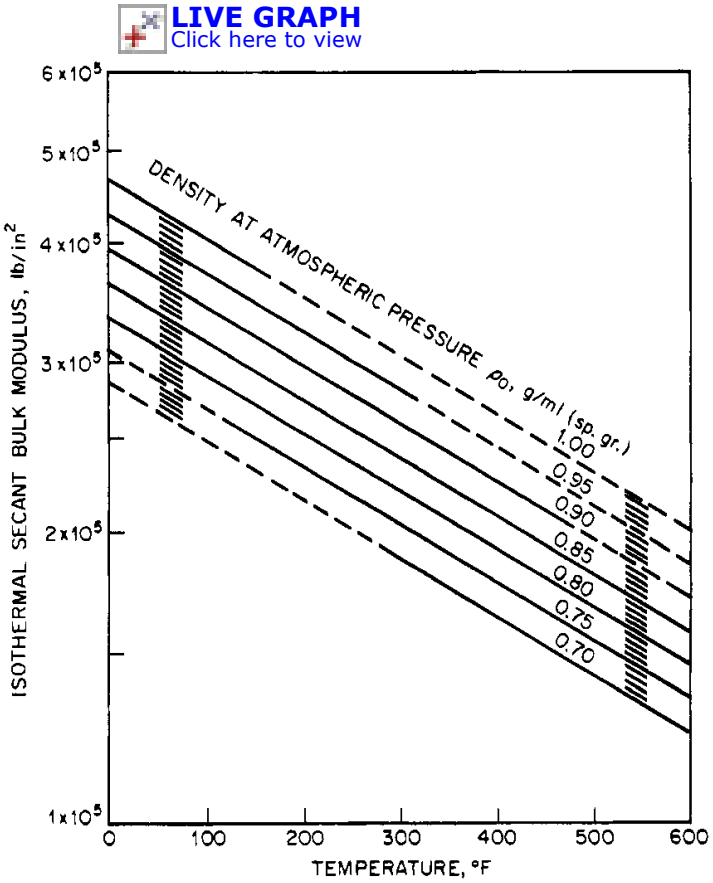


FIGURE 3 Isothermal secant bulk modulus at 20,000 lb/in^2 gage for petroleum oils (1 lb/in^2 = 6.895 kPa; °C = (°F − 32)/1.8).

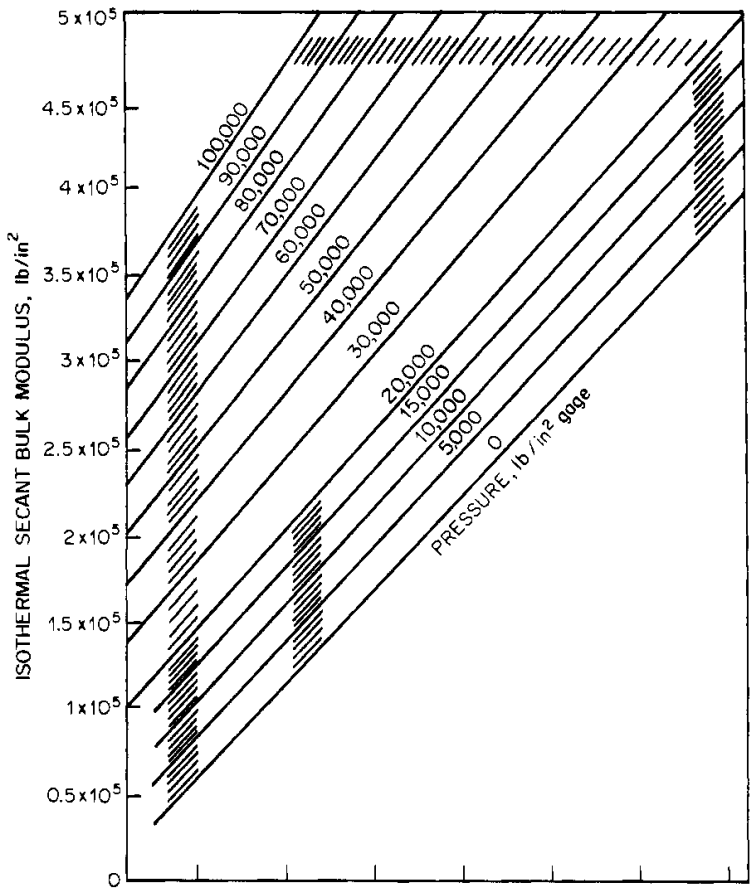


FIGURE 4 Pressure correction for isothermal secant bulk modulus for petroleum oils (1 lb/in² = 6.895 kPa)

Petroleum Institute (API).[4-7] Figure 3 relates density (mass per unit volume) and temperature to the isothermal secant bulk modulus at 20,000 lb/in² (137,900 kPa), and Figure 4 corrects for various pressures.

The isentropic tangent bulk modulus is needed to calculate the speed of sound in hydraulic fluids and can be readily obtained from Figures 3 and 4 as follows:

1. Read the isothermal secant bulk modulus at the desired temperature from Figure 3.
2. Using the value for isothermal secant bulk modulus obtained from Figure 3, go to Figure 4 and locate the intersection of the pressure line with that value. Move vertically to the pressure line representing twice the normal pressure. Read the adjusted isothermal secant bulk modulus for the double value of pressure.
3. Multiply the adjusted isothermal secant bulk modulus by 1.15 to obtain the value of the isentropic tangent bulk modulus (compensation for the ratio of specific heats).

The isothermal tangent bulk modulus has been shown to be approximately equal to the secant bulk modulus at twice the pressure[5] within ±1%. The relationship between isothermal bulk modulus K_t and isentropoic bulk modulus K_s is

$$K_s = K_t c_p / c_v \tag{4}$$

The value of c_p/c_v for most hydraulic fluids is approximately 1.15.

Pulsation Control Pulsation control can be achieved by judicious use of acoustic filters and side branch accumulators. Acoustic filters are liquid-filled devices consisting of volumes and chokes that use reactive filtering techniques to attenuate pulsations. Side

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branch resonators are of two types: quarter-wavelength resonant stubs and gas-charged accumulators. (The terms *accumulators, dampeners,* and *dampers* are used interchangeably in the liquid filter industry.)

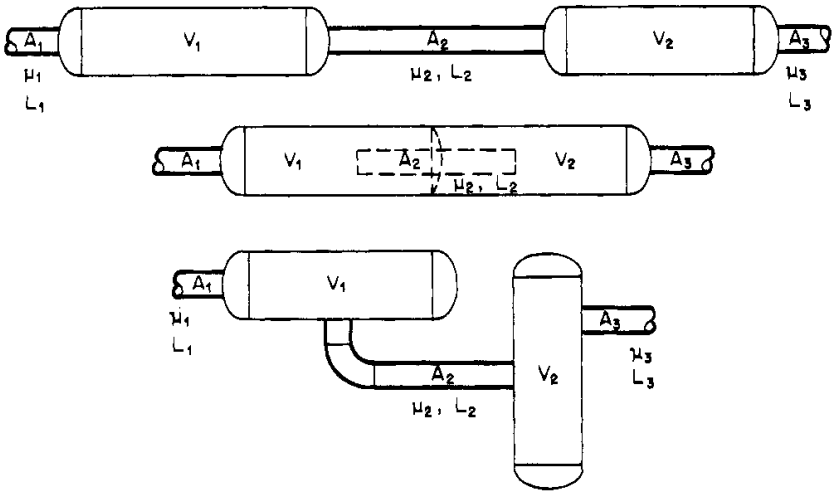
Acoustic Filters An acoustic filter consisting of two volumes connected by a small-diameter choke can significantly reduce the transmission of pulsations from the pump into the suction and discharge piping systems. The equations given in Figure 5 can be used to calculate the resonant frequency for a simple volume-choke-volume filter. The filter should be designed to have a resonant frequency no more than one-half the lowest frequency desired to be reduced, referred to as the cutoff frequency. Such a filter is called a low-pass filter because it attenuates frequencies above the cutoff frequency.

A special case for symmetric liquid-filled filters can be obtained by choosing equal chamber and choke lengths. This reduces the equation in Figure 5 to

$$f = \frac{ad}{\pi\sqrt{2}CD} \tag{5}$$

where d = choke diameter, in (mm)
L = chamber and choke length, ft (m)
D = chamber diameter, in (mm)

Normally, a good filter design will have a resonant frequency less than one-half the plunger frequency and will have a minimal pressure drop. For example, a triplex pump running at 600 rpm generates pulsations at all multiples of 10 Hz. The largest amplitudes



$$\mu_j = \frac{A_j}{L_j + 1/2\sqrt{\pi A_j}} \quad \text{for } j = 1, 2, 3$$

$$f = \frac{a}{2\pi}\sqrt{\frac{1}{2}\left[\frac{\mu_1 + \mu_2}{V_1} + \frac{\mu_2 + \mu_3}{V_2} \pm \sqrt{\left(\frac{\mu_1 + \mu_2}{V_1} - \frac{\mu_2 + \mu_3}{V_2}\right)^2 + \frac{4\mu_2^2}{V_1 V_2}}\right]}$$

FOR EQUAL VOLUMES, THE RESONANT FREQUENCY IS APPROXIMATELY:

$$f = \frac{a}{\sqrt{2}\,\pi}\sqrt{\frac{\mu_2}{V_1}}$$

FIGURE 5 Two-chamber resonator system with both ends open. V = volume, ft³ (m³); f = resonant frequency, Hz; L = choke tube length, ft (m); A = choke tube area, ft² (m²); a = acoustic velocity, ft/s (m/s); m = acoustic parameter

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would normally be at 30 Hz, 60 Hz, 90 Hz, and so on. The filter resonant frequency should be set at 15 Hz or lower. For water, in which the velocity of sound is 3200 ft/s (976 m/s), and a volume bottle size inner diameter of 19 in (482.6 mm),

in USCS units
$$\frac{L}{d} = \frac{3200}{\pi\sqrt{2}\,(15)(19)} = 2.53\,\frac{\text{ft}}{\text{in}}$$

in SI units
$$\frac{L}{d} = \frac{976}{\pi\sqrt{2}\,(15)(482.6)} = 0.03\,\frac{\text{m}}{\text{mm}} \qquad (6)$$

If the choke diameter is selected to be 1.049 in (26.64 mm), the length of each volume bottle and choke tube is 2.65 ft (0.81 m). See also Reference 8.

Side Branch Accumulators Liquid-filled, quarter-wavelength, side branch accumulators reduce pulsations in a narrow frequency band and can be effective on constant/speed positive displacement pumps. However, in variable-speed systems, accumulators with or without a bladder can be made more effective by partially charging them with a gas (nitrogen or air) because the gas charge cushions hydraulic shocks and pulsations. If properly selected, located, tuned, and charged, a wide variety of accumulators (weight- or spring-loaded; gas-charged) can be used in positive displacement pump systems to prevent cavitation and waterhammer, damp pulsations, and reduce pressure surges.[9,10] Improper sizing or location can aggravate existing problems or cause additional ones. Typically, the best location for accumulators is as close to the pump as possible.

Gas-charged dampeners, or accumulators, such as those depicted in Figure 6, are most commonly used and can be quite effective in controlling pulsations. These devices are commercially available from several sources. Their location and volume and the pressure of the charge are important to their effectiveness. When gas-charged dampeners are used, the gas pressure must be monitored and maintained because the gas can be absorbed into the liquid. The system pressure can sometimes be lower than the gas charging pressure, such as on start-up; therefore, a valve should be installed to shut off the accumulator during start-up to eliminate gas leakage to the primary liquid. When the valve is closed, the accumulator is decoupled from the system and is not effective. Accumulators with integral check valves should be adjusted so pressure transients do not close the check valve and render the accumulator ineffective. Accumulators that have bladders (Figures 6b, d, and e) to separate the gas charge from the liquid have some distinct advantages, particularly if gas absorption is a problem. Accumulators with flexible bladders must be carefully maintained because failure of a bladder could release gas into the liquid system and could compromise the effectiveness of the dampener.

The in-line gas dampener (refer to Figure 6e) has a cylinder around the pipe containing a gas volume and bladder. The liquid enters the dampener through small holes in the circumference of the pipe and impinges upon the bladder, which produces the same acoustic effect as a side branch configuration.

It is not always possible to design effective pulsation control systems using simplified techniques. For complicated piping systems with multiple pumps, an electroacoustic

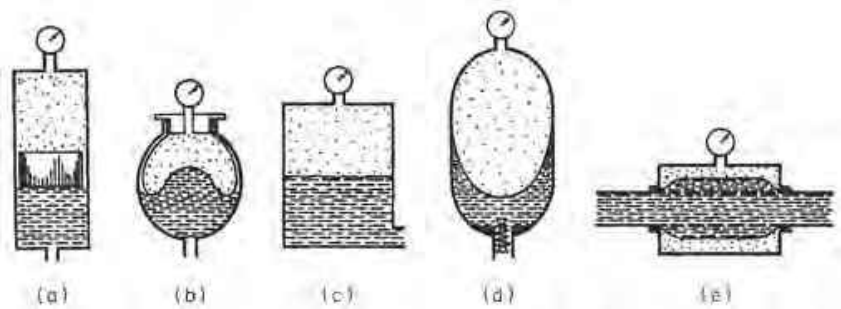


FIGURE 6 Types of accumulators: (a) piston, (b) diaphragm, (c) gas-charged, (d) bladder, (e) in-line

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LIVE GRAPH
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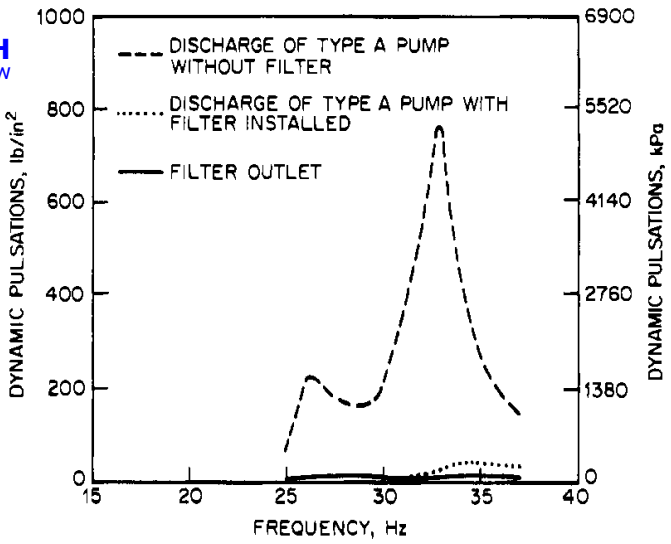


FIGURE 7 Effect of acoustic filter on pulsations

analog[1] is recommended for designing optimum filters or accumulators. This tool has become widely accepted for designing reliable piping systems for reciprocating liquid pumps and gas compressor units. The reduction in pulsations in a liquid pump system in a nuclear plant is shown in Figure 7. These results were obtained with a two-volume acoustic filter system designed with the electroacoustic analog. The volume diameter was 19.3 in (49 cm), and the length was 4 ft (1.2 m). The choke tube diameter was 0.8 in (2 cm), and its length was 7 ft (2.1 m). The speed of the triplex (2-cm) pump was 360 rpm, and the filter resonant frequency was set at 8.1 Hz for a velocity of sound of 4550 ft/s (1390 m/s).

Piping Vibrations When mechanical resonances are excited by pulsations, vibrations in the pump and piping can sometimes be 20 times higher than under off-resonant conditions. When the mechanical resonances coincide with the acoustic resonances, an additional amplification factor as high as 300 can be encountered.

Piping system mechanical natural frequencies can be calculated using simplified design procedures to provide effective detuning from known excitation sources. A nomogram (see Figure 8) for calculating the lowest natural frequency of uniform steel piping spans[11] can be used in designing piping systems and in diagnosing and solving vibration problems. For example, welded between two bottles, a 4-in (102-mm) pipe that is 10 ft (3 m) long and has an inner diameter of 3.826 in (97.18 mm) would have a mechanical frequency of 74 Hz. If the pipe was an equal-leg L bend ($L = 5$), the natural frequency would be 50 Hz.

To minimize piping vibration problems, all unnecessary bends (considering routing and thermal flexibility) should be eliminated because they provide a strong coupling point between pulsation excitation forces and the mechanical system. When bends are needed, use the largest enclosed angle possible and locate restraints near each bend. Piping should also have supports near all piping size reductions and at large masses (valves, accumulators, flanges, and so on). Small auxiliary piping connections (vents, drains, pressure test connections, and so on) should be designed so the mass of the valve and flange is effectively tied back to the main piping, thus eliminating relative vibration.

Diagnostics and Instrumentation The diagnosis of vibrations in positive displacement pumps should usually include dynamic pressure measurements in the cylinders and piping near the pump. These measurements can be obtained by the use of piezoelectric or strain gage pressure transducers. If cavitation or flashing is suspected, a pressure transducer capable of measuring static and large dynamic pressures should be used; even then, cavitation-produced pressure shocks may damage the transducer.

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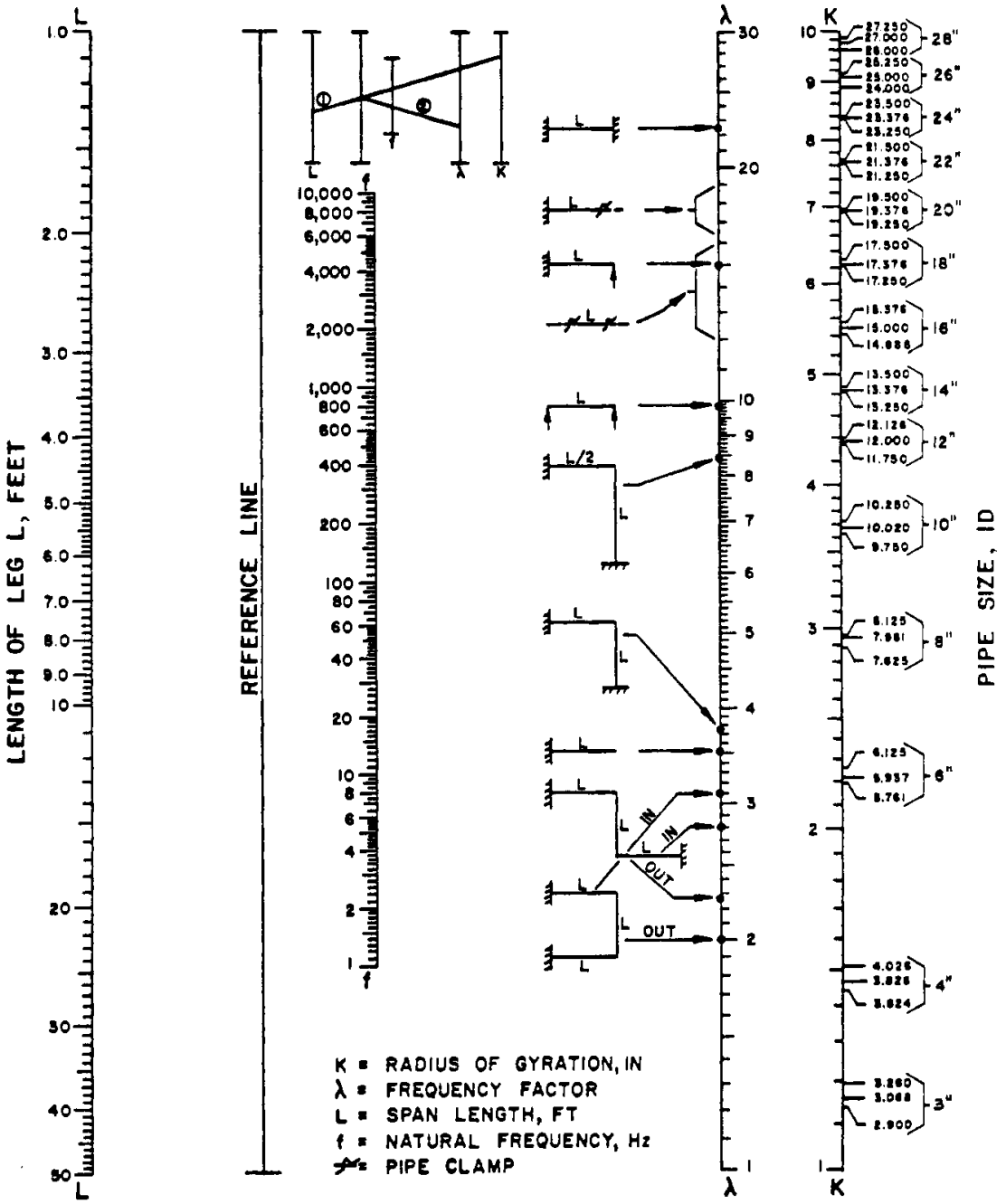


FIGURE 8 Natural frequency of uniform steel piping spans (1 ft = 0.3048 m; 1 in = 2.54 cm)

Accelerometers with low frequency characteristics may be used with electronic integrators to obtain accurate vibration displacement data from the pump case, cylinders, or piping. Similarly, velocity or seismic pickups may be employed. Maximum vibrations usually occur at the middle of piping spans and at unsupported elbows (out of plane).

Real-time analyzers and oscilloscopes may be used to display the resulting signals. A field example showing the diagnosis of cavitation at the pump suction using a strain gage diaphragm pressure transducer is given in the oscilloscope trace of Figure 9. The vapor pressure (gage) for this system was 25 lb/in^2 (172 kPa). Note that the negative half of the cycle is flattened when vapor pressure is reached and that very high amplitude pressure spikes are apparent. For liquids with dissolved gases, lower pulsation amplitudes can produce cavitation; however, the cavitation is usually less severe.

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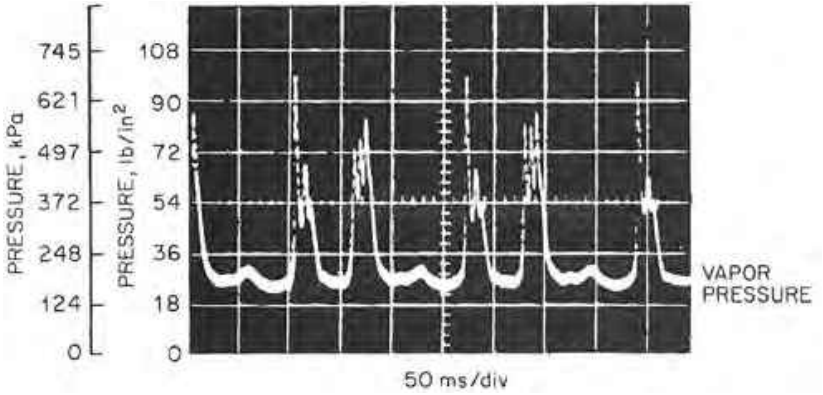


FIGURE 9 Complex wave data showing cavitation effects of pressure wave in a liquid piping system

Typical field pulsation data obtained on a three-plunger pump are shown in Figure 10, which is a frequency spectrum of the pulsations made by a real-time analyzer. Each spike represents a frequency multiple of running speed. Note that the third spike, representing the plunger frequency, has the largest amplitude; however, the components at one and two times pump speed are also significant, which means that these frequency components should be considered for pulsation control.

The vibration frequencies of the piping should be compared with pulsation frequencies to evaluate potential pulsation excitation of mechanical resonances. A check of the piping mechanical natural frequencies from the nomogram (refer to Figure 8) should be made to evaluate the possibility of a mechanical resonance. High vibrations produced by low-level pulsations at a particular frequency are indicative of a mechanical resonance, which can usually be corrected by additional piping restraints or snubbers. After the causes of the pulsations and vibrations are diagnosed, the techniques presented previously can be used to develop solutions.

Thermal Problems In pump systems with high thermal gradients, large forces and moments on the pump case can cause misalignment of the pump and its driver as well as pump case distortion resulting in vibrations, rubbing (wear), bearing failure, seal leakage, and so on. High stresses can be imposed on the piping, resulting in local yielding or damage to the piping restraints, snubbers, or support system. Misalignment problems commonly exhibit a high second-order component of the shaft vibrations. Proximity

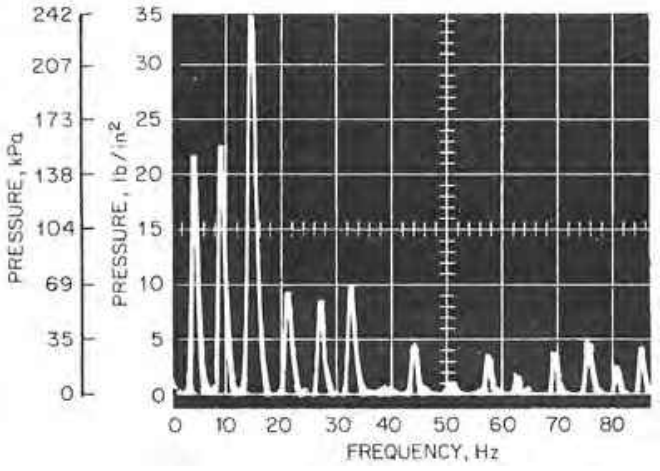


FIGURE 10 Typical field data recorded on a three-plunger pump

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probes can be used at the bearings to measure movement of the shaft relative to the bearing centerline.

Diagnosis of Shaft Failures Pump and driver shafting can experience high stresses during start-up and normal operation because of the uneven torque loading of the positive displacement pumping action. Shaft failures are strongly influenced by the torsional resonances of the system, which are the angular natural frequencies of the system.

Torsional vibrations can be measured using velocity-type torsional transducers that mount on a stub shaft. Alternatively, they may be gauged by measuring the gear tooth passing frequency with a magnetic transducer or proximity probe and using frequency-to-voltage converters to give the change in tooth passing frequency (the torsional vibrational velocity). Spectral analysis of these signals defines the torsional amplitudes and natural frequencies. The stresses can be calculated by using the mode shape of the specific resonant natural frequency and combining all the torsional loads. Torsional natural frequencies, mode shapes, and stresses can be calculated by using either the Holzer technique or digital computer programs.[12]

Torsional problems can usually be solved by changing the coupling stiffness between the driver and pump or by using a flywheel in an effective location. The addition of a flywheel will tend to smooth the torque oscillations. Pumps with a greater number of cylinders and equal cylinder phasing usually operate more smoothly with lower shaft stresses.

REFERENCES

1. Von Nimitz, W. W. "Reliability and Performance Assurance in the Design of Reciprocating Compressor and Pump Installation." 1974 Purdue Compressor Technology Conference.
2. Sparks, C. R., and Wachel, J. C. "Pulsation in Centrifugal Pump and Piping Systems." *Hydrocarbon Processing,* July 1977, p. 183.
3. Hayward, A. T. J. "How to Estimate the Bulk Modulus of Hydraulic Fluids." *Hydraulic Pneumatic Power,* Jan. 1970, p. 28.
4. Wright, W. A. "Prediction of Bulk Moduli and Pressure-Volume-Temperature Data for Petroleum Oils." *ASLE Transactions 10*:349, 1967.
5. API Technical Data Book, *Petroleum Refining,* 2nd ed., Washington, D.C., 1972.
6. Klaus, E. E., and O'Brien, J. A. "Precise Measurement and Prediction of Bulk-Modulus Values for Fluids and Lubricants." *Journal of Basic Engineering,* September 1964, p. 469.
7. Noonan, J. W. "Ultrasonic Determination of the Bulk Modulus of Hydraulic Fluids." *Materials Research and Standards,* December 1965, p. 615.
8. Hicks, E. J., and Grant, T. R. "Acoustic Filter Controls Reciprocating Pump Pulsations." *Oil and Gas Journal,* January 15, 1979, p. 67.
9. American National Standard for Reciprocating Power Pumps for Nomenclature, Definitions, Application and Operation, ANSI/HI 6.1-6.5-2000, Hydraulic Institute, Parsippany, NJ www.pumps.org.
10. *Machine Design,* Fluid Power Reference Issue, vol. 52, no. 21, Penton Publications, Cleveland, 1980.
11. Wachel, J. C., and Bates, C. L. "Techniques for Controlling Piping Vibration and Failures." ASME paper 76-Pet-18, 1976.
12. Szenasi, F. R., and Blodgett, L. E. "Isolation of Torsional Vibrations in Rotating Machinery." *Proceedings of the National Conference on Power Transmission,* vol. II, Illinois Institute of Technology, 1975.

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FURTHER READING

Positive Displacement Pumps: Reciprocating, API Standard 674, 2nd ed., 1995, American Petroleum Institute, Washington, D.C. www.api.org.

Positive Displacement Pumps: Controlled Volume, API Standard 675, 2nd ed., 1994, American Petroleum Institute, Washington, D.C. www.api.org.

SECTION 3.5
DISPLACEMENT PUMP FLOW CONTROL

WILL SMITH

FLOW CONTROL IN INDIVIDUAL PUMPS

An inherent characteristic of positive displacement pumping of relatively incompressible liquids is that flow rate is proportional to displacement rate and independent of pressure levels. The capacity of a centrifugal pump operating at constant speed varies from a maximum flow at no developed pressure to zero flow at a definite limiting pressure known as shutoff head. The *average* capacity of positive displacement pumps at constant speed is, within design limits of pressure, practically constant, even though flow rate pulsations do occur as individual displacements are forced into the discharge pipe.

Flow control of positive displacement pumps is accomplished by

- Changing the displacement rate
- Changing the displacement volume
- Changing the proportion of the displacement delivered into the piping system

Throttle Control in Direct-Acting Steam Pumps Direct-acting pumps are controlled by speed change, which is effected by throttling the flow of steam (or other motive gas) to the drive cylinder. The magnitude of excess force on the drive piston over that required for the driven, or pumping, piston to force pumpage into the piping system dictates the stroke rate and therefore the capacity of the pump. A sensor that detects the desired result of pumping (pressure, level, flow rate, and so on) may be used to modulate the steam throttle valve.

Speed Control in Power-Driven Pumps Speed modulation is the most common means of flow control for power-driven pumps. The most rudimentary speed control is intermittent (start-stop) operation. The average capacity over relatively long time periods depends upon the percentage of time the pump operates at 100% versus the percentage of time it

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operates at zero flow. Of course, consideration must be given to the frequency of starts because electric motors may overheat if there is insufficient time for cooling after the inrush of starting current.

When continuous modulation of flow is desired, means of varying the driver speed or the ratio of speed reduction between the driver and pump are employed.

DRIVER SPEED CONTROL

1. Multispeed constant-torque motors offer limited steps in speeds.
2. Variable-frequency motor controls permit a wide range of fully modulated speeds but, at present, are limited in maximum horsepower (Section 9.2.2).
3. Direct current and wound rotor alternating current motors permit modulation of speeds within definite limits of torque capability (Section 9.2.2).
4. Gasoline and diesel engines provides speed variation capabilities within the limits where driver torque is adequate to satisfy the constant-torque requirements of the pump (Section 9.1.3).
5. Steam or gas turbines can operate over a limited speed range within their particular output torque limits (Sections 9.1.2 and 9.1.5).

SPEED CHANGER CONTROL Because *most* positive displacement pumps operate at significantly lower speed than *most* drivers, a speed reduction unit is coupled between driver and pump. Capacity control by varying the ratio of the speed reduction is quite convenient. A number of methods are used.

1. *Variable-ratio belt drives* change the speed by changing the pitch diameter of driver and driven sheaves in response to capacity requirements.
2. *Hydraulic torque converters* (hydrodynamic drives) vary the speed by regulating the *amount* of active drive fluid in the coupling. Because these converters relate torque to speed, they are sensitive to the driven torque requirement change that may derive from system pressure changes (Section 9.2.3).
3. *Hydroviscous drives* vary the slip between input and output shafts by adjusting the *distance* between elements in a viscous fluid environment (Section 9.2.3).
4. *Eddy-current couplings* also vary the slip between driver and driven elements but do so by varying the strength of a magnetic field (Section 9.2.1).

Variable Displacement Flow Control Rather than by speed control (changes in the number of displacements in a given period of time), flow may be varied by changing the *volume* displaced per stroke.

Variable strokes are generally limited to small metering or "controlled volume" pumps. Although large-capacity variable-stroke pumps have been built, the complexity of the mechanism involved and the development of other effective and economical means of flow control have precluded their use.

Changing Delivery to the System Another means of capacity control is *not* to vary the pump capacity but to alter the amount of pumpage delivered to the system. There are two popular means of doing this.

SUCTION VALVE UNLOADING Suction valve unloaders cause the displacement to be ineffective during the strokes for which the unloaders are activated. They command either full or zero delivery while the pump is kept running. Thus, current inrush problems from frequent motor starts are avoided. Because discharge pressure is not developed when suction valves are unloaded, energy consumption is held to a minimum.

A pump valve is unloaded by mechanically preventing the valve plate from returning to its seat. If the suction valve is held away from its seat, liquid will ebb and flow through the valve from suction header to pump cylinder and from cylinder to header during the stroking of the plunger or piston.

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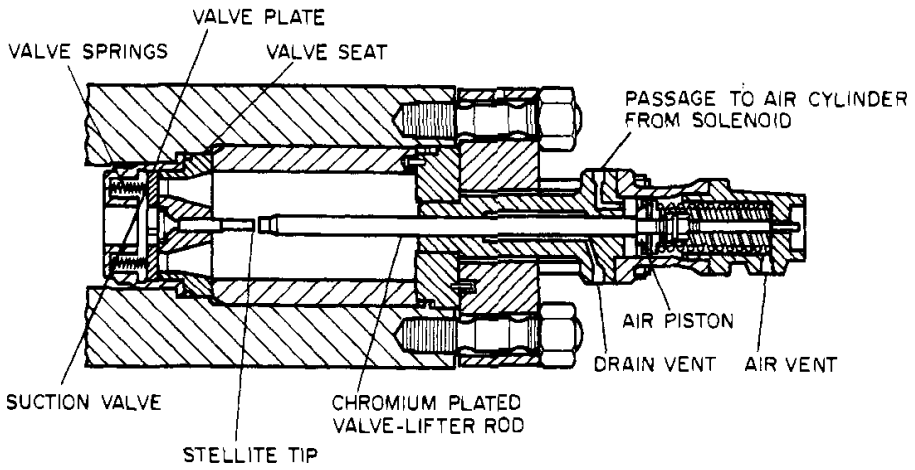


FIGURE 1 Synchronized unloading keeps the valve open through the use of a suction valve unloader

The unloader mechanism does not move the valve. The valve is lifted off its seat by the pressure differential created by normal pump operation. In synchronized unloading, the suction valve unloader (Figure 1) is retracted only when it is certain that the incoming flow will keep the valve open.

The unloader never resists pressure in the cylinder; it merely holds the valve plate against valve springs that had already been compressed and moves away from the valve while the springs are still compressed. This permits the valve to close when the plunger begins its discharge stroke.

For smooth transition between full flow and zero flow, the unloading and loading are accomplished in an immediate and identical sequence of the pumping order of the plungers. It is the function of the synchronized suction valve unloader distributor (Figure 2) to control unloading events in that proper and immediate sequence. Similarity can be drawn to the gasoline engine distributor in its function of assuring the proper and immediate sequence of firing of the spark plugs.

The distributor is driven directly by the crankshaft so its position is positively indexed to the stroking of the plungers. It provides an electrical signal to a solenoid valve, which

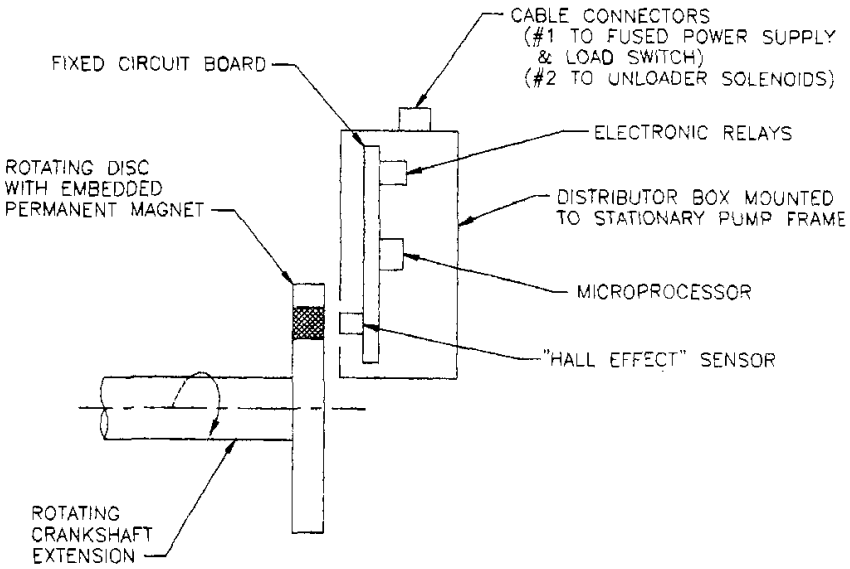


FIGURE 2 "Hall effect" microprocessor distributor system

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admits motive fluid pressure to the unloader chamber when the suction valve to the particular cylinder has been opened. Then, in sequence, each of the remaining suction valves is retained so, in the first revolution of the pump after a control signals the distributor to initiate unloading, all suction valves are unloaded and the pump capacity is reduced to zero. When the control again signals for capacity, each suction valve is released, in sequence, during a single revolution of the crank.

While unloaded, the discharge valves do not function; they remain closed and serve as check valves against the pressurized discharge system. The suction valves remain open. Liquid enters from and is dispelled back into the suction manifold through the open suction valves. Therefore, no work is applied to the liquid, not even that required to open the suction and discharge valves. When the control calls for loading, the load is applied in steps equal to the number of plungers. Thus the pumping horsepower, as well as the liquid inertia in the pumping system, is changed in a number of steps equal to the number of plungers over the time span of one crank revolution.

The unloader consists of a spring-loaded pneumatic cylinder attached to the suction valve cover. The spring advances holding fingers that keep the valve away from its seat. Pneumatic pressure, applied through a solenoid valve in response to the distributor signal, creates a force on a piston that opposes the spring force and allows the suction valve to load (close). The system is normally arranged to fail in the unloaded position on either electrical or pneumatic pressure failure.

Distributor systems have evolved from a) direct mechanical linkages between reciprocating parts and the suction valves, to b) electromechanical interfacing, to c) electronic proximitor systems, to d) a "Hall effect" sensor signaling a microprocessor. The objective of each of these systems is to cause suction valve unloader solenoids to activate or deactivate in the appropriate sequence and at the appropriate position of the plunger during the suction stroke.

BYPASS CONTROL Ultimate delivery of pumpage to the system may be controlled by a discharge-to-suction bypass valve. Pump operation is continuous, with the possibility of modulating the flow to the system by regulating the portion "bypassed" to suction. A serious disadvantage is the full power utilization and full-load wear and tear even when at zero system capacity.

Evaluation of Flow Control Options Proper selection of the means of capacity control requires consideration of system control requirements and then evaluation of the options available to satisfy those requirements.

Precision, responsiveness, and degree of modulation required may limit options. First cost, maintenance requirements, and the effect on pump operating cost should then be weighed for viable options. To evaluate the effect of operating costs, it is necessary to determine (or reasonably assume) the expected pattern of operation throughout the capacity range. For example, if practically all operation except infrequent start-ups is to be at full pump capacity, little penalty should be assigned to the full power usage of a bypass valve or to the part-load losses inherent with hydroviscous or eddy-current drives.

If smooth modulation of flow is not essential, start-stop or suction valve unloading may be the preference. If smooth modulation *and* considerable operation at part capacity are necessary, variable-frequency motors or variable-pitch V-belt drives may be indicated.

Table 1 and Figure 3 provide some general guidelines. Each application should be evaluated on the merits of requirements and options.

FLOW CONTROL IN COMBINED DISPLACEMENT AND CENTRIFUGAL PUMPS

Systems that involve both centrifugal and reciprocating positive displacement pumps deserve some special consideration. Centrifugal pumps are often used as suction boosters to overcome acceleration head requirements peculiar to reciprocating pumps but are rarely used to supplement flow. Some unique characteristics of each type of pump that affect the other type must be considered in the design, operation, and control or the interrelated system.

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TABLE 1 Comparison of several capacity control schemes for positive displacement pumps

Control Method	Degree of Modulation	First Cost	Operating Cost	Comments
		Directing-acting pumps		
Steam throttle	Full: 0 to 100%	Low	Low	Steam pressure required to balance liquid piston force and overcome breakaway friction. Steam volume throttled to produce desired capacity.
		Power pumps		
Start-stop	0 *or* 100%	Low	Low	Limited in frequency of starts because of temperature rise from inrush.
Multispeed motors	Steps dependent on motor winding	Medium	Low	Cost of motor controller switch gear motors must be assessed.
Variable frequency	Full: 0 to 100% +	High	Low	Limited by current-handling capacity of solid-state controller.
Direct current	Full: 0 to 100%	High	Medium	Drive is torque-speed sensitive, pump is torque-pressure sensitive at all speeds. Check drive for required torque at minimum and maximum speeds.
Wound rotor motors	Full: 0 to 100%	High	Low	Drive is torque-speed sensitive. Pump is torque-pressure sensitive at all speeds. Check drive for required torque at minimum and maximum speeds.
Combustion engines	Variable for limited range	High	Low	Torsional analysis required to avoid high torsional stresses.
Steam or gas turbines	Variable for limited range	Medium	Medium	Drive is torque-speed sensitive. Pump is torque-pressure sensitive at all speeds. Check drive for required torque at minimum and maximum speeds.
Hydraulic torque converter	Full: 0 to 100%	High	High	Low full speed efficiency.

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TABLE 1 Comparison of several capacity control schemes for positive displacement pumps (*Continued*)

Control Method	Degree of Modulation	First Cost	Operating Cost	Comments
		Power pumps		
Hydroviscous speed control	Full: 0 to 100%	High	High at part load	Efficiency proportional to capacity. Loss to heat exchanger.
Fluid coupling	Full: 0 to 100%	Medium	High	Full-speed slip loss greater than hydroviscous. Loss to heat exchanger.
Magnetic (eddy) coupling	Full: 0 to 100%	High	High	Full-speed slip loss.
Suction valve unloaders	0 *or* 100%	Low	Low	Synchronization with suction stroke avoids start-stop and shock problems.
Bypass valve	Limited	Low	High	Uses full power at zero capacity with full valve wear.
Variable-pitch belt	Limited	Low	Low	Limited to belt drive horsepower.

Pertinent Positive Displacement Pump Characteristics

1. The discharge pressure produced is a function of the system requirement only and is independent of pump capacity.
2. The flow rate pulses between maximum and minimum values for each revolution of the crankshaft.

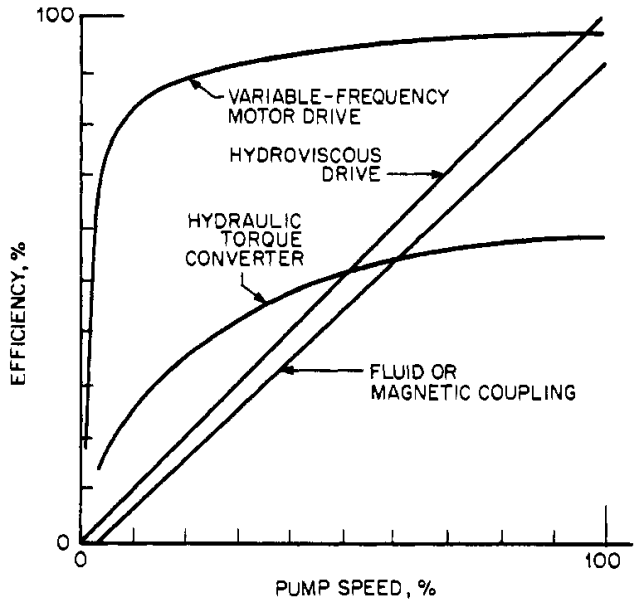


FIGURE 3 Relative efficiencies of speed control options

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3. The pulsating flow imposes an acceleration head that adds to the net positive inlet pressure required (Section 3.1).
4. Being more energy-efficient, the positive displacement pump is normally the lead pump and the centrifugal pump is the supplement when operated in parallel.

Pertinent Centrifugal Pump Characteristics

1. The dynamic head (or pressure rise) produced is a function of pump capacity as well as of system requirements.
2. For satisfactory operation, the flow must be kept within a limited range of the best efficiency capacity (Section 2.1.3).
3. When used as a suction booster, the centrifugal pump must be designed so it cannot introduce air into the gas-intolerant reciprocating pump. (Gas in positive displacement pumps, like liquids in positive displacement compressors, may cause severe hydraulic and mechanical shock.)
4. Centrifugal pumps do not generate acceleration heads that impose on net positive suction head required. However, if a centrifugal pump is connected in parallel to a common suction line with a reciprocating pump, some of the system acceleration head loss from the positive displacement pump may affect the centrifugal pump.

Only two cases need to be considered: centrifugal pumps feeding (1) in series into reciprocating pumps to increase suction NPSHA to the reciprocating pump or (2) in parallel to augment the delivered capacity. It would be most unusual to encounter a positive displacement pump feeding into the suction of a centrifugal pump because of the high pressure that could be imposed on the centrifugal pump suction and because of the amplification of flow pulsations resulting from interaction of the characteristics of the two pumps, which could be deleterious to both pumps.

Series Operation, Suction Boost The flow rate is always determined by the positive displacement pump capacity and is independent of the centrifugal booster pump. The capacity is therefore controlled as with any positive displacement pump.

The purpose of the centrifugal pump is to supply sufficient pressure to satisfy the suction requirements of the positive displacement pump. The centrifugal pump must be sized to provide the maximum rate of flow of the positive displacement pump at a pressure high enough to meet the NPSH requirement of the latter.

With a flow-controlled positive displacement pump, speed control of the booster might be considered, as illustrated in Figure 4. The system pressure requirement can be satisfied by the positive displacement pump at all flow rates; therefore, total energy requirements may be reduced by using the less efficient centrifugal pump to develop no more than the required suction head.

If the centrifugal booster pump is run at constant speed, its share of the total head requirement will increase as system capacity is reduced.

Parallel Operation To increase the capacity of an existing pumping system, it is sometimes convenient to consider paralleling a positive displacement pump with a centrifugal pump. The interactions of the two pumps and the system must be considered and controlled for satisfactory operation.

Suction stabilizers and discharge dampeners, desirable for most reciprocating pump installations, become mandatory when a centrifugal pump and a positive displacement pump are connected in parallel. These devices are used to isolate, as much as possible, the effects of positive displacement pump flow pulsations from the sensitive centrifugal pump operating characteristics.

Centrifugal pumps in parallel with positive displacement pumps can supplement the system capacity only while they generate sufficient discharge pressure to satisfy the system requirement at the higher total flow. Undampened flow pulsations shift the centrifugal shutoff flow point relative to system flow at the frequency of the reciprocating

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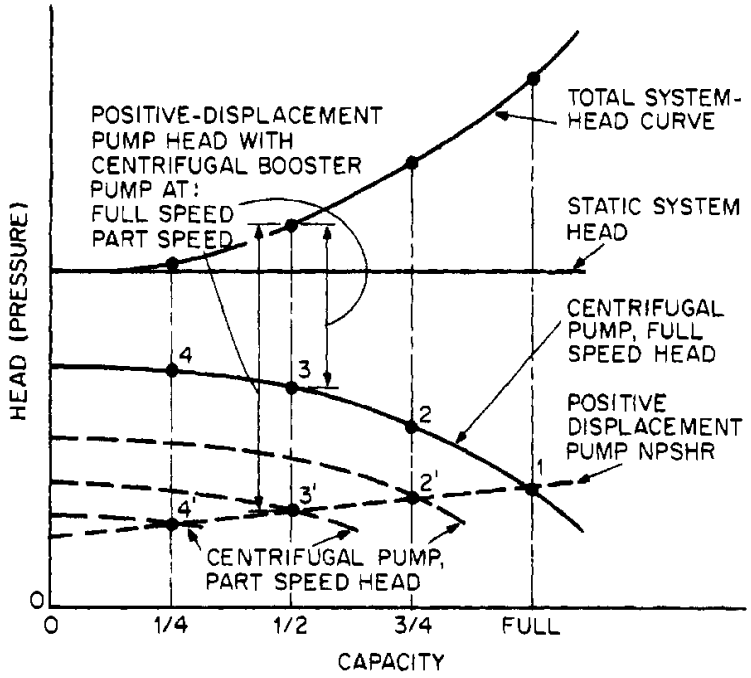


FIGURE 4 A centrifugal booster pump can satisfy the NPSH requirement of a positive displacement pump by operating either on its characteristic full speed curve 1-2-3-4 or with speed reduction along curve 1′-2′-3′-4′

pulsation. From the system standpoint, the total flow varies along curve 1-2-3-2-1 in Figure 5. From the centrifugal pump standpoint, it varies along its characteristic curve, 1′-2′-3′-2′-1′. The reciprocating pump operates along curve 1″-2″-3″-2″-1″.

For purely static system-head curves, the centrifugal pump theoretically would not "sense" the pulsating flow. For dynamic system heads, the centrifugal pump flow varies

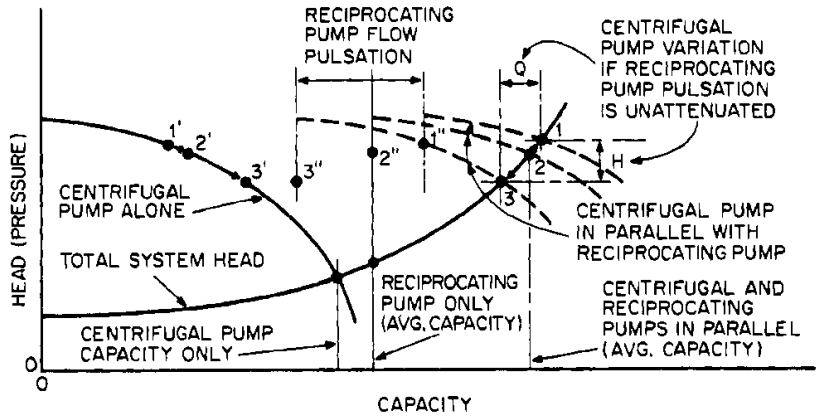


FIGURE 5 Parallel operation of undampened reciprocating and centrifugal pumps

1. The reciprocating pump will operate at a capacity corresponding to its speed regardless of its discharge pressure.
2. If the system head is only static (constant head), the centrifugal pump capacity is constant and the system flow variation corresponds to the reciprocating pump variation.
3. If the system head is partially dynamic (as illustrated), the centrifugal pump will experience flow variations.
   a. The flatter the centrifugal pump curve, the greater the capacity and smaller the head variations on the centrifugal pump.
   b. The flatter the system curve, the lower the capacity and greater the head variations on the centrifugal pump.

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contracyclically with the reciprocating pump pulsations. The magnitude of the centrifugal pump flow variation due to the positive displacement pump pulsations is a function of the system head curve, the centrifugal pump head-capacity curve, the degree of flow pulse imposed on the centrifugal pump discharge header, and the liquid bulk modulus (or compressibility).

FLOW CONTROL IN POSITIVE DISPLACEMENT PUMPS IN SERIES

Whenever positive displacement pumps are installed in series (such as for multiple-station pipeline service), some variable-capacity capability is mandatory. Otherwise, any variation in capacity between series pumps or series groups of pumps would result in disastrous interstation pressures.

The flow is determined by the first pump in the line. Subsequent pumps should be equipped with capacity controls responsive to their inlet pressures. The preferred control is a continuous-modulation type, such as variable speed, unless large reservoir capacity is available between pumps.

Recirculation from discharge to suction through a throttling bypass valve may be used, at the expense of energy, to match the delivery from downstream pumps to upstream pumps. Excessive bypass flow in relation to through flow can result in significant temperature rise of the liquid.

When downstream pump capacity is less than the upstream delivery, the interpump pressure will rise, indicating a need for increased capacity. When the downstream pump capacity exceeds the upstream delivery, the downstream pump will "draw down" the interpump pressure to the point where net positive inlet pressure difficulties are ensured.

Reservoir accumulators between pumps permit a limited degree of mismatch in capacities, allowing the use of incremental capacity controls, such as suction valve unloading and intermittent pump bypass.

In any event, effective discharge dampeners on the upstream pump and suction stabilizers on the downstream pump are necessary to prevent the inherent pressure pulsations and control modulation variances from adversely affecting operation.

FLOW CONTROL IN POSITIVE DISPLACEMENT PUMPS IN PARALLEL

Because the capacity of positive displacement pumps is virtually independent of head, there is no problem from the head-capacity characteristics in parallel operation—even for pumps of different sizes.

Particular attention to suction acceleration head requirements is necessary when more than one pump is taking suction from a common suction pipe. Because it cannot be assured that two pumps will not, at some point, have simultaneous suction strokes, the acceleration head requirements of each pump must be added for that portion of the suction line that furnishes the total flow. This suggests extra large suction lines, individual ones from the liquid source, or suction stabilizers sized for the total flow with individual outlet connections for each pump.

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SECTION 3.6
DIAPHRAGM PUMPS

STEPHEN D. ABLE
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Diaphragm pumps are a class of displacement pumps featuring flexible membranes in combination with check valves that are used to move fluids into and out of pumping chamber(s). These pumps are used extensively in transfer and metering applications requiring flows of up to 300 gallons per minute (1150 liters per minute). They are quite versatile, handling a wide variety of fluids including chemicals, dry powders, food additives, glues, paints, pharmaceutical products, slurries, tailings, and wastewater. A distinguishing feature of all diaphragm pumps is the absence of seals or packing, meaning they can be used in applications requiring zero leakage. There are three main categories of diaphragm pumps: 1) mechanically driven, 2) hydraulically driven, and 3) air-operated.

MECHANICALLY DRIVEN DIAPHRAGM PUMPS

Many industries are served by mechanically driven diaphragms pumps. They are used in construction, chemical, and water treatment applications.

Construction Industry Mechanically driven diaphragm pumps are widely used in the construction industry for dewatering applications where pumps may ingest rocks or other debris. A popular make of this type of pump contains a spring on the plunger rod (see Figures 1 and 2). If the operating pressure exceeds the maximum recommended pumping pressure, the spring compresses and does not move the diaphragm. The spring can compress and thus keep a rock from being pushed through the wall of the pumping chamber or cause the drive mechanism to fail.

In single-diaphragm pumps, the pumped liquid can have a lot of inertia if the suction and discharge lines are relatively long. A simple accumulator on the suction (inlet) side of the pump enables the pump to draw liquid from the accumulator while it simultaneously draws liquid through the suction line.

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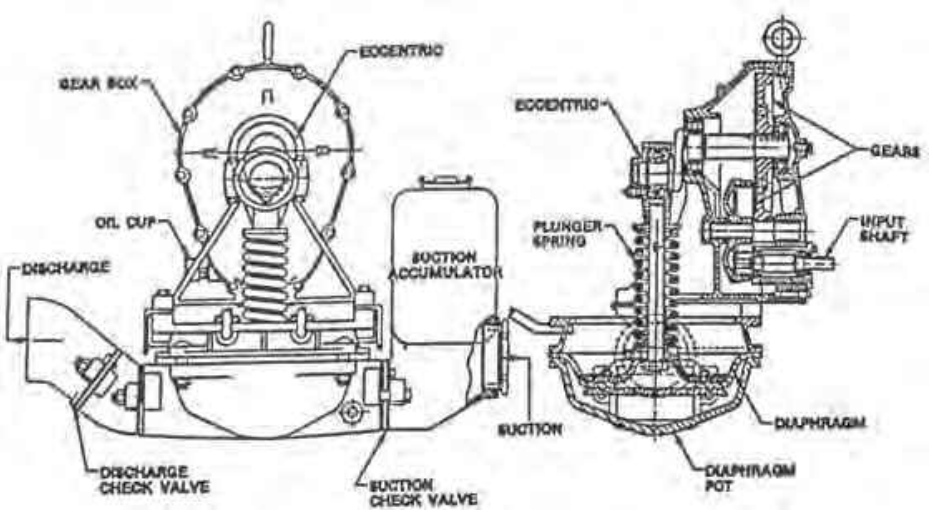


FIGURE 1 Cross-section of a mechanically driven single-diaphragm pump for the construction industry (Gorman-Rupp)

During the discharge stroke, the accumulator can refill with liquid from the suction line. If the discharge line from the pump is relatively long, the inertia of the liquid can be great, as mentioned earlier, and can impose severe loads on the diaphragm and drive mechanism. The spring on the plunger rod can absorb some of the drive energy early in the discharge stroke and "give it back" during the latter part of the discharge stroke, greatly reducing the inertia loading on the diaphragm and drive mechanism.

Mechanically driven diaphragm pumps in the construction industry operate by a reciprocating plunger, usually secured to plates on both sides of the diaphragm. The diaphragms are customarily fabric-reinforced elastomers (usually synthetic rubbers) similar in many ways to the fabric-reinforced materials used in pneumatic tires. The diaphragms are normally molded with a convoluted section between the central clamped area and the clamped periphery. This convoluted section permits longer strokes than would be possible otherwise.

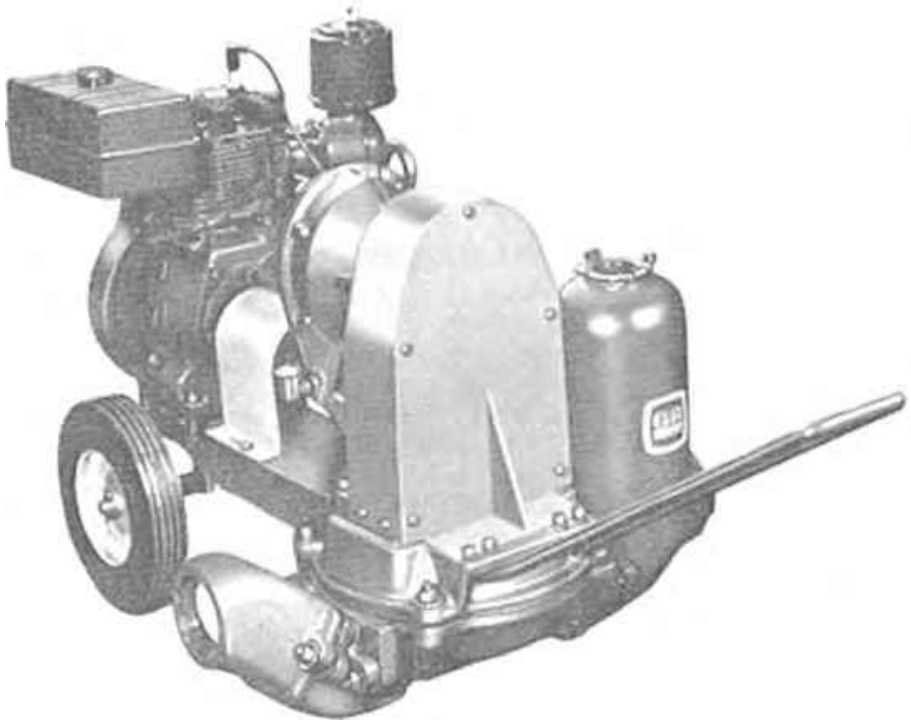


FIGURE 2 Mechanically driven single-diaphragm pump, engine-powered (Gorman-Rupp)

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These pumps are sometimes duplexed so that the reciprocating means acts alternately on two diaphragms with one on a suction stroke, while the other is on a discharge stroke and vice versa. A connector called a *walking beam* is pivoted between two diaphragms. As one diaphragm is pushed down on a discharge stroke, the other diaphragm is simultaneously pulled up on a suction stroke. The pumping chambers with inlet and outlet check valves are manifolded together to a common inlet and a common outlet. The principle advantage of the duplex diaphragm pump is its more constant flow (two pressure pulsations per cycle).

Mechanically driven diaphragm pumps are used in the construction industry for dewatering foundations and cofferdams, as well as in sewage treatment plants for pumping lime slurries. They are normally limited to 50 ft (15.2 m) of differential head and are capable of suction lifts of as much as 25 ft (7.6 m).

Chemical and Water Treatment Industries Another type of mechanically driven diaphragm pump is used for the injection or transfer of chemicals into process streams at pressures up to 250 lb/in^2 (17 bar). These pumps are designed to enable the easy adjustment of their capacities, so precise volumes of chemicals can be injected. Typically, capacities can be adjusted through a 20:1 range. Injection repeatability is generally plus or minus 3%.

A wide range of chemicals can be handled. Wetted materials include PVC, PVDF, Polypropylene, 316SS, Alloy 20, and Alloy C22. Diaphragms are PTFE or PTFE with elastomeric backing. Ball type check valves are usually employed.

Applications for this type of pump include the injection of acids and bases for pH control, biocides, chlorination, coagulants, and fertilizers. There are two basic configurations for pumps in this class: *electromagnetic pumps* (solenoid) and *motor-driven pumps*.

Electromagnetic (electronic) pumps (see Figure 3) are used in a variety of low-power applications with flows from 0.026 to 26 gallons per hour (0.1 to 100 liters per hour) at pressures up to 250 lb/in^2 (17 bar). These metering pumps employ an electronic control circuit that pulses an electromagnet that, in turn, generates the linear motion of an armature-shaft-diaphragm assembly. Each electronic pulse results in one discharge stroke of the pump. At the end of the stroke, a set of springs returns the diaphragm assembly to its initial position, drawing more fluid into the pump chamber in preparation for the next stroke.

These pumps are inherently safe, as they can be run indefinitely in the stalled condition without damage to the pump or overpressuring most systems. An additional feature of certain electronic pumps is the regulation of pulse strength through electronic power control, which leads to smoother fluid injection. Capacity is usually controlled by adjusting

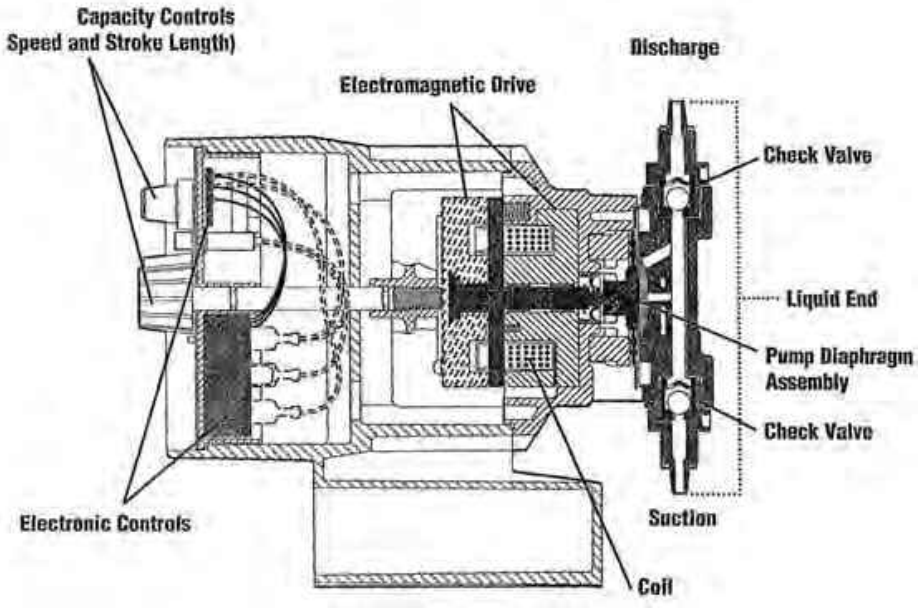


FIGURE 3 Electromagnetic diaphragm pump (Milton-Roy, subsidiary of Sundstrand Corp.)

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the stroke rate, but the stroke length can also be adjusted. Combining these adjustments provides a wide range of outputs.

Advances in the electronic controls have led to the capability to control output manually from 4 to 20 mA process signals, digital pulses from external sources (such as flow meters), or serial data communications signals from computers. Yearly maintenance is recommended for low-pressure applications, but as pressures rise, diaphragms and check valves will need to be replaced more frequently.

Motor-driven, mechanically actuated diaphragm pumps are used in applications for flows from 2 to 300 gallons per hour (approximately 10 to 1000 liters per hour) again at pressures up to 250 lb/in^2 (17 bars). Since motor-driven, mechanically actuated diaphragm pumps are positive displacement pumps, capacities cannot be adjusted by the use of a throttling valve.

Three techniques are used in the industry to adjust motor-driven pump capacities. Some designs employ the use of a "mechanical lost motion" mechanism in which an adjustable mechanical stop interrupts the spring-loaded crosshead from following the cam for a portion of the stroke. This decreases the effective stroke length (see Figure 4). Other designs employ the use of an adjustable crank machined with a "variable eccentricity." The adjustment of the crank's position changes the pump's effective stroke length (see Figure 5). The third design technique is the use of AC or DC variable speed motors.

At higher flow rates, attention should be paid to the system design to ensure the proper handling of pressure pulsations, due to the acceleration and deceleration of the process fluid in the lines. This is especially true for pumps having a mechanical lost motion configuration, due to the diaphragm's rapid starts and stops. In the higher flow systems, pulsation dampeners are frequently employed in the discharge lines and occasionally need to be used in the suction lines to control pulsations. When discharge pressures are low, backpressure valves are employed to provide a system pressure sufficient to decelerate the fluid in the suction line at the end of every suction stroke. If the flow of the process fluid in the suction line has not stopped by the beginning of the discharge stroke, the accuracy of the injection is diminished. Yearly maintenance is recommended for low-pressure applications, but as pressures rise, diaphragms and check valves will need to be replaced more frequently.

Other Mechanically Driven Diaphragm Pumps Automotive fuel pumps are usually of the diaphragm type. The diaphragm is moved mechanically by a cam on a suction stroke

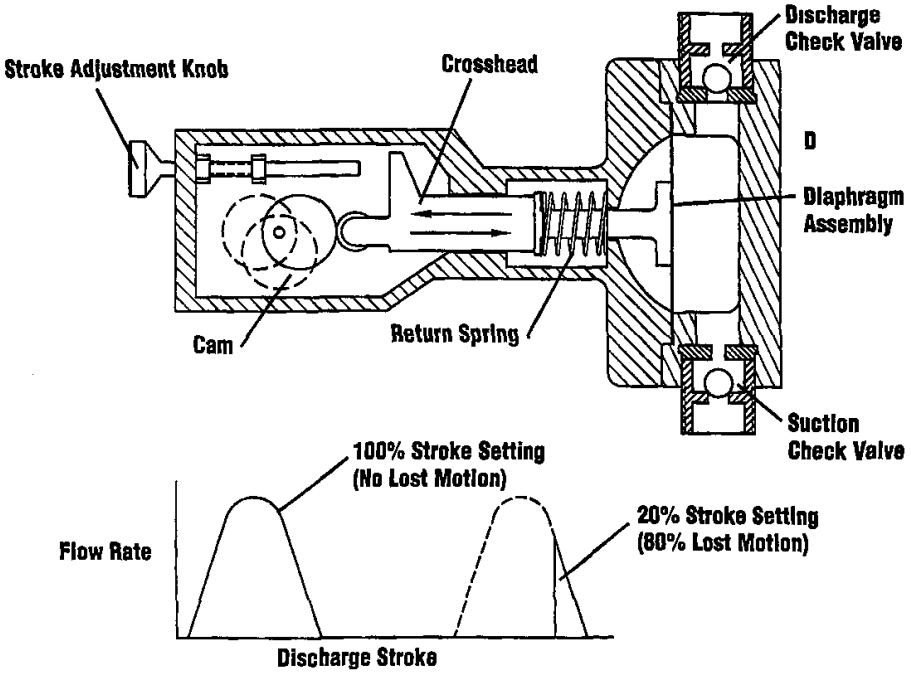


FIGURE 4 Diaphragm of a typical mechanical lost motion mechanism (Milton-Roy, subsidiary of Sundstrand Corp.)

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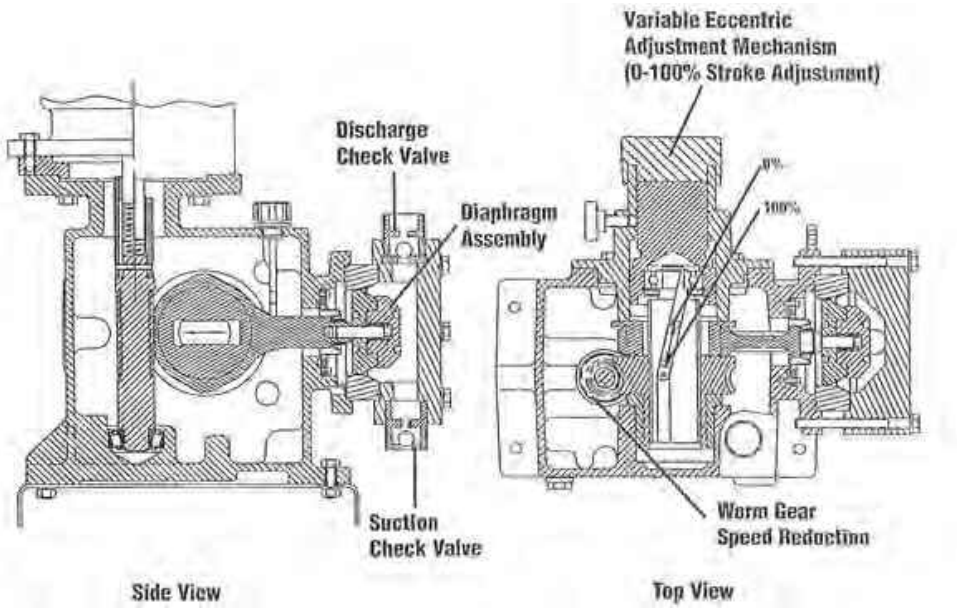


FIGURE 5 Cross-section of a variable eccentric mechanically driven diaphragm pump (Milton-Roy, subsidiary of Sundstrand Corp.)

and is returned by a spring for the discharge stroke. Thus, the spring determines the discharge pressure and provides a nearly constant pump pressure to the carburetor regardless of the pump (engine) speed or the rate of fuel consumption. Hand-operated diaphragm pumps are used extensively as bilge pumps on sailboats where the loss of power is a major concern and hand operation is essential.

HYDRAULICALLY DRIVEN DIAPHRAGM PUMPS

Hydraulically driven diaphragm pumps are used in applications for the transfer or injection of chemicals into process streams at pressures up to 7500 lb/in^2 (approximately 500 bar). Because the diaphragm is pressure-balanced, the stresses in the diaphragms are low. Therefore, these pumps tend to require minimal maintenance. The pump's capacities can be adjusted to match the specific process requirement by adjusting the effective stroke length or stroking speed of the pump. Effective stroke lengths are adjusted by either a hydraulic lost motion, a mechanical lost motion, or by varying the eccentric's offset. The repeatability of the injected flow is plus or minus 1% or better.

Applications range from 0.26 to 26,000 gallons per hour (1 to 100,000 liters per hour). At flows above 26 gallons per hour (100 liters per hour), most pump models employ capacity adjustments based on variable eccentric or variable speed technology to avoid significant pressure spikes due to the rapid acceleration and deceleration of the fluid in the pipes.

As with the mechanical diaphragm pumps, a wide range of chemicals can be handled. Wetted materials include PVC, Polypropylene, PVDF, 316 SS, Alloy 20, Alloy C-22, Titanium, and Inconel. Diaphragms for pressures up to 4350 lb/in^2 (300 bar) are typically composed of PTFE or PTFE with an elastomeric backing. Diaphragms above 4350 lb/in^2 (300 bar) are typically 316 SS, Alloy C, or PEEK. Optional features include fluid temperature control jackets, diaphragm rupture detection capabilities, and remote diaphragm head designs. Typical applications include the injection of acids and bases for pH control, corrosion inhibitors, methanol, coagulants, primary process blending, process slurries, and drag reducers. Three types of liquid ends are used: the disc diaphragm, shown in Figure 6, the tubular diaphragm in Figure 7, and the high performance diaphragm in Figure 8.

The disc diaphragm pump is equipped with process-side and suction-side restraining plates to prevent overdisplacement of the diaphragm during system upsets. When the

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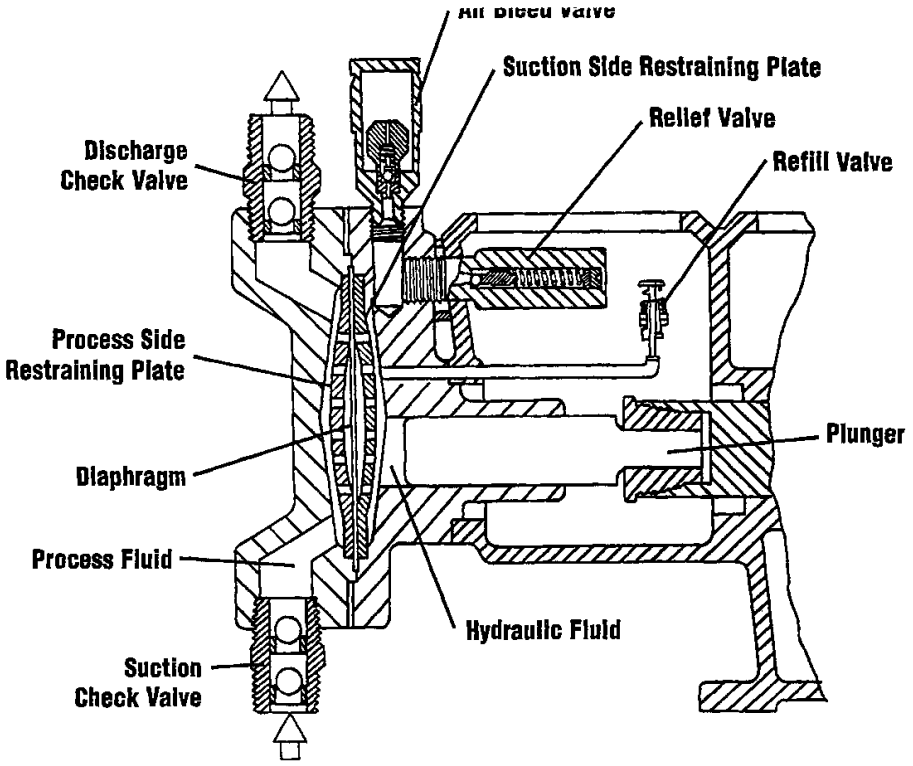


FIGURE 6 Diagram of a disc diaphragm pump (Milton-Roy, subsidiary of Sundstrand Corp.)

diaphragm reaches the suction-side restraining plate, the hydraulic oil pressure drops, causing the refill valve to open and replenish the oil. When the diaphragm hits the process-side restraining plate, the hydraulic pressure rises, causing the relief valve to open, venting some oil. The fluid volume between the restraining plates is typically 150% of the maximum displaced volume of the pump. Therefore, the diaphragm does not contact both restraining plates during the same stroke.

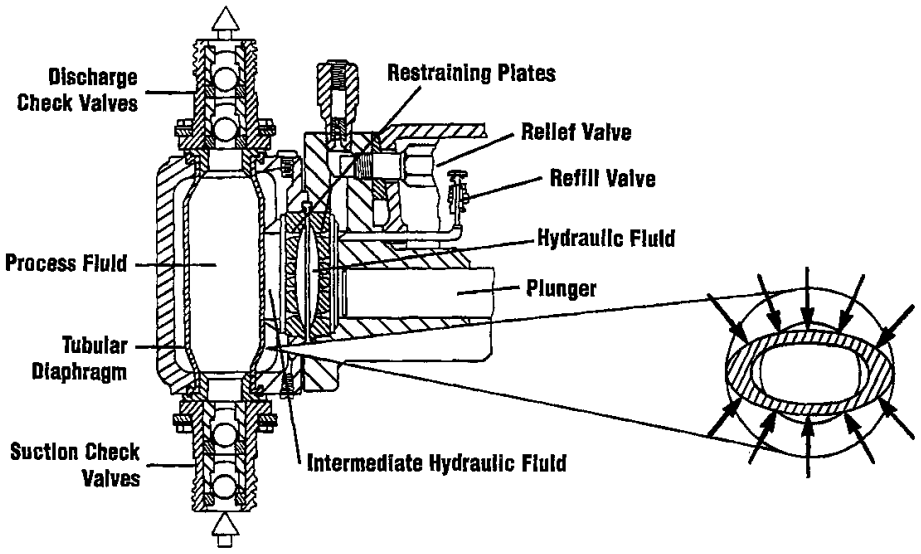


FIGURE 7 Diagram of a tubular diaphragm configuration (Milton-Roy, subsidiary of Sundstrand Corp.)

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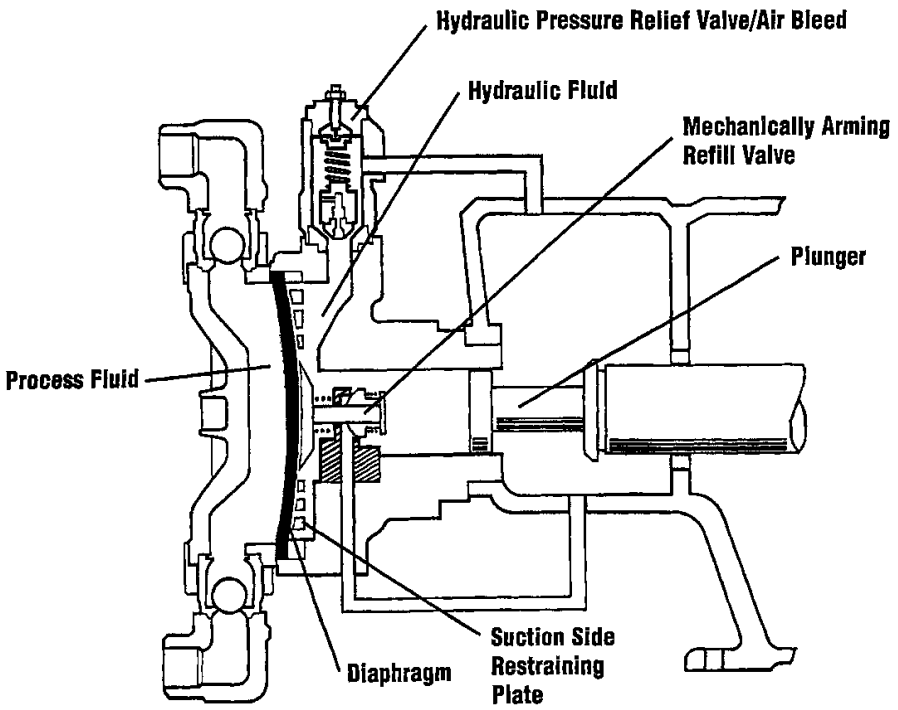


FIGURE 8 Diagram of a high performance diaphragm design (Milton-Roy, subsidiary of Sundstrand Corp.)

The tubular diaphragm configuration is a variation of the disc diaphragm design. A diaphragm shaped in the form of a tube is placed in a chamber in front of the disc diaphragm assembly. This design eliminates the process fluid flowing through the front-restraining plate, reducing viscous losses and wear in case of slurries. The chamber must be filled with a precise amount of hydraulic fluid to avoid overdisplacing the tube.

The high-performance diaphragm configuration eliminates the use of a process-side restraining plate providing the throughflow performance of a tubular design while eliminating the possibility of overdisplacing the tube during startup and maintenance. With a mechanically arming, pressure-sensitive refill valve, the hydraulic fluid can only be replenished when the diaphragm is in the most rearward position. This eliminates the possibility of overfilling the hydraulic chamber and therefore overdisplacing the diaphragm during system upsets (blocked suction or discharge lines).

Most problems with hydraulic diaphragm pumps occur due to incorrect system designs. Pressures above 9 lb/in^2 (0.6 bar) should be maintained in the pump diaphragm heads during the suction stroke to stop vapor buildups in the hydraulic or process-side cavities. Pressures at 3 lb/in^2 (0.2 bar) can be handled in some applications with modified designs and special hydraulic fluids. NPSH calculations should include viscous losses in the check valves and contour plates (if so equipped).

In addition, since hydraulic diaphragm pumps are reciprocating machines, acceleration losses also have to be considered. Peak acceleration/deceleration losses occur at the beginning and end of the stroke, while peak viscous losses occur at midstroke. The losses are not additive. The manufacturer should be contacted to provide guidance in performing these calculations.

As with mechanically driven diaphragm pumps, at higher flows, pulsation dampeners should be considered to ensure the proper handling of pressure pulsations due to the acceleration and deceleration of the process fluid in the lines. This is especially true for pumps having a mechanical lost motion configuration due to the diaphragm's rapid starts and stops. When discharge pressures are low, backpressure valves are employed to provide a system pressure sufficient to decelerate the fluid in the suction line at the end of every suction stroke. If the flow of the process fluid in the suction line has not stopped by the beginning of the discharge stroke, the accuracy of the injection is compromised.

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AIR-OPERATED DIAPHRAGM PUMPS (AODPS)

In general, diaphragm pumps of all types are sealless, have no dynamic seals or packing, are self-priming, and have an infinitely variable flow rate and pressure rate within the pressure and capacity ranges of the pump. *Air-operated diaphragm pumps* (AODPs) can also run dry indefinitely, and the discharge can be throttled to zero flow indefinitely.

The most common types of AODPs are the double-diaphragm pumps (duplex pumps). These contain two diaphragm chambers and two flexible diaphragms. The diaphragms are connected to each other through a connecting rod and are clamped at the outer edges of the diaphragm. The shaft-connected diaphragms move in the same linear direction simultaneously. Compressed air directed to the back side of the left diaphragm moves both diaphragms to the left, while air is exhausted to the atmosphere from the back side of the right diaphragm. After completion of a stroke, an air distribution valve directs compressed air from the supply to the back side of the right diaphragm and exhausts air to the atmosphere from the left chamber. This continuous reciprocating motion, along with properly operating internal check valves, creates an alternating intake and discharge of pumped liquid into and out of each chamber that results in a nearly continuous pumping action from the combined chambers.

A diaphragm pump air motor contains an air distribution valve that shifts positions at the end of each stroke of the pump. The air distribution valve alternately directs supply air pressure to one chamber and exhausts the other. Air motors often use a two-stage valve to control the reciprocating motion of the pump (see Figure 11). The pilot valve supplies a pilot air pressure signal to the air distribution valve throughout the entire stroke of the pump, even though pressure oscillations in the pumping system may occur. The pilot valve is not connected directly to the diaphragm's connecting rod, which provides a "deadband" to prevent the power valve from erroneously shifting just after the end of each stoke. The two views contained in Figure 11 depict the position of the moving parts just before the pilot valve is moved by its contact with the diaphragm washer. Depending on its position, the pilot rod alternately pressurizes and exhausts the large end of the air distribution valve. Other valve design configurations pressurize and exhaust both ends of the air distribution valve.

The two common types of liquid check valves that are used in a diaphragm pump are the *flap valve* and the *ball valve*. A flap valve pump (see Figures 9 and 10) can handle nearly marble-sized solids. Because the discharge is from the bottom of the diaphragm chambers, the pump is ideally suited for pumping solids in suspension that may tend to settle out, particularly when the pumping rate is reduced or when the pump is shut down. The bottom outlet enables foreign matter to be easily pumped out of the chambers.

The popular air-operated, double-diaphragm pump with ball valves (see Figures 12 and 13) features the inlet at the bottom of the diaphragm chambers, and the outlet is at the top. The top discharge arrangement has the advantage of enabling air or vapors to be easily expelled from the chambers. Trapped air or vapors in pumps having bottom outlets can reduce the volumetric displacement of the pumps as the air or vapor is alternately compressed and expanded, instead of the liquid being displaced. This can be a concern in low-flow applications requiring relatively high pumping pressure and that handle viscous liquids. In higher flow applications, a sufficient turbulence is present and air or vapors mix with the pumped liquid to purge the pumping chambers of the gases.

The performance chart of a typical 2-in (51-mm) air-operated, double-diaphragm pump (see Figure 14) is similar to that of other pump types but contains air consumption rather than horsepower consumption. With a constant supply pressure of compressed air, work and energy relationships in the air chambers, as well as liquid flow losses within the pump, result in a downward sloping head-capacity curve, similar to a centrifugal pump. The following are noteworthy features of air-operated diaphragm pumps:

- With the pump shut off, there is no power consumption. Air consumption is approximately proportional to the flow rate; there is zero air consumption at a zero flow rate and maximum air consumption at a maximum flow rate. This feature enables diaphragm pumps to be used in applications where the flow rate must be varied over a wide range or from no flow to a high flow rate.

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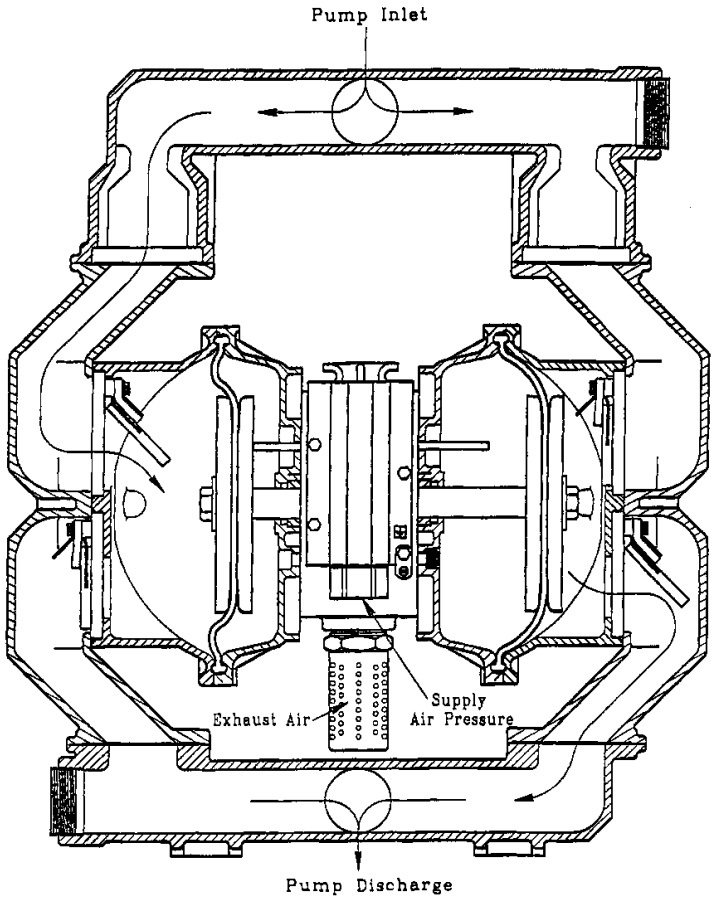


FIGURE 9 Cross-section of an air-operated double-diaphragm pump with flap check valves (Ingersoll-Rand Fluid Products)

- The pump discharge pressure remains the same for a given capacity and air inlet pressure regardless of the specific gravity of the liquid being pumped. The discharge head varies with the specific gravity of the pumped fluid, because pressure and not head is primarily maintained by the air pressure. For centrifugal pumps, the discharge pressure is directly proportional to the specific gravity of the liquid being pumped, while head is fixed at a given point on the performance curve. The supplied air pressure, pump flow rate, and NPSH set the discharge pressure for a given air operated pump, not the head, which is a characteristic of a centrifugal pump operated at a fixed speed.
- The NPSH required for an air-operated diaphragm pump is defined in the same manner as with any reciprocating pump. The NPSH required is determined to occur when the volumetric efficiency drops by a measurable amount (normally 3%) at a fixed speed or flow rate.

MATERIALS OF AIR-OPERATED DIAPHRAGM PUMPS

Check valves for diaphragm pumps are of three types: flap, ball, and poppet (see Figure 15). Flap valves preferably hang in a vertical position with a horizontal flow through the valves. Their principle advantage is the capability to handle large objects in suspension. Flap valves with elastomeric hinges are extensively used. Ball valves are arranged to provide a vertical flow through the valve seats. Poppet valves are usually guided by a valve stem and are spring-loaded. The poppet type of valve is not position-sensitive and can

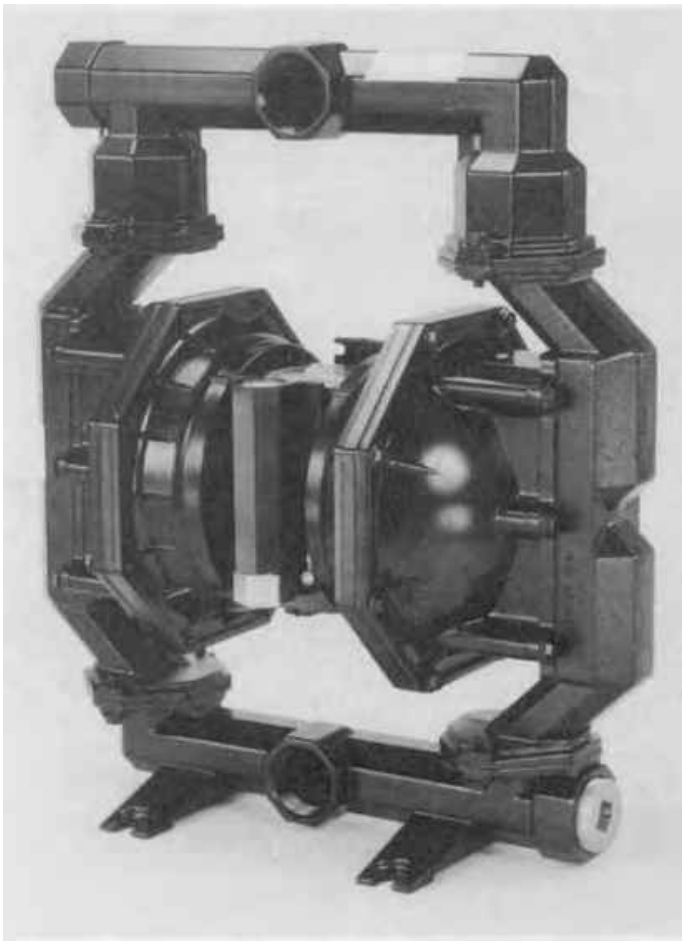


FIGURE 10 ARO air-operated double-diaphragm pump with flap check valves (Ingersoll-Rand Fluid Products)

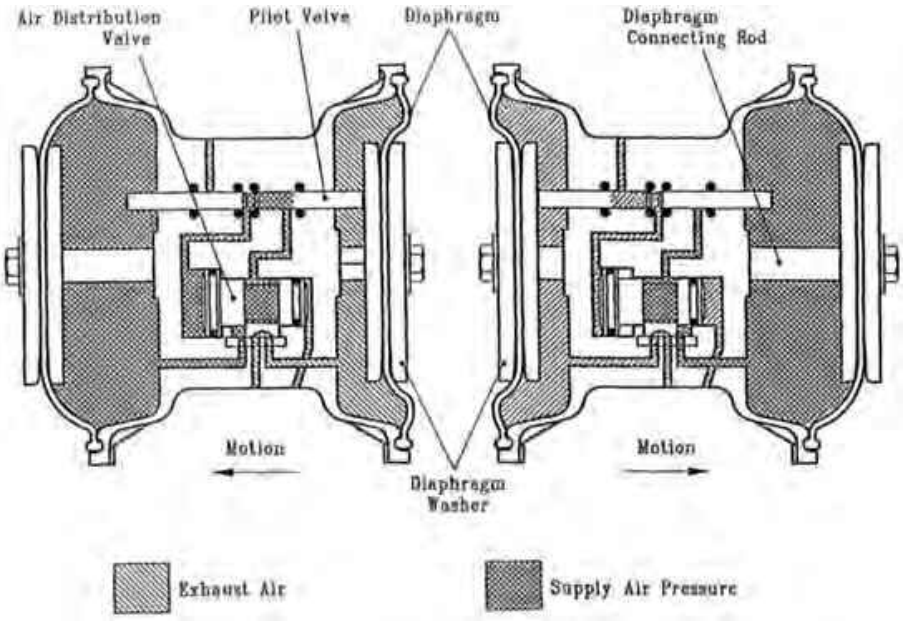


FIGURE 11 Cross-section of an air-operated double-diaphragm pump air motor operation (Ingersoll-Rand Fluid Products)

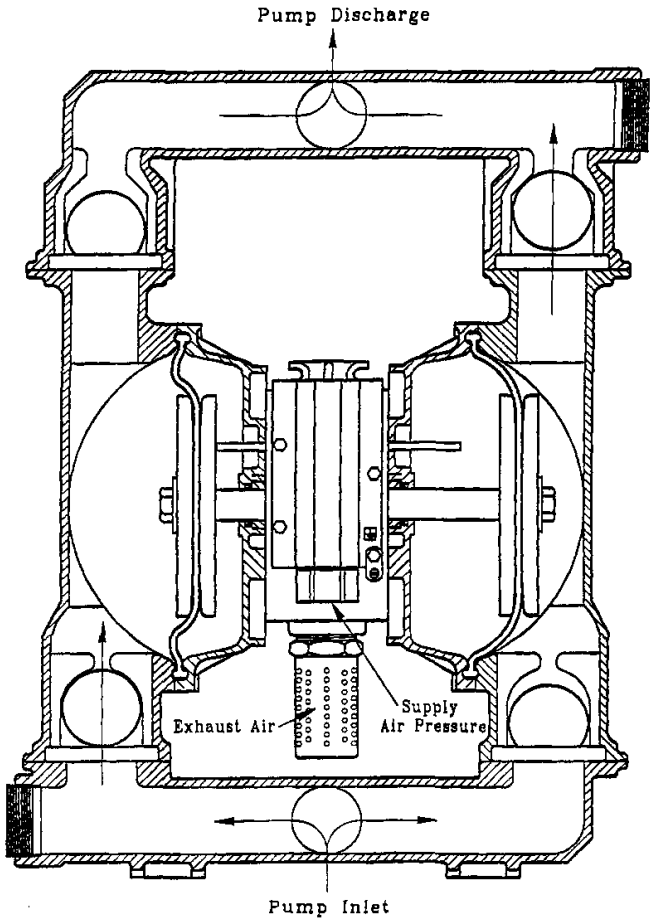


FIGURE 12 Cross-section of an air-operated double-diaphragm pump with ball check valves (Ingersoll-Rand Fluid Products)

operate in any orientation. Since many diaphragm pumps are used to pump abrasive slurries, the valves usually have elastomeric faces or are elastomeric balls. Large ball check valves may have metal cores covered by a thick wall of synthetic rubber. They may also be made of solid rubber, a thermoplastic elastomer, or Teflon®.

Diaphragms are customarily made of fabric-reinforced synthetic rubber, thermoplastic elastomers, or PTFE. Diaphragm materials include most of the synthetic rubbers: neoprene, Buna N, EPR, Viton®, thermoplastic elastomers such as Hytrel® and Santoprene® as well as Teflon®. Many combinations of pump and diaphragm materials can cover a wide range of pumped products. For some solvents and aggressive acids or alkalis, Teflon® diaphragms can be used either directly or as an overlay on the conventional diaphragm.

The diaphragms in air-operated, double-diaphragm pumps are essentially balanced and act simply as membranes separating the compressed air from the product being pumped. The only unbalance occurs during the suction stroke of one diaphragm while it is being pulled by the shaft-connected other diaphragm. When the suction lift is minimal, the unbalance may be negligible.

Pump case materials include cast iron, aluminum, stainless steel, Carpenter 20, Hastelloy C, and plastics reinforced with glass or metal fibers.

APPLICATIONS OF AIR-OPERATED DIAPHRAGM PUMPS

General Diaphragm pumps are ideally suited for handling abrasive slurries because 1) the liquid velocity through the check valves and pumping chambers does not exceed the

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FIGURE 13 ARO air-operated double-diaphragm pump with ball check values (Ingersoll-Rand Fluid Products)

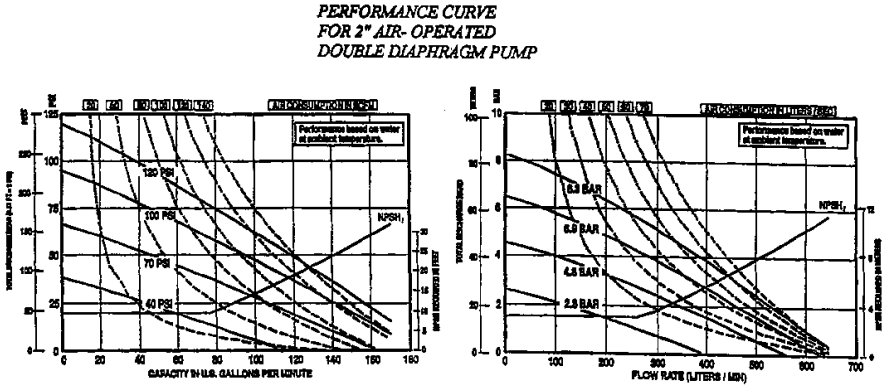
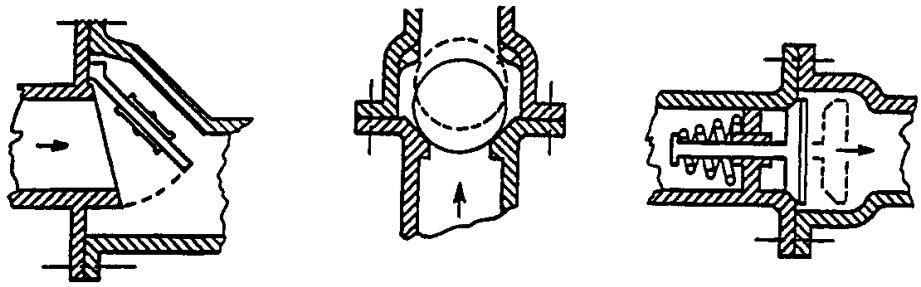


FIGURE 14 Performance curve for a 2-in air-operated double-diaphragm pump



FLAP BALL POPPET

FIGURE 15 Three basic types of check valves

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pipeline velocity and 2) the scouring and abrasions from the slurry are minimal. Because no close-fit sliding or rubbing parts exist and velocities are low, these pumps can be used for liquids with viscosities up to 50,000 SSU (11,000 cSt). Because the turbulence and mixing are minimal, they are ideally suited for shear-sensitive materials, such as latex.

Pumping Dry Powders Since diaphragm pumps can pump air as well as liquids, they are used successfully to pump dry powders. The air acts as a fluid medium for the powders in suspension, and the pump moves the air containing the suspended powder. Sometimes it is necessary to inject air into the powder to lower the apparent specific gravity and to get the powder into suspension.

Pump Controls An air pressure regulator in the compressed air supply line can control the pumping pressure. An air line valve can control the pumping rate. Thus, the pressure and capacity are easily controlled. Although air-operated diaphragm pumps are displacement pumps, they are not positive ones. The maximum pumping pressure cannot exceed the pressure of the compressed air powering the pumps. Table 1 describes the control of diaphragm pumps with the features inherent with each drive type.

Liquids Handled and Applications A variety of liquids can be handled by AODPs, yet the compatibility of pump case materials, valve materials, and diaphragms is the basic application limitation. Liquids and slurries handled by diaphragm pumps include ceramic slurry, paint, cement grout, chemicals, glue, resins, petroleum products, driller's mud, mill scale, ore concentrates, printer's ink, sewage, filter aids, latex, waste oils, wood preservatives, core washes, asphaltic coatings, bilge waste, radioactive waste, lapping compounds, porcelain frit, mine tailings, volatile solvents, coolants with metal fines, varnish, acids, coatings, soapstone slurries, explosives, lime slurries, yeast, chocolate, and wine. Typical applications of AODPs include tank and container loading and unloading, fluid filtration, spray painting, adhesive application, process mixing and batching, cutting oil, machine coolant, lubrication, sump pumping, dewatering, and waste water treatment.

LIMITATIONS AND ADVANTAGES OF AIR-OPERATED DIAPHRAGM PUMPS

This section outlines the pros and cons of AODPS. Table 1 also details the controls of the different types of pumps. The limitations of AODPs are as follows:

TABLE 1 Control of diaphragm pumps

Feature	Electro-Magnetically Driven	Mechanically Driven	Hydraulically Driven	Air-Operated
Can run dry:	Yes	Yes	Yes	Yes
Self-priming:	Yes	Yes	Yes	Yes
Discharge can be shutoff without damage:	Yes	No	Yes	Yes
Relief valve required:	No	Yes	Integral to pump	No
Flow controlled by:	Motor speed or stroke-length adjustment	Motor speed or stroke-length adjustment	Motor speed or stroke-length adjustment	Motor speed or discharge valve

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- AODPs are not practical for pumping rates above about 300 gpm (1150 ℓ/m).
- AODPs are not manufactured for operating air pressures above 125 lb/in^2 gauge (8.6 bar). However, some versions are available that increase the pressure ratio by 2:1 or 3:1.
- Ice formation in air motors can occur (basic to the physics of operation), but the effect can be minimized by proper application and design.
- Diaphragms have a finite life. Fluids with abrasives and higher process temperatures can limit a diaphragm's life, but many material choices are available, including Teflon®, and several thermoplastic elastomers.

An AODP's advantages are as follows:

- They are self-priming from a dry start.
- The pumps have an infinitely variable flow rate and pressure-within-pressure and capacity ranges.
- AODPs have no dynamic seals or packing.
- They can run dry indefinitely.
- The discharge can be throttled to zero flow indefinitely.
- No air is used when the AODP is deadheaded. Electrically driven pumps will consume a significant portion of the rated power when no demand flow exists.
- AODPs are suited for use in hazardous environments (no electric power required).
- Power is used in proportion to the pumping rate.
- They can be used in confined areas without heat buildup.
- AODPs can pump abrasive slurries and solids in suspension.
- The pumps can handle viscous liquids up to 50,000 SSU (11,000 cSt).
- A minimal degradation of the viscosity of shear-sensitive materials occurs.
- They can pump dry powders in air suspension.
- No close-fit, sliding, or rotating parts come in contact with liquid.
- No bypass is required as in other displacement pumps.
- When properly maintained, there is zero leakage.
- They are simple to maintain and to repair.
- There is no bedplate and no coupling to align.
- AODPs can be used in handling aggressive chemical solutions.
- They can handle a wider range of materials than any other type of pump.

SECTION 3.7
SCREW PUMPS

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Screw pumps are a special type of rotary positive displacement pump in which the flow through the pumping elements is truly axial. The liquid is carried between screw threads on one or more rotors and is displaced axially as the screws rotate and mesh (see Figure 1). In all other rotary pumps, the liquid is forced to travel circumferentially, thus giving the screw pump with its unique axial flow pattern and low internal velocities a number of advantages in many applications where liquid agitation or churning is objectionable.

The applications of screw pumps cover a diversified range of markets including navy, marine, and utilities fuel oil services; marine cargo; industrial oil burners; lubricating oil services; chemical processes; petroleum and crude oil industries; power hydraulics for navy and machine tools; and many others. The screw pump can handle liquids in a range of viscosities, from molasses to gasoline, as well as synthetic liquids in a pressure range from 50 to 5000 lb/in^2 (3.5 to 350 bar) and flows up to 8000 gal/min (1820 m^3/h).

Because of the relatively low inertia of their rotating parts, screw pumps are capable of operating at higher speeds than other rotary or reciprocating pumps of comparable displacement. Some turbine-attached lubricating oil pumps operate at 10,000 rpm and even higher. Screw pumps, like other rotary positive displacement pumps, are self-priming and have a delivery flow characteristic, which is essentially independent of pressure, provided there is sufficient viscosity in the liquid being pumped.

Screw pumps are generally classified into single- or multiple-rotor types. The latter is further divided into timed and untimed categories.

The single-screw or progressive cavity pump (see Figure 2) has a rotor thread that is eccentric to the axis of rotation and meshes with internal threads of the stator (rotor housing or body). Alternatively, the stator is made to wobble along the pump centerline.

Multiple-screw pumps are available in a variety of configurations and designs. All employ one driven rotor in a mesh and one or more sealing rotors. Several manufacturers have two basic configurations available: single-end (in Figure 3) and double-end (in Figure 4) construction, of which the latter is the better known.

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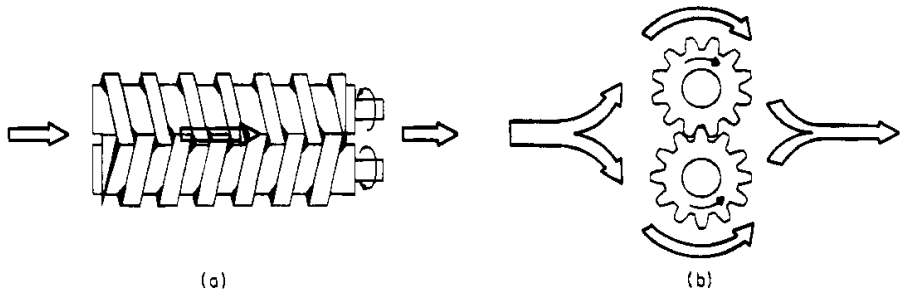


FIGURE 1 Diagrams of screw and gear elements, showing (a) axial and (b) circumferential flow

As with every pump type, certain advantages and disadvantages can be found in a screw pump design. These should be recognized when selecting the best pump for a particular application. The ***advantages*** of a screw pump design are as follows:

- A wide range of flows and pressures
- A wide range of liquids and viscosities
- High speed capability, allowing the freedom of driver selection
- Low internal velocities
- Self-priming, with good suction characteristics
- A high tolerance for entrained air and other gases
- Low velocities for minimum churning or foaming
- Low mechanical vibration, pulsation-free flow, and quiet operation
- A rugged, compact design that is easy to install and maintain
- High tolerance to contamination in comparison with other rotary pumps

The ***disadvantages*** are as follows:

- A relatively high cost because of close tolerances and running clearances
- Performance characteristics sensitive to viscosity changes
- High pressure capability may require long pumping elements
- Four sealing points between rotor and casing for timed two-screw design

THEORY

In screw pumps, it is the intermeshing of the threads on the rotors and the close fit of the surrounding housing that creates one or more sets of moving seals in a series between the pump inlet and outlet. These sets of seals or locks, as they are sometimes referred to, act

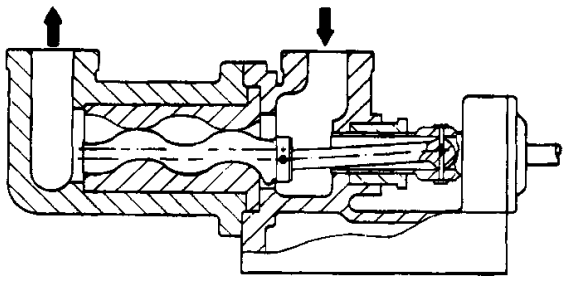


FIGURE 2 The single-screw or progressive cavity pump

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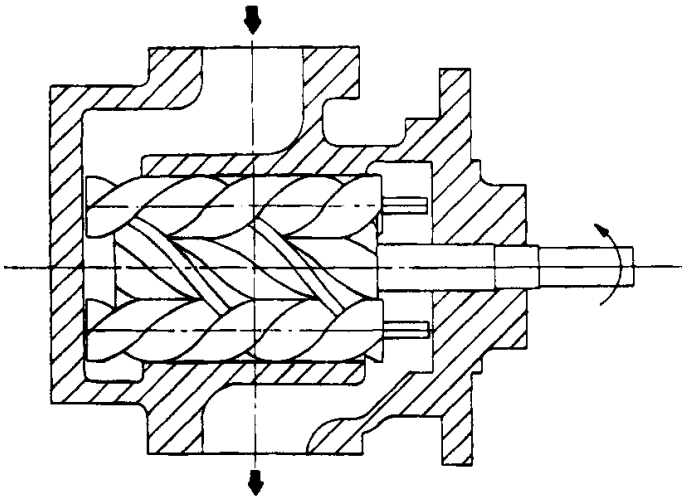


FIGURE 3 Multiple-screw single-end arrangement

as a labyrinth and provide the screw pump with its positive pressure capability. The successive sets of seals form fully enclosed cavities (see Figure 5) that move continuously from inlet to outlet. These cavities trap liquid at the inlet and carry it along to the outlet, providing a smooth flow.

Delivery Because the screw pump is a positive displacement device, it will deliver a definite quantity of liquid with every revolution of the rotors. This delivery can be defined in terms of displacement volume V_D, which is the theoretical volume displaced per revolution of the rotors and is dependent only upon the physical dimensions of the rotors. It is generally measured in cubic inches (cubic millimeters) per revolution. This delivery can also be defined in terms of theoretical capacity or flow rate Q_t measured in U.S. gallons per minute (cubic meters per hour), which is a function of displacement and speed N in rpm:

In USC units: $$Q_t = \frac{V_D N}{231}$$

In SI units: $$Q_t = 6 \times 10^{-8}\, V_D N$$

If no internal clearances existed, the pump's actual delivered or net flow rate Q would equal the theoretical flow rate. Clearances, however, do exist with the result that whenever a pressure differential occurs, there will always be internal leakage from outlet to inlet. This leakage, commonly called *slip* S, varies depending upon the pump type or model, the amount of clearance, the liquid viscosity at the pumping conditions, and the

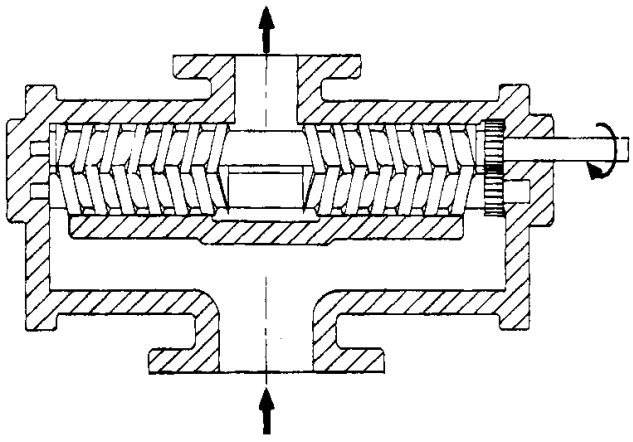


FIGURE 4 Mutliple-screw double-end arrangement

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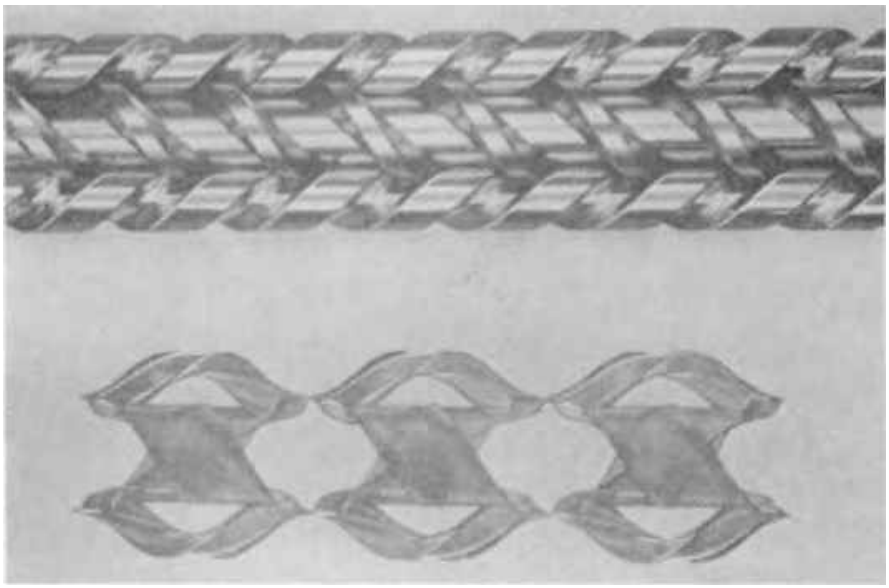


FIGURE 5 Axially moving seals and cavities. Alternate cavities filled with oil shown below (Imo Pump)

differential pressure. For any given set of these conditions, the slip for all practical purposes is unaffected by speed. The delivered flow rate or net flow rate therefore is the theoretical flow rate less the slip: $Q = Q_t - S$. If the differential pressure is almost zero, the slip may be neglected and $Q = Q_t$.

The theoretical flow rate of any pump can readily be calculated if all essential dimensions are known. For any particular thread configuration, assuming geometric similarity, the size of each cavity mentioned earlier is proportional to its length and cross-sectional area. The thread pitch measured in terms of the same nominal diameter, which is used in calculating the cross-sectional area (see Figure 6), defines the length. Therefore, the volume of each cavity is proportional to the cube of this nominal diameter, and the pump's theoretical flow rate is also proportional to the cube of this nominal diameter and the speed of rotation N (rpm):

$$Q_t = kD^3N$$

or, bringing the pitch into evidence,

$$Q_t = k_1 \times \text{pitch} \times D^2N.$$

From Figure 6, where pitch = KD, it follows that $k = k_1K$.

Thus, for a given geometry, it can be seen that a relatively small increase in pump size can provide a large increase in flow rate.

Slip can also be calculated,[1,2] but usually it depends upon empirical values developed by extensive testing. These test data are the basis of the design parameters used by every

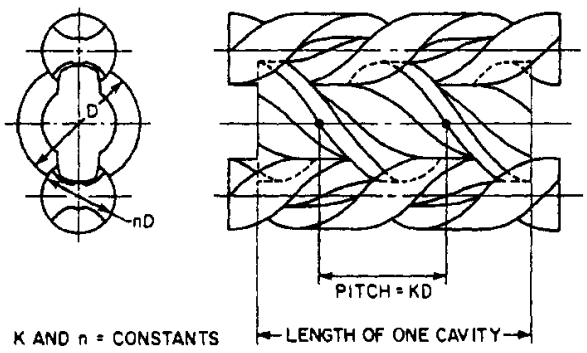


FIGURE 6 Screw-thread proportions, showing lead, diameters, and cavity length

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pump manufacturer. Slip generally varies approximately as the square of the nominal diameter. The net flow rate therefore is

$$Q_t = kD^3N - S$$

Pressure Capability As mentioned earlier, screw pumps can be applied over a wide range of pressures, up to 5000 lb/in^2 (345 bar), provided the proper design is selected. Internal leakage must be restricted for high-pressure applications. Close-running clearances and high accuracy of the conjugate rotor threads are requirements. In addition, an increased number of moving seals between the inlet and outlet are employed, as in classic labyrinth-seal theory. The additional moving seals are obtained by a significant increase in the length of the pumping elements for a given size of rotor and pitch. Here the minimum pump length is sacrificed in order to gain pressure capability.

The internal leakage in the pumping elements resulting from the differential pressure between the outlet and inlet causes a pressure gradient across the moving cavities. The gradient is approximately linear (see Figure 7) when measured at any instant. Actually, the pressure in each moving cavity builds up gradually and uniformly from inlet to outlet pressure as the cavities move toward the outlet. In effect, the pressure capability of a screw pump is limited by the allowable pressure rise across any one set of moving seals. This pressure rise is sometimes referred to as *pressure per closure* or *pressure per lock* and generally is of the order of 125 to 150 lb/in^2 (8.6 10 10.3 bar) with normal running clearances, but it can go as high as 500 lb/in^2 (35 bar) when minimum clearances are employed.

The above analysis is used in the design of pumps for handling liquid products where there are sufficient lubricating properties in the liquid to allow the screws to contact the bores of the casing. However, when handling products containing significant amounts of dissolved and/or entrained gas, it is preferable to have no rubbing contact between the rotating elements and the casing bores. For these pumps, the rotors are relatively short and stiff to ensure there is no rotor to bore contact with the lower-viscosity, gas-laden multiphase fluids.

Design Concepts The pressure gradient in the pump elements of all the types of screw pumps produces various hydraulic reaction forces. The mechanical and hydraulic techniques employed for absorbing these reaction forces are among the fundamental differences in the types of screw pumps produced by various manufacturers. Another fundamental difference lies in the method of engaging, or meshing, the rotors and maintaining the running clearances between them. Two basic design approaches are used:

- The *timed rotors* approach relies upon an external means for phasing the mesh of the threads and for supporting the forces acting on the rotors. In this concept, theoretically, the threads do not come into contact with each other or with the housing bores in which they rotate (refer to Figure 4).
- The *untimed rotors* approach relies on the precision and accuracy of the screw forms for the proper mesh and transmission of rotation. They utilize the housing bores as journal bearings supporting the pumping reactions along the entire length of the rotors (refer to Figure 3).

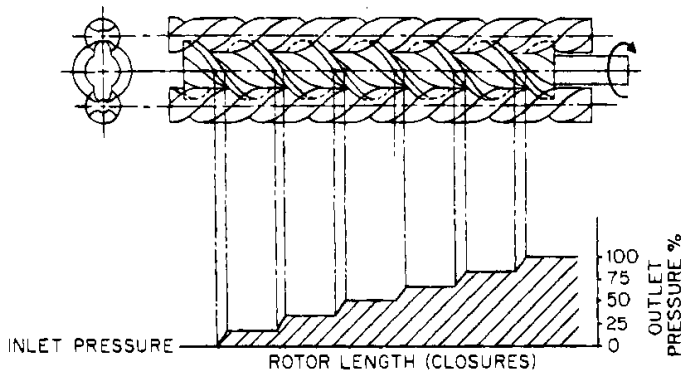


FIGURE 7 Pressure gradient along a screw set

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Timed screw pumps require separate timing gears between the rotors and separate support bearings at each end to absorb the reaction forces and maintain the proper clearances. Untimed screw pumps do not require gears or external bearings and thus are considerably simpler in design.

CONSTRUCTION

Basic Types As indicated in the introduction, three major types of screw pumps exist:

- Single-rotor
- Multiple-rotor timed
- Multiple-rotor untimed

The second and third types are available in two basic arrangements, single-end and double-end. The double-end construction (see Figure 8) is probably the best-known version, as it has been by far the most widely used for many years because of its relative simplicity and compactness of design.

Double-End Screw Pumps The double-end arrangement is basically two opposed, single-end pumps or pump elements of the same size with a common driving rotor that has an opposed, double-helix design with one casing. As can be seen from Figure 8, the fluid enters a common inlet with a split flow going to the outboard ends of the two pumping elements and is discharged from the middle or center of the pump elements. The two pump elements are, in effect, pumps connected in parallel. The design can also be provided with a reversed flow for low-pressure applications. In either of these arrangements, all axial loads on the rotors are balanced, as the pressure gradients in each end are equal and opposite.

The double-end screw pump construction is usually limited to low- and medium-pressure applications, with 400 lb/in^2 (28 bar) being a good practical limit to be used for planning purposes. However, with special design features, applications up to 1400 lb/in^2 (97 bar) can be handled. Double-end pumps are generally employed where large flows are required or where very viscous liquids are handled.

Single-End Screw Pumps All three types of screw pumps are offered in the single-end construction. As pressure requirements in many applications have been raised, the single-end design has come into much wider use because it provides the only practical means

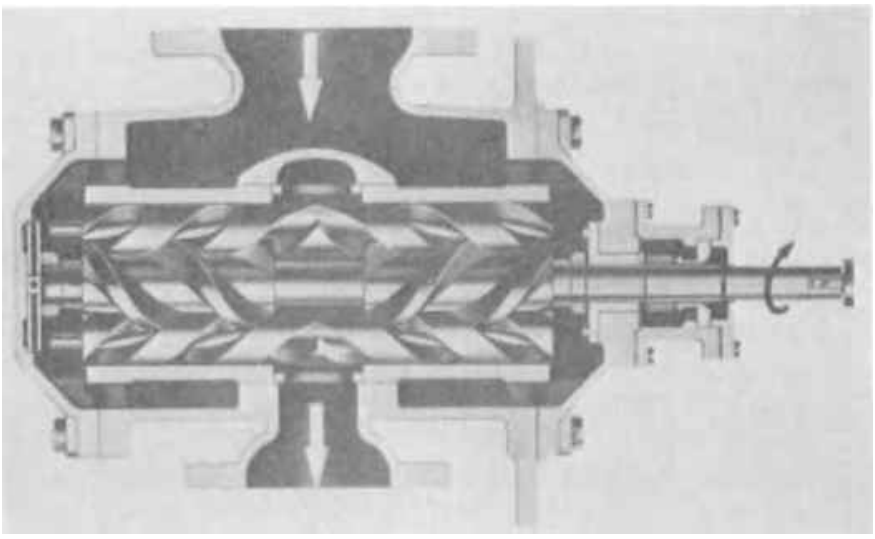


FIGURE 8 Double-end pump. Flow path provides axial balance (Imo Pump)

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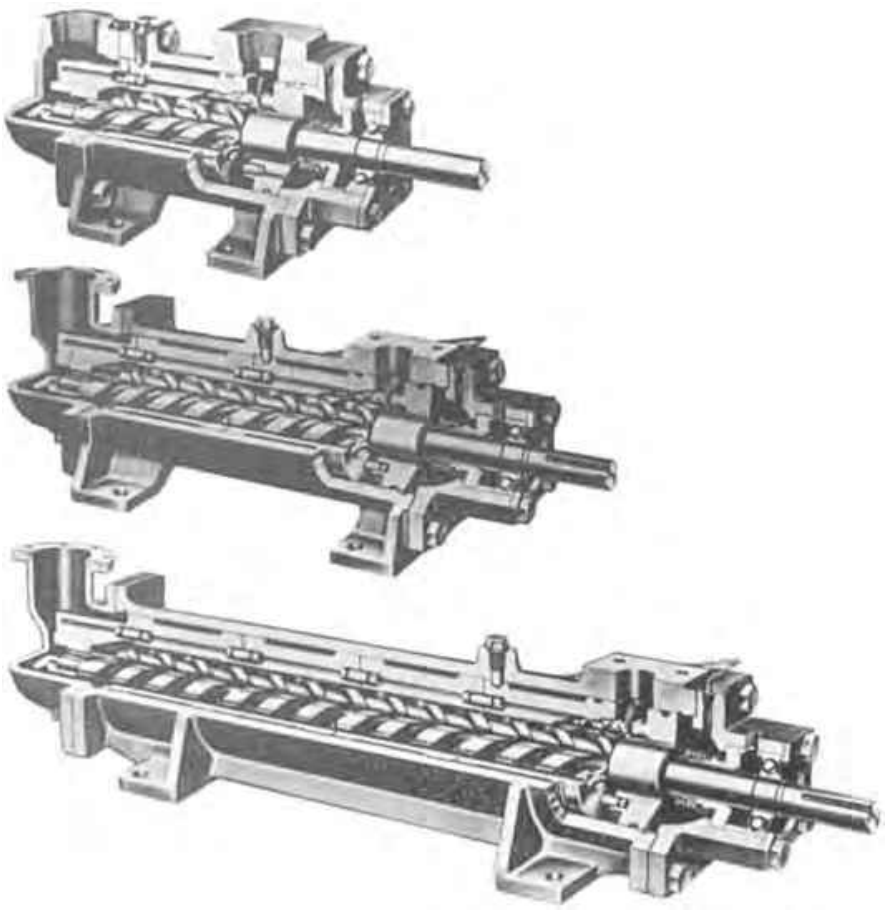


FIGURE 9 Increasing pump pressure capability by modular design (Imo Pump)

for obtaining the greatest number of moving seals necessary for high-pressure capabilities. The main penalties of the single-end pump are the requirement and complexity of balancing the axial loads.

The single-end construction is most often employed for handling low-viscosity fluids at medium-to-high pressures or hydraulic fluids at very high pressures. The single-end design for high pressures is developed by literally stacking a number of medium-pressure, single-end pumping elements in a series within one pump casing (see Figure 9). The single-end construction also offers the best design arrangement for quantity manufacturing.

Special mention must be made of the single-end, single-rotor design (see Figure 10). The pump elements of this design consist of only a stator and one rotor. The stator has a double-helix internal thread and is constructed of hard chrome stainless or tool steel material. One version of this design uses internally developed pressure in the pump to compress the elastomeric stator on the rotor, thus maintaining minimum running clearances.

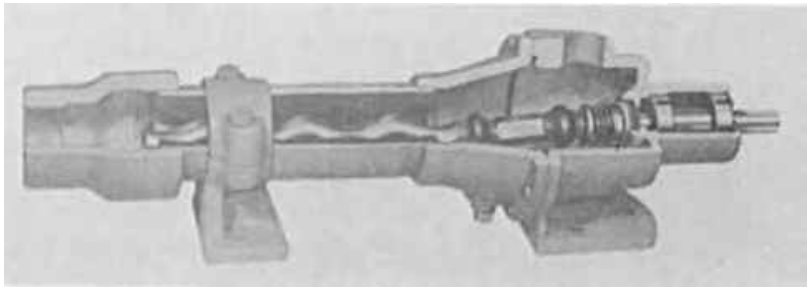


FIGURE 10 Single-rotor pump (Robbins-Meyers)

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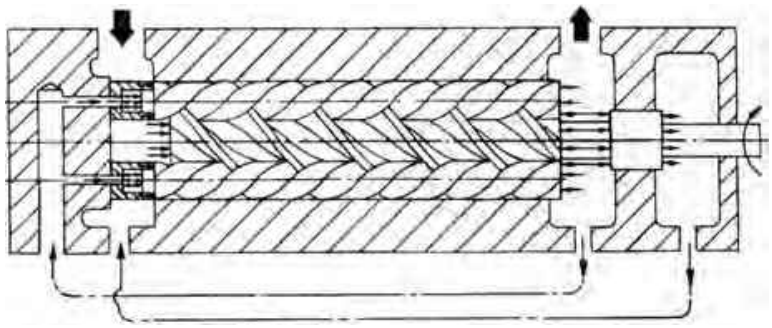


FIGURE 11 Axial balancing of power and idler rotors

In most single-end designs, special axial balancing arrangements must be used for each of the rotors, and, in this respect, the design is more complicated than the double-end construction. For smaller pumps, strictly mechanical thrust bearings can be used for differential pressures up to 150 lb/in^2 (10.3 bar), while hydraulic balance arrangements are used for higher pressures. For large pumps, hydraulic balance becomes essential at pressures above 50 lb/in^2 (3.5 bar).

Hydraulic balance is provided through the balance piston (see Figure 11) mounted on the rotors between the outlet and seal or bearing chambers, which are at inlet pressure. This piston is exposed to discharge pressure in the outlet chamber and is equal in area to the exposed area of the driven rotor threads; thus, the hydraulic forces on the rotor are canceled out.

Timed Design Timed screw pumps having timing gears and rotor support bearings are furnished in two general arrangements: internal and external. The internal version has both the gears and the bearings located in the pumping chamber and the design is relatively simple and compact (see Figure 12). This version is generally restricted to the handling of clean lubricating fluids, which serve as the only lubrication for the timing gears and bearings.

The external timing arrangement is the most popular and is extensively used. It has both the timing gears and the rotor support bearings located outside the pumping chamber (see Figure 13). This type can handle a complete range of fluids, both lubricating and nonlubricating, and, with proper materials, has good abrasion resistance. The timing gears and bearings are oil-bath-lubricated from an external source. This arrangement requires the use of four stuffing boxes or mechanical seals, as opposed to the internal type, which employs only one shaft seal.

The main advantage of the timed screw pump is that the timing gears transmit power to the rotors with no contact between the screw threads, thus promoting long pump life. The gears and rotors are timed at the factory to maintain the proper clearance between the screws. With certain designs and loading characteristics, the bearings at each end of

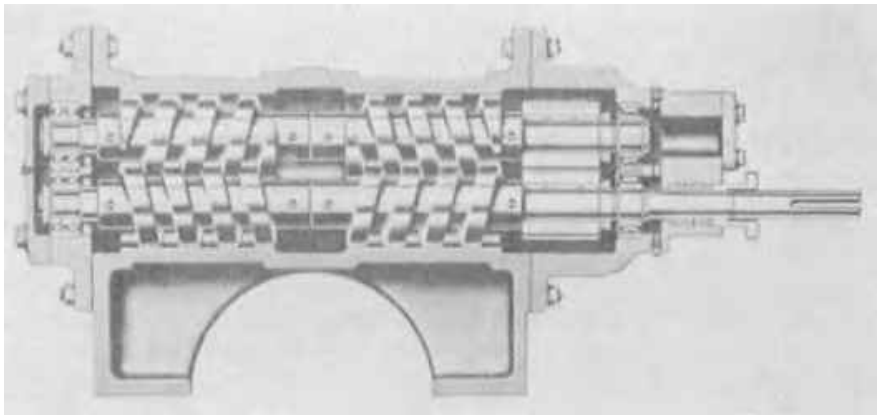


FIGURE 12 Double-end internal gear design (Flowserve Corporation)

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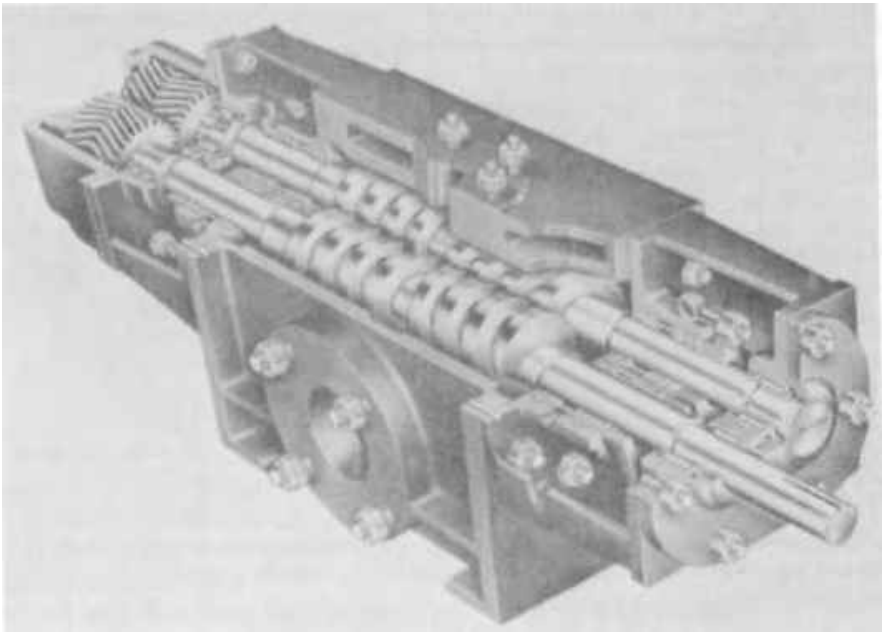


FIGURE 13 Double-end external gear design (Warren)

the rotating elements can support the rotors so that they do not come in contact with the housing; hence, no liner is required. One set of these bearings also positions and supports the timing gears. Under more heavily loaded conditions, the body bores act as sleeve bearings and provide additional support for the rotors.

The timing gears can be either spur or helical, herringbone, hardened-steel gears with tooth profiles designed for efficient, quiet, positive drive of the rotors. Antifriction radial bearings are usually of the heavy-duty roller type, while the thrust bearings, which position the rotors axially, are either double-row, ball-thrust or spherical-roller types.

The housing can be supplied in a variety of materials, including cast iron, ductile iron, cast steel, stainless steel, and bronze. In addition, the rotor bores of the housing can be lined with industrial hard chrome for abrasion resistance.

Since the rotors are not generally in metallic contact with the housing or with one another, they can also be supplied in a variety of materials, including cast iron, heat-treated alloy steel, stainless steel, Monel, and nitralloy. The outside of the rotors can also be furnished with a variety of hard coating materials such as nickel-based alloys, tungsten carbide, chrome oxide, or ceramic.

Untimed Design The untimed type of screw pump has rotors that have generated mating-thread forms that enable any necessary driving force to be transmitted smoothly and continuously between the rotors without the use of timing gears. The rotors can be compared directly with precision-made helical gears with a high helix angle. This design usually employs three rotor screws with a center or driven rotor that meshes with two close-fitting sealing or idler rotors symmetrically positioned about the central axis (see Figure 14). A close-fitting housing provides the only transverse bearing support for both driven and idler rotors.

The use of the rotor housing as the only means for supporting idler rotors is a unique feature of the untimed screw pump. No outboard support bearings are required on these rotors. The idler rotors in their related housing bores are, in effect, partial journal bearings that generate a hydrodynamic film. The key parameters of rotor size, clearance, surface finish, speed, fluid viscosity, and bearing pressure are related as in a journal bearing. Because, in this design, the screws transmit the driving torque to the idlers and because the casing bores support the rotors, the pumped fluid must have some lubricating properties. This prevents this pump design from being used in applications with any solids, abrasives or gas in the pumpage.

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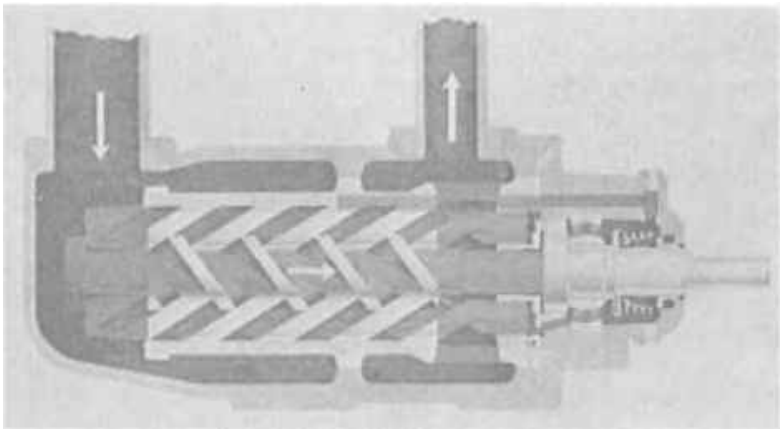


FIGURE 14 Single-end design (Roper)

Since the idler rotors are supported by the bores along their entire length, no significant bending loads are applied to them. The central driven rotor is also not subjected to any significant bending loads because of the symmetrical positioning of the idler rotors and the use of two threads on all the rotors (see Figure 15). This is quite different from the two-rotor design common to the timed type where the hydraulic forces generated in the pump create bending loads on the intermeshing pairs of screws.

In contrast to timed pumps, the untimed design, with its absence of timing gears and bearings, appears very simple, but its success depends entirely upon the accuracy and finish of the rotor threads and rotor housing bores. Special techniques and machine tools have been developed to manufacture these parts. The combination of design simplicity and manufacturing techniques has enabled this design to be used for very long rotor lengths, with a multiplicity of sealing closures, for applications up to 5,000 lb/in^2 (350 bar).

In special applications handling highly aerated oils, a four-rotor design is sometimes employed in the untimed type. Three idler rotors are equally spaced radially at 120° around the driven rotor. This design is not truly a positive displacement pump, and it falls outside the scope of this section.

The rotors in untimed pumps are generally made of gray or ductile iron or carbon steel. The thread surfaces are often hardened for high pressure and abrasive resistance. Flame hardening, induction hardening, and nitriding are currently used. Through-hardened tool steel or stainless steel can be used in some critical applications.

Rotor housings, or liners, are made of gray pearlitic iron, bronze, or aluminum alloy. In many instances, the bores as well as the rotors can be treated by the application of dry lubricant or toughening coatings. The pump casings are made of gray iron or ductile iron or cast steel where shock or other safety requirements demand it.

In many untimed designs, an antifriction bearing is employed on the shaft end of the driven rotor (refer to Figure 9) to provide precise shaft positioning for mechanical seal and

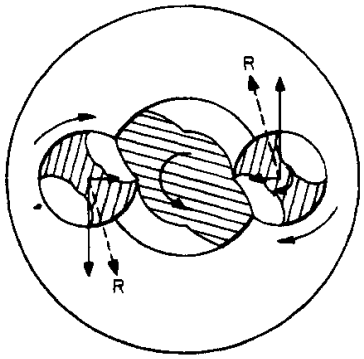


FIGURE 15 Force diagram on rotor set

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coupling alignment. This bearing can be either an external grease-sealed bearing or an internal type with the pumped fluid providing the lubrication. The bearing also supports overhung loads with belt or gear drives.

Seals As with any rotary pump, the sealing arrangement for the shafts is important and is often critical. Every type of rotary seal has been used in screw pumps at one time or another. Except for canned or sealless arrangements, all types of pumps require at least one rotary seal on the drive shaft. The timed screw pumps with external timing and bearings require additional seals at each rotor end to separate the pumped fluid from the lubricating oil necessary for the gears and bearings.

For drive shafts, rotary mechanical seals as well as stuffing boxes or packings are used, depending on the manufacturer and/or customer preference. Double back-to-back arrangements or tandem mounted seals with a flushing liquid are sometimes used for very viscous or corrosive substances.

Many different sealing face materials such as carbon, bronze, cast iron, Ni-resist, carbides, or ceramics are used, along with different secondary elastomer seals of various materials chosen to suit the application. Balanced and nonbalanced seals can also be used to handle significant pressures. In some pump designs, the seals are subjected to only the suction pressure, while in other designs the seals must seal against the full discharge pressure. In spite of these advantages, the stuffing box is still preferred by some users. The stuffing box requires regular maintenance and tightening, which many users find objectionable, but a mechanical seal failure usually results in a major shutdown.

PERFORMANCE

Performance considerations of screw pumps are closely related to applications, and so any discussion must cover both. In the application of screw pumps, certain basic factors must be considered to ensure a successful installation. These are fundamentally the same regardless of the liquids to be handled or the pumping conditions.

In most cases, the pump selection for a specific application is not difficult if all the operating parameters are known. It is often quite difficult, however, to obtain this information, particularly inlet conditions and fluid viscosity. It is a common feeling that, inasmuch as the screw pump is a positive displacement device, these items are unimportant.

In any screw pump application, regardless of design, suction lift, viscosity, and speed are inter related. Generally, low operating speeds are required for high-viscosity fluids while higher speeds are possible with lower-viscosity fluids.

Inlet Conditions The key to obtaining good performance from a screw pump, as with all other positive displacement pumps, lies in a complete understanding and control of inlet conditions and the closely related parameters of speed and viscosity. To ensure quiet, efficient operation, it is necessary to completely fill the moving cavities between the rotor threads with liquid as they open to the inlet, and this becomes more difficult as the viscosity, speed, or suction lift increases. It can be said that, if the liquid can be properly introduced into the rotor elements, the pump will perform satisfactorily.

It must be remembered that a pump does not pull or lift liquid into itself. Some external force must be present to push the liquid into the rotor threads initially. Normally, atmospheric pressure is the only force present, but in some applications a positive inlet pressure is available.

As with centrifugal pumps, the concept of NPSH required by the pump is used to ensure that there is sufficient pressure available at the inlet to fill the rotor cavities and suppress cavitation at the inlet. With rotary pumps this is sometimes referred to as *Net Inlet Pressure Required.*

Since the purpose of the NPSH available is to fill the moving inlet cavities with liquid, it is apparent that the faster the pump is operated *the greater is the amount of* NPSH that is required.

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Fluids and Vapor Pressure In many cases, screw pumps handle a mixture of liquids and gases, and therefore the general term *fluid* is more descriptive. Most of these fluids, especially petroleum products because of their complex nature, contain certain amounts of entrained and dissolved air or some other gas, which is released as a vapor when the fluid is subject to reduced pressures. If the pressure drop required to overcome entrance losses is sufficient to reduce the static pressure significantly, vapors are released in the rotor cavities and cavitation results.

In all screw pump applications, the absolute static pressure must never be allowed to drop below the vapor pressure of the fluid. This will prevent vaporization or cavitation. Cavitation, as mentioned previously, results when fluid vaporizes in the pump inlet because of incomplete filling of the pump elements and a reduction of pressure. Under these conditions, vapor bubbles, or voids, pass through the pump and collapse as each moving cavity moves into a domain of higher pressure. The result is noisy vibrations, the severity depending on the extent of vaporization or the incomplete filling and the magnitude of the discharge pressure. Since the cavitation bubbles occupy space normally taken by liquid, the output flow rate is reduced by this amount. It is therefore important to be fully aware of the characteristics of entrained and dissolved air as well as of the vapor pressure of the fluid to be handled. This is particularly true when a suction lift exists.

Net Positive Suction Head The suction conditions of screw pumps are normally defined by the *Net Positive Suction Head* (NPSH) available at the pump inlet. In some cases with systems open to the atmosphere, the inlet conditions can also be defined as suction lift. Suction lift occurs when the total available pressure at the pump inlet is below the atmospheric pressure and is normally the result of a change in elevation and pipe friction. Since atmospheric pressure at sea level corresponds to 14.7 lb/in^2 (1 bar) absolute, or 30 in Hg (762 mm Hg), in a system with the suction open to the atmosphere, this is the maximum amount of pressure available for moving the fluid, and suction lift cannot exceed these figures. In practice, the available pressure is lower because some of it is used up in overcoming friction in the inlet lines, valves, and fittings. It is considered the best practice to keep the suction lift as low as possible to ensure that an adequate NPSH is available to fill the suction cavities of the pumping screw. In all cases, the system NPSHA (available) should be greater than the pump NPSHR (required).

The NPSHA for a screw pump is calculated in the same way as for a centrifugal pump and the same basic application guidelines apply. The pump manufacturers test their pumps and measure the NPSHR (required) under various operating conditions. The end user must understand the suction pressure required and ensure that there is sufficient NPSHA (available) to suppress the cavitation. Generally the NPSHA should be at least 3 ft (1 m) greater than the NPSHR to provide a reliable application.

Many screw pumps operate with suction lifts of approximately 5 to 15 in Hg (127 to 381 mm Hg). Lifts corresponding to 24 to 25 in Hg (610 to 635 mm Hg) are not uncommon, and installations can operate satisfactorily when the absolute suction pressure is much lower. In the latter cases, however, the pumps usually take the fluid from tanks under a vacuum, and no entrained or dissolved air or gases are present. Great care must be taken when selecting pumps for these applications since the inlet losses can easily exceed the net suction head available for moving the fluid into the pumping elements.

The defining of suction requirements by the user and the stating of pump suction capabilities by the manufacturer have always been complex problems. In many cases, the NPSHA is difficult to predict due to changes in the fluid characteristics and the operating conditions. In addition, the NPSH required by the pump is a function of many variables, such as pump design, fluid characteristics, and operating conditions. If the operating conditions can be accurately defined, the pump manufacturers can predict the NPSHR and in many cases can provide pump modifications that can minimize the NPSHR.

To enable the pump manufacturer to offer the most economical selection and also assure a quiet installation, accurate suction conditions should be clearly stated. Specifying a lower NPSHA than actually exists may result in selection of a pump that operates at a lower speed than necessary. This means not only a larger and more expensive pump, but also a costlier driver. If the NPSHA is lower that stated, the outcome could be a noisy pump installation.

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Many known instances of successful installations exist where screw pumps were properly selected for low NPSHA conditions. Unfortunately, many other installations with equally low NPSHAs exist, which are not so satisfactory. This is because proper consideration was not given at the time the pump was specified and selected to the actual suction conditions at the pump inlet. Frequently, suction conditions are given as "flooded" simply because the source feeding the pump is above the inlet. In many cases, no consideration is given to outlet losses from the tank or to pipe friction in the inlet lines, and these can be exceptionally high in the case of viscous liquids.

When it is desired to pump extremely viscous products, care should be taken to use the largest feasible size of suction piping to eliminate all unnecessary fittings and valves, and to place the pump as close as possible to the source of the supply. In addition, it may be necessary to supply the liquid to the pump under some pressure, which can be supplied by elevation, air pressure, or mechanical means. These actions will provide the maximum NPSH possible to the pump inlet.

Entrained and Dissolved Air Although screw pumps can handle significant quantities of gas as discussed in the later multiphase section, entrained or dissolved gas in a pump system designed only for liquids can create a problem and must be considered in the application. This is particularly true of installations where recirculation occurs and the fluid is exposed to air through either mechanical agitation, leaks, or improperly located drain lines.

Most liquids will dissolve air or other gases retaining them in the solution, the amount being dependent upon the liquid itself and the pressure to which it is subjected. It is known, for instance, that lubricating oils at atmospheric temperatures and pressures will dissolve up to 10% air by volume and that gasoline will dissolve up to 20%. When pressures below the atmosphere exist at the pump inlet, dissolved air will come out of the solution. Both this and the entrained air will expand in proportion to the existing partial pressure of the air (= absolute pressure minus the vapor pressure of the liquid). This expanded air will accordingly take up a proportionate part of the available volume of the moving cavities, with a consequent reduction in delivered flow rate.

One of the apparent effects of handling liquids containing entrained or dissolved gas is noisy pump operation. When such a condition occurs, it is usually dismissed as cavitation. This should not be the case, particularly with screw pumps. With properly designed systems and pumps, quiet, vibration-free operation can be produced and should be expected. Noisy operation is inefficient; steps should be taken to make corrections until the objectionable conditions are overcome. Correct system inlet designs and optimized pump designs with a proper speed selection can go a long way toward overcoming the problem.

In some applications, the amount of gas can be significant and can make up the majority of the fluid volume. See the later section on handling special multiphase applications.

Viscosity It is not often that a screw pump is called upon to handle liquids at a constant viscosity. Normally, because of temperature variations, a wide range of viscosities will be encountered. For example, a pump may be required to handle a viscosity range from 150 to 20,000 SSU, the higher viscosity usually resulting from cold-starting conditions. This is a perfectly satisfactory range for a screw pump, but a better and a more economical selection may be possible if additional information can be obtained. This information includes such things as the amount of time the pump is required to operate at the higher viscosity, whether the motor can be overloaded temporarily, whether a multi-speed motor can be used, and if the discharge pressure will be reduced during the period of high viscosity.

Quite often, only the type of liquid is specified, not its viscosity, and assumptions must be made for the operating range. For instance, Bunker C or No. 6 fuel oil is known to have a wide range of viscosity values and usually must be handled over a considerable temperature range. The normal procedure in a case of this type is to assume an operating viscosity range of 20 to 700 SSF. The maximum viscosity, however, might easily exceed the higher value if extra-heavy oil is used or if exceptionally low temperatures are encountered. If either should occur, the result may be improper filling of the pumping elements, noisy operation, vibration, and overloading of the motor.

Although it is the maximum viscosity and the expected NPSHA that are used to determine the size of the pump and to set the speed, it is the minimum viscosity that affects the

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capacity. Screw pumps must always be selected to give the specified capacity when handling the expected minimum viscosity since this is the point at which the maximum slip, and hence minimum flow rate, occurs (see Figure 16). It should also be noted that the minimum viscosity often determines the selection of the pump model because most manufacturers have special lower-pressure ratings for handling liquids having a viscosity of less than 100 SSU.

Non-Newtonian Liquids The viscosity of most liquids is unaffected by any agitation or shear to which they may be subjected as long as the temperature remains constant. These liquids are accordingly known as *true* or *Newtonian* liquids, for which the viscosity is constant. Another class of liquids, however, such as cellulose compounds, glues, greases, paints, starches, slurries, and candy compounds, displays changes in viscosity as agitation is varied at constant temperature. The viscosity of these substances depends upon the shear rate at which it is measured, and these fluids are termed *non-Newtonian.*

If a substance is known to be non-Newtonian, the expected viscosity under the actual pumping conditions should be determined, because it can vary quite widely from the viscosity under static conditions. Since a non-Newtonian substance can have an unlimited number of viscosity values (as the shear rate is varied), the term *apparent viscosity* is used to describe its viscous properties. The apparent viscosity is expressed in absolute units and is a measure of the resistance to the flow at a given shear rate. It has meaning only if the shear rate used in the measurement is also given.

The grease-manufacturing industry is very familiar with the non-Newtonian properties of its products, as evidenced by the numerous curves that have been published of the apparent viscosity plotted against the rate of shear. The occasion is rare, however, when one can obtain accurate viscosity information when it is necessary to select a pump for handling these products.

It is practically impossible in most instances to give the viscosity of grease in the terms most familiar to the pump manufacturer, such as Saybolt Seconds Universal or Saybolt Seconds Furol, but only a rough approximation would be of great help. For applications of this type, data taken from similar installations are most helpful. Such information should consist of the type, size, flow rate, and speed of the installed pumps; the suction pressure; the temperature at the pump inlet flange; the total working suction head; and, above all, the pressure drop in a specified length of piping. From the latter, a satisfactory approximation of the effective viscosity under the operating conditions can be obtained.

If accurate shear rate-viscosity data are available, they can be used to more accurately predict pump performance. The shear rates in various areas of the pump can be calculated to determine the viscosity changes of the liquid as it passes through the pump. In this way, the effect on suction loss, slip, and friction loss can be analyzed to help predict the NPSH, flow rate, and power.

Speed It was previously stated that viscosity and speed are closely tied together and that it is impossible to consider one without the other. Although rotative speed is the

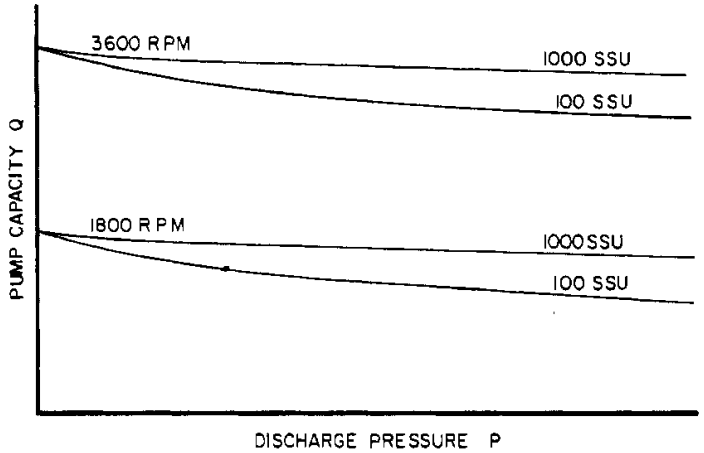


FIGURE 16 Flow rate (capacity)-vs-pressure rise

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ultimate outcome, the basic speed that the manufacturer must consider is the internal axial velocity of the liquid going through the rotors. This is a function of pump type, design, and size.

Rotative speed should be reduced when handling liquids of high viscosity. The reasons for this are not only the difficulty of filling the pumping elements, but also the mechanical losses that result from the shearing action of the rotors on the substance handled. The reduction of these losses is frequently of more importance than relatively high speeds, even though the latter might be possible because of positive inlet conditions.

Capacity The delivered flow rate (capacity) of any screw pump, as stated earlier, is the theoretical capacity less the internal leakage, or the slip, when handling vapor-free liquids. For a particular speed, $Q = Q_t - S$, where the standard unit of Q and S is the U.S. gallon per minute (cubic meter per minute).

The delivered capacity of any specific rotary pump is reduced by

- Decreasing speed
- Decreased viscosity
- Increased differential pressure

Because of the internal clearances between rotors and their housing, lower viscosities and higher pressures increase the slip, which results in a reduced flow rate for a given speed. The impact of these characteristics can vary widely for the various types of pumps. The slip, however, is not measurably affected by changes in speed and thus becomes a smaller percentage of the total flow at higher speeds. This is a significant factor in the handling of low-viscosity fluids at higher pressures, particularly in the case of untimed screw pumps that favor high speeds for the best results and best volumetric efficiency. This will not generally be the case with pumps having support-bearing speed limits.

Pump volumetric efficiency E_v is calculated as

$$E_v = \frac{Q}{Q_t} = \frac{Q_t - S}{Q_t}$$

with Q_t varying directly with speed. As stated previously, the theoretical capacity of a screw pump varies directly as the cube of the nominal diameter. Slip, however, varies approximately with the square of the nominal diameter. Therefore, for a constant speed and geometry, doubling the rotor size will result in an eightfold increase in theoretical flow rate and only a fourfold increase in slip. It follows therefore that the volumetric efficiency improves rapidly with increases in the rotor size, assuming the internal clearances can be maintained.

On the other hand, viscosity changes affect the slip inversely in a complex manner that depends on pump size, speed, and the level of the viscosity itself. Testing or advanced analysis is required to establish this relationship.[1] However, for a given viscosity in the range of 100 to 10,000 SSU an acceptable approximation is that slip varies approximately with the differential pressure. Thus, in this case, a change from 100 to 200 lb/in^2 (7 to 14 bar) would double the slip.

Figure 17 shows typical results for flow rate and volumetric efficiencies as functions of pump size.

Pressure Screw pumps do not in themselves create pressure; they simply transfer a quantity of fluid from the inlet to the outlet side. The pressure developed on the outlet side is solely the result of resistance to the flow in the discharge line. The slip characteristic of a particular pump type and model is one of the key factors that determine the acceptable operating range, and it is generally well defined by the pump manufacturer.

Power The brake horsepower (bhp), or, in SI units, the brake kilowatts, required to drive a screw pump is the sum of the theoretical liquid horsepower (kilowatts) and the internal power losses. The theoretical liquid power *twhp* (*tkW*) is the actual work done in moving the fluid from its inlet pressure condition to the outlet at the discharge pressure.

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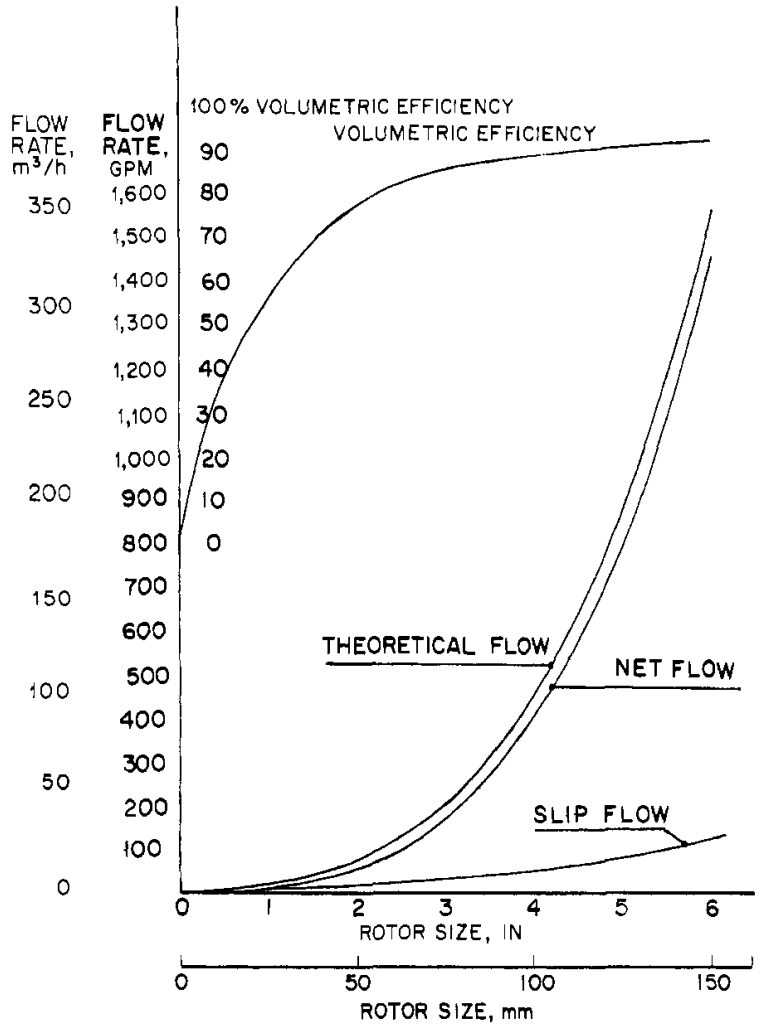


FIGURE 17 Typical variations of flow rate and volumetric efficiency as functions of pump size

Note that this work is done on all the fluid of the theoretical capacity, not just the delivered capacity, as the slip does not exist until a pressure differential D*P* occurs. Screw pump power ratings are expressed in terms of horsepower (550 ft-lbf/sec) in USCS units and in terms of kilowatts in SI units. The theoretical liquid horsepower (kilowatts) can be calculated as follows:

$$twhp = \frac{Q_t \Delta P}{1714} \left(tkW = \frac{Q_t \Delta P}{36} \right)$$

It should be noted that the theoretical liquid horsepower (kilowatts) is independent of the viscosity and is a function only of the physical dimensions of the pumping elements, the rotative speed, and the differential pressure.

The internal power losses are of two types: mechanical and viscous. The mechanical losses include all the power necessary to overcome the frictional drag of all the moving parts in the pump, including rotors, bearings, gears, and mechanical seals. The viscous losses include all the power lost from the fluid drag effects against all the parts in the pump as well as from the shearing action of the fluid itself. It is probable that the mechanical loss is dominant when operating at low viscosities and high speeds, and the viscous loss is the larger of these two losses at high-viscosity and slow-speed conditions.

In general, the losses for a given type and size of pump vary with the viscosity and the rotative speed and may or may not be affected by pressure, depending upon the type and

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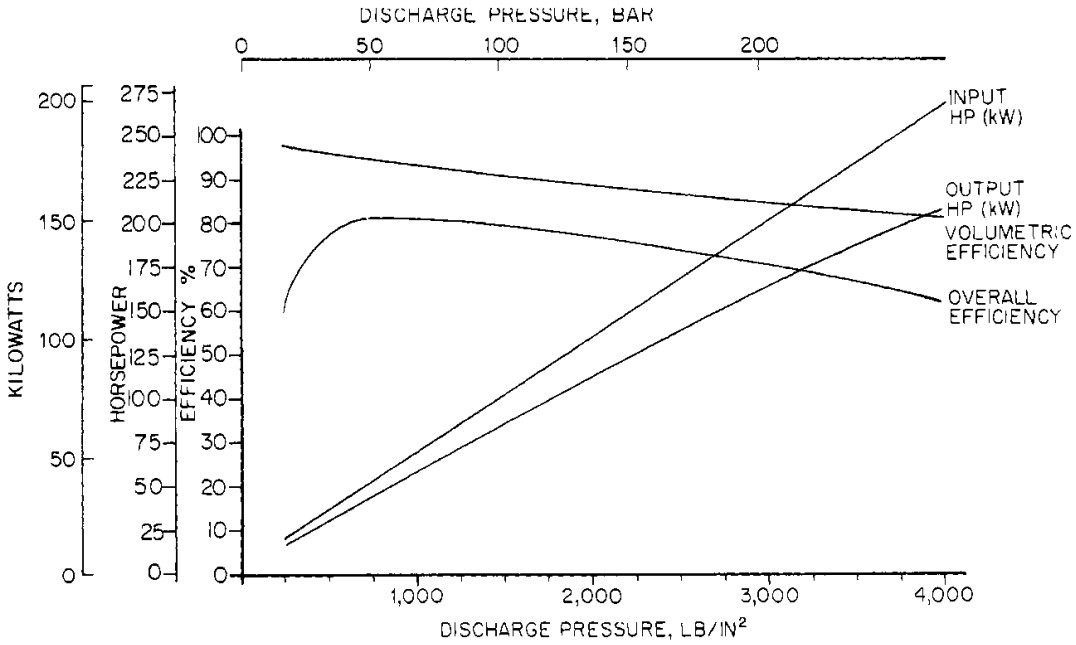


FIGURE 18 Typical overall efficiency curves

model of pump under consideration. These losses, however, must always be based upon the maximum viscosity to be handled since they will be highest at this point.

The actual pump power output, *whp* (*wkW*), or the delivered liquid horsepower (kilowatts), is the power imparted to the liquid by the pump at the outlet. It is computed similarly to theoretical liquid horsepower (kilowatts) using Q in place of Q_t. Hence, the value will always be less. The pump efficiency E_p is the ratio of the pump power output to the brake horsepower (see Figure 18).

In many cases the efficiency of the pump is not that significant, since there may be only a few pumps capable of handling high viscosity and difficult fluids. The main requirement is to provide a reliable equipment package that is capable of meeting the specified requirements.

SPECIAL MULTIPHASE APPLICATIONS

Screw pumps have been used with gas-entrained applications for many years, but recent process changes in oil field technologies have created requirements for pumping multiphase fluids, containing more than just nominal amounts of gases. In many oil well applications, the liquid oil flow eventually degenerates into all sorts of difficult multiphase mixtures of oil, gas, water, and sand. In the past, it was common for the gas to be separated and flared off at the well head with only the liquid product to be retained for further processing. If the gas is to be processed as well, separators, compressors, and dual pipelines are required to handle the gas phase. Therefore, a pump, which can handle these difficult liquids with high gas content, can save significant equipment costs as well as operating costs. Under various conditions, the well output can vary from 100% liquid to 100% gas and all possible combinations. The applications also require the pumping equipment to be able to switch rapidly between the extremes or to handle slugs of liquid or gas, while maintaining the full discharge pressure. Since these applications involve solid contaminants, internal bearing and non timed pump designs are not suitable. The single rotor design with elastomer stator has proven suitable for lower pressure applications while the timed two-screw type of pump has been proven capable of pumping these multiphase products at higher pressues.[2]

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Oil well applications can include traditional on-shore sites as well as off-shore platform installations. Subsea installations are also being used to reduce the high costs of equipment and operation of traditional oil platforms. In these applications, the pumping equipment is mounted on the sea bed with piping running to floating production, storage and off-loading facilities or to onshore gathering facilities. Although the actual pumping conditions are similar to surface installations, the installation and operating environments are far more challenging.

When pumping multiphase products with high *gas void fractions* (GVF), the pump must be designed with a small pitch to provide a sufficient number of locks. The key to pumping multiphase products is to ensure that some liquid is always available to seal the screw clearances and reduce the slip. Even a small amount of recirculated liquid is sufficient to provide this seal and enable the screw pump to operate with GVF-values approaching 100%. Depending on a number of factors, the volume of liquid required to seal and cool the screws can be two to five percent of the total inlet volume flow rate. In order to ensure that sufficient liquid is available at conditions of high GVF, a separate liquid flush can be provided or a trapping type of pump casing can be used. This type of casing has the suction and discharge openings located on the upper side of the casing to trap the liquid inside the screw chambers to seal the screw clearances. In some cases, some of the liquid portion is separated from the multiphase mixture being pumped. This separated liquid can then be recirculated back to the screws and mechanical seals to provide sealing and cooling liquid at times when the product is almost all gas. Figure 19 shows a special screw pump designed for multiphase applications with a casing designed to trap the liquid inside the casing bores to seal the screw clearances.[3]

When pumping liquids, the slip through the internal clearances increases with the differential pressure and decreases with increasing viscosity. However, in multiphase applications, as the GVF increases, the slip decreases until the inlet volume flow rate is equal to the pump displacement. This results in an almost constant inlet volume flow rate, regardless of the differential pressure. This performance can be explained by examining the pressure drop of the multiphase product across the finite clearances. This theoretical analysis confirms that a small amount of liquid in the clearances will effectively seal these clearances and reduce the slip to near zero. Figure 20 shows the typical performance characteristics for

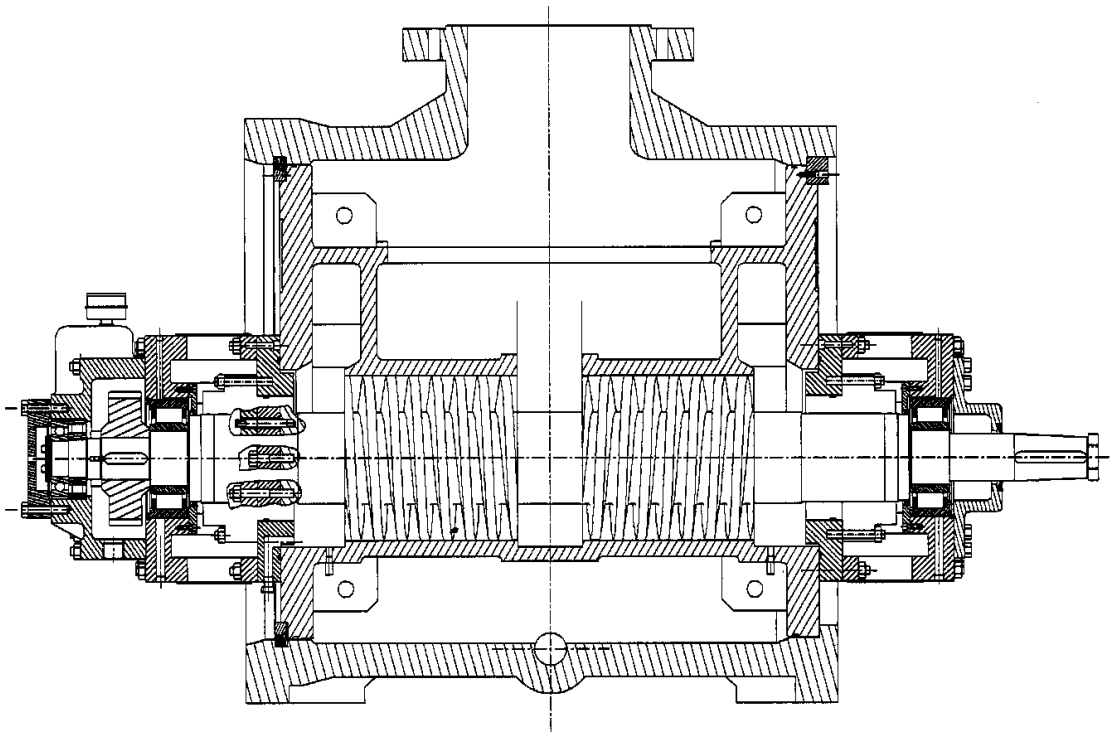


FIGURE 19 Special screw pump for multiphase applications (Flowserve Corporation)

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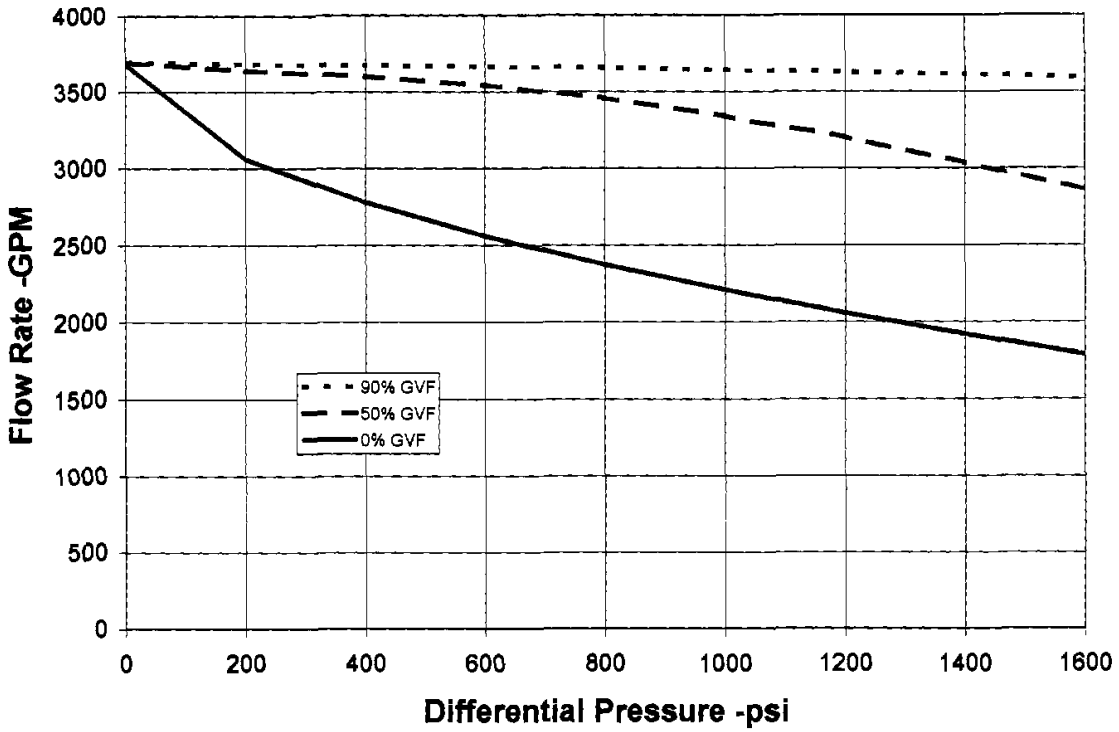


FIGURE 20 Typical performance of a screw pump on multiphase fluid (m^3/h = gpm/4.403; MPa = psi/145.04)

a screw pump in a multiphase application when pumping mixtures of air and water at various GVF values.[1,4]

It should be emphasized that screw pumps must be sized for the inlet volume conditions. Since the gas portion of the multiphase product is compressible, the inlet pressure and temperature conditions must be known in order to calculate the gas volume. The pump will ingest a fixed volume of product and the amount of liquid being pumped will depend on how much of this volume is being displaced by gas at the inlet to the screws.

When pumping higher GVF multiphase products some significant design changes must be made to the screw pumps to improve reliability and ensure proper pumping action. The normal practice used with viscous liquid pumping of using the casing bores to help support the rotor loads is not used with multiphase liquids due to the lower liquid viscosity, the high percentage of gas and the entrained sand and other contaminants present in oil field applications. As a result, pumps for multiphase applications are generally designed with stiffer rotors so that the rotors do not contact the casing bores. This design change is accomplished by using larger diameter shafts with shorter bearing spans. The integral shaft and rotor design used by some manufacturers significantly increases the rotor stiffness and minimizes the rotor deflection under load.

Another design feature used for multiphase applications involves limiting the pitch of the screws to ensure there are sufficient screw locks available to generate the pressure rise and produce a more gradual pressures rise across the screw length. Since the volume of the gas is compressed as the multiphase fluid moves through the screw passages, any large volume changes will result in pressure pulsations and resulting damaging vibration. A general guideline for higher pressure multiphase applications requires a minimum of three screw locks in order to minimize the pressure pulsations. In multiphase applications the maximum pressure rise occurs in the last lock when it opens to the full discharge pressure of the pump. Special porting of the screw profiles can also be used to help minimize the pressure pulsation and resulting vibration.

Again the mechanical seals are the most critical items in the pump required to ensure pump reliability. The pump should be designed to provide the best environment for the mechanical seals to operate properly. This includes limiting rotor deflection at the seal area and providing suitable flush or barrier fluid. Since these pumps can be designed to operate with the seals in either the suction or the discharge it is normally preferable to have

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the seals work against the lower suction pressure as opposed to the higher discharge pressure. When sealing against variable high suction or discharge pressures, the barrier fluid pressure must be controlled to ensure that it is always greater than the product pressure.

In many multiphase applications, the drive motors are equipped with variable speed controls to allow the pump to operate over a wide speed range to match specific start-up and operating well conditions. In these cases the speed is controlled to maintain a constant pressure at the pump suction. If the well flow rate increases, the suction pressure increases, causing the speed of the pump to also increased to maintain a constant suction pressure and thereby match the flow rate from the well. Multiphase pumps are often used as booster pumps to draw down the well head pressure and thereby increase the well flow rate and boost production. These pumps are often sized to handle high flow rates and can then be installed on a cluster of wells to boost the pressure into the pipeline.

Subsea A special area of application for multiphase pumps is subsea. With special modifications, the rotary screw multiphase pumps can be coupled to submersible motors and mounted on the sea bed, instead of on surface platforms. The idea of pumping multiphase products directly from a subsea well head to onshore or floating production and storage facilities by means of submersible multiphase pumps has significant potential savings in separating equipment and platforms.[5] These special subsea pumps often require much higher pressure capabilities than surface mounted pumps, to handle both the high ambient subsea pressures and also the high discharge pressures required. Again the mechanical seals are a critical design area with special considerations required to ensure a reliable design. The sealing environment can be significantly improved by maintaining a constant differential pressure across the seal faces by means of a pressure compensator. This pressure compensator senses the product pressure on the seals and then provides a slight pressure boost to the barrier fluid to maintain this constant differential pressure. The pump and system must be designed in conjunction with the submersible motor, pressure compensator and lube oil cooler. Obviously, the motor and valve control functions as well as the lube oil top-up capabilities must be integrated into the complete pump module to provide a reliable unit to operate in the subsea environment. Subsea multiphase pumps are being investigated for deeper water applications and higher boost pressures. Initial subsea pumping installations involved water depths up to 1500 m, with boost pressures of 1250 psi (86 bar), applications up to 3000 m water depth with pressure boosts to 3000 psi (207 bar) are now being pursued. Figure 21 shows a special two-screw multiphase

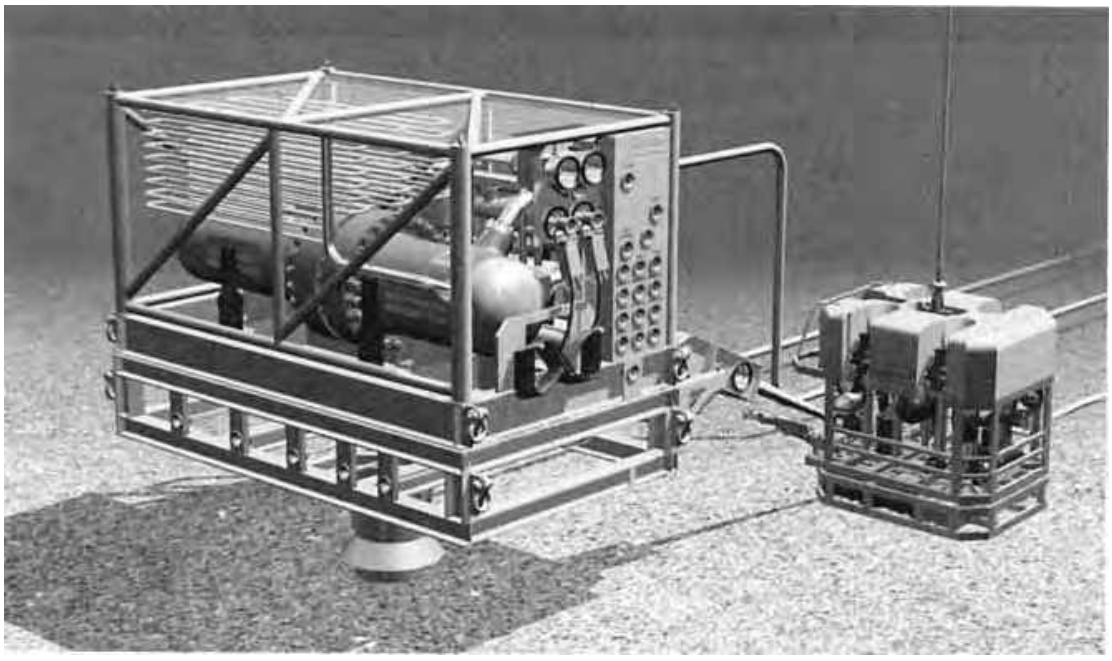


FIGURE 21 Multiphase screw pump module for subsea deployment (Flowserve Corporation)

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pump module, configured for subsea applications with high-pressure, shell-type casing design, integrated submersible motor, pressure-compensator and oil cooler.

Wet Gas Compression Another application for multiphase screw pumps is for "wet gas compression." This application involves pumping multiphase products that are essentially gas but can also have occasional high liquid content. These applications generally involve fluids with gas content in the 98% GVF range. There are two areas of concern with this type of application. As previously indicated, there must always be some liquid available in the pump to seal the clearances between the screws and the casing bores. Also with this high gas content, special considerations must be made for the thermodynamic heating characteristics of the compressed gas. Since the gas provides much lower heat absorption than the normal liquid, the product temperature can increase significantly as the pressure is increased. The pump must also be able to handle the full range of GVF values from slugs of liquid to 100% gas. For these applications special screw designs, liquid injection and recirculating liquid separators can be used to ensure there is sufficient liquid available to seal and cool the rotors.

INSTALLATION AND OPERATION

Rotary pump performance and life can be improved by following the recommendations on installation and operation outlined in this section.

Pipe Size Resistance to the flow usually consists of differences in elevation, fixed resistances or restrictions such as orifices, and pipe friction. Nothing can be done about the first, since this is the basic reason for using a pump. Something can be done, however, about restrictions and pipe friction. Significant amounts of money can be wasted because of piping that is too small for the job. Certainly, not all pipe friction can be eliminated as long as fluids must be handled in this manner, but every effort should be made to use the largest pipe that is economically feasible. Numerous tables are available for calculating frictional losses in any combination of piping. Among the most recent are the tables in the *Hydraulic Institute Engineering Data Book*[6] along with other similar hydraulic engineering references.

Before any new installation is made, the cost of larger-size piping, which will result in lower pump pressures, should be carefully balanced against the cost of a less expensive pump, a smaller motor, and a savings in power over the expected life of the system. The larger piping may cost a little more in the beginning, but the ultimate savings in power will often substantially offset the original cost. These facts are particularly true for extremely viscous fluids.

Foundation and Alignment The pump should be mounted on a smooth, solid foundation readily accessible for inspection and repair. It is essential that the driver shaft and the pump shaft are in proper alignment. The manufacturer's recommendation of concentricity and parallelism should always be followed and checked occasionally. For high temperature applications the alignments should be checked with the equipment at operating temperature.

The suction pipe should be as short and straight as possible with all joints airtight. It should not contain places where air or other entrapped gases may collect. If it is not possible to have the fluid flow to the pump under gravity, a foot or check valve should be installed at the end of the suction line or as far from the pump as possible. All piping should be independently supported to avoid strains on the pump casing.

Start-Up A priming connection should be provided on the suction side, and a relief valve should be set from five to ten percent above the maximum working pressure on the discharge side. Under normal operating conditions with completely tight inlet lines and wetted pumping elements, a screw pump is self-priming. Starting the unit may involve simply

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opening the pump suction and discharge valves and starting the motor. It is always advisable to prime the unit before the initial startup to wet the screws. In new installations, the system may be full of air, which must be removed. If this air is not removed, the performance of the unit will be erratic, and, in certain cases, air in the system can prevent the unit from pumping. Priming the pump should preferably consist of filling not only the pump with fluid but as much of the suction line as possible.

The discharge side of the pump should be vented at startup. Venting is especially essential when the suction line is long or when the pump is initially discharging against the system pressure.

If the pump does not show a discharge of liquid after being started, the unit should be shut down immediately. The pump should then be primed and tried again. If it still does not pick up fluid promptly, there may be a leak in the suction pipe, or the trouble may be traceable to an excessive suction lift from an obstruction, throttled valve, or another cause. Attaching a gage to the suction pipe at the pump will help locate the trouble.

Once the screw pump is in service, it should continue to operate satisfactorily with practically no attention other than an occasional inspection of the mechanical seal or packing for excessive leakage and a periodic check to be certain that the alignment is maintained within reasonable limits.

Care should be taken to ensure that the suction pipe is free of any solid contaminants, left over from the construction phase. Flushing of the suction pipe before connecting to the pump is recommended. In some applications, suction strainers are installed to prevent solid particles from reaching the pump. If required, the strainer can be removed after the system start-up phase is completed.

Noisy Operation Should the pump develop noise after satisfactory operation, this is usually indicative of an excessive suction lift resulting from cold liquid, air in the liquid, misalignment of the coupling, or, in the case of an old pump, excessive wear.

Shutdown Whenever the unit is shut down, if the operation of the system permits, both the suction and discharge valves should be closed. This is particularly important if the shutdown is for an extended period because leakage in the foot valve, if the main supply is below the pump elevation, could drain the oil from the unit and necessitate repriming as in the initial starting of the system.

Abrasives One other point has not yet been discussed, and this is the handling of liquids containing abrasives. Since screw pumps depend upon close clearances for proper pumping action, the handling of abrasive fluids usually causes rapid wear. Much progress has been made in the use of harder and more abrasive-resistant materials for the pumping elements so that a good job can be done in some instances. It cannot be said, however, that performance is always satisfactory when handling liquids laden excessively with abrasive materials. On the whole, screw pumps should not be used for handling fluids of this character unless a shortened pump life and an increased frequency of replacements are acceptable. In these cases, reducing the operating speed can maximize the operating life of the pump.

CONCLUDING REMARKS

As indicated, screw pumps are manufactured in a number of different configurations and designs to suit a variety of different applications. Generally, screw pumps should be considered if there is a requirement for high pressure, high viscosity, or low hydraulic pulsation levels. Screw pumps can be used with lubricating and non-lubricating products and on many special and difficult applications. The keys to the success of screw pump applications include accurate knowledge of the fluid characteristics, proper pump selection and sizing, correct system design, suitable installation conditions, and the proper operating and maintenance procedures.

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REFERENCES

1. Cooper, P., and Prang, A. J. "Enhanced Multiphase Flow Predictions in Twin Screw Pumps." *Proceedings of the 21st International Pump Users Symposium,* Texas A&M University, College Station, TX, March 2004, pp. 69–76.
2. Vetter, G., and others. Tutorial: "Multiphase Pumping with Twin-Screw Pumps—Understand and Model Hydrodynamics and Hydroabrasive Wear." *Proceedings of the 17th International Pump Users Symposium*, Texas A&M University, College Station TX, March 2000, pp. 153–169.
3. Prang, A. "Selecting Multiphase Pumps." *Chemical Engineering*, February 1997, pp. 74–79.
4. Vetter, G., and Wincek, M. "Peformance Prediction of Twin Screw Pumps for Two-Phase Gas/Liquid Flow." *Pumping Machinery—1993*, FED-Vol. 154, ASME, 1993, pp. 331–340.
5. Cooper, P., Prang, A. J., and Thamsen, P. U. "Applying Multiphase Pumps Subsea." *Proceedings of the Seventh European Congress on Fluid Machinery*, Institution of Mechanical Engineers, London, Paper No. C556/006/99, April 1999.
6. *Hydraulic Institute Engineering Data Book,* 2nd Edition, 1990, Hydraulic Institute, Parsippany, NJ www.pumps.org.

SECTION 3.8
VANE, GEAR, AND LOBE PUMPS

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ROBERT A. PLATT
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DEFINITIONS AND NOMENCLATURE

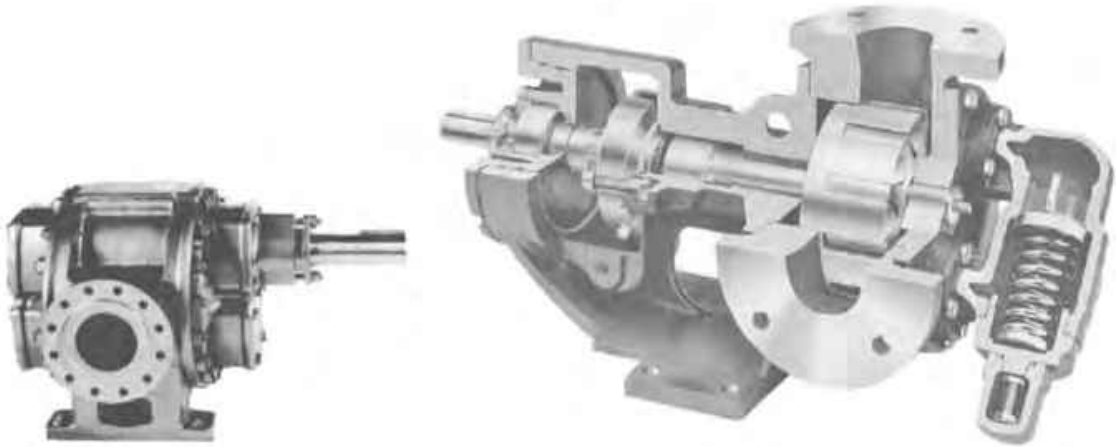
Vane, gear, and lobe pumps are positive displacement rotary pumps. The Hydraulic Institute defines them as mechanisms consisting of a casing with closely fitted vanes, gears, cams, or lobes that provide a means for conveying a fluid. Their principle motion is rotating, rather than reciprocating, and they displace a finite volume of fluid with each shaft revolution. When describing them, the general term *fluid* is used, rather than the more restrictive *liquid.* Fluid, in this case, is understood to include not only true liquids, but mixtures of liquids, gases, vapors, slurries, and solids in suspension as well.

How They Work Pumping in a vane, gear, or lobe pump begins with the rotating and stationary parts of the pump defining a given volume or cavity of fluid enclosure. This enclosure is initially open to the pump inlet but sealed from the pump outlet and expands as the pump rotates. As rotation continues, the volume progresses through the pump to a point where it is no longer open to the pump inlet but not yet open to the pump outlet. It is in this intermediate stage where the pumping volume or cavity is completely formed.

Depending on the particular pump, there can be more than one cavity in existence at any one time. As this happens, fluid also fills the clearances between the pumping elements and pump body, forming a seal and lubricating the pumping elements as they in turn pump the fluid. Rotation continues and the cavities progress, moving fluid along the way. Soon a point is reached where the seal between the captured fluid volume and outlet part of the pump is breached. At this point the vanes, gears, or lobes force the volume of captured fluid out of the pump. While this is happening, other cavities are simultaneously opening at the inlet port to receive more fluid in a continual progression from suction to discharge ports.

For optimum pumping action, the *open-to-inlet* (OTI) volume should expand slowly and continuously with pump rotation. The *closed-to-inlet-and-outlet* (CTIO) pumping cavity volume should remain constant once it is formed, and the *open-to-outlet* (OTO) volume

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(a) (b)

FIGURE 1a, 1b Typical rotary gear pumps (1b: Viking Pump)

should expand slowly and continuously with pump rotation. At no time should any fluid in the pumping chambers be simultaneously open to both the inlet and outlet if the pump is truly a positive displacement pump. When these conditions are met, the result is a very smooth continuum of flow with minimal pulsations or pressure spikes.

With rotary pumps (see Figure 1), a driver turns one shaft and rotor assembly, which in turn physically meshes with another to form the cavities that move the fluid. This is known as an *untimed arrangement*. For some applications, however, there would be problems with the gears meshing this way. For instance, stainless steel gears will gall and seize if rubbed against each other.

To circumvent this, the timed pump was developed. It uses timing gears physically located outside the pumping chamber to transmit torque between the pump shafts and synchronize the pumping elements relative to each other. By preventing them from contacting each other, they eliminate many of the problems of material compatibility, and dry running. Lobe pumps (Figure 2) are built this way.

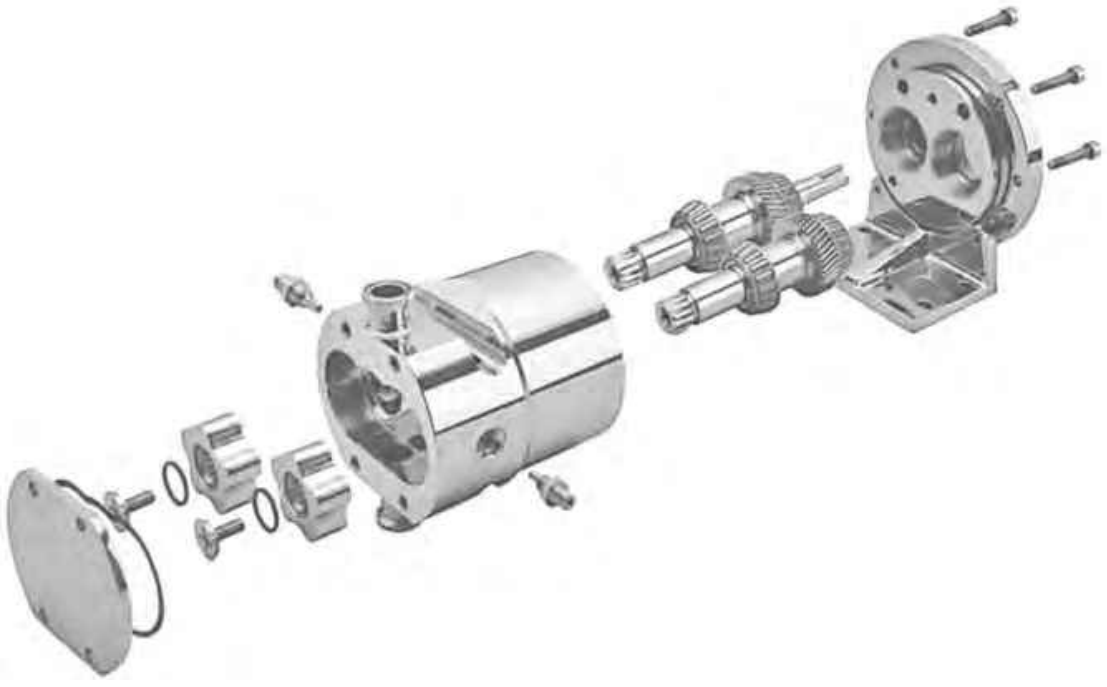


FIGURE 2 Lobe pump (Viking Pump)

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In some special cases, pressure-balancing designs, relief valve arrangements, or other such considerations have led to some designs where the pump must only operate in one direction. However, the principle of operation of most rotary pumps permits them to operate equally well in either direction.

Main Components The pumping chamber of a rotary pump is the area containing the pumped fluid while the pump is operating. Fluid enters the pumping chamber through the inlet port and leaves through the outlet port. The body is that part of the pump that surrounds the boundaries of the pumping chamber and is also referred to as a *casing, housing,* or *stator*. *End plates* can either be part of the body or separate parts, and they serve to close off the ends of the body to form the pumping chamber. End plates can also be referred to as *pump covers*.

The rotating assembly generally refers to all those parts that rotate when the pump is operating. *Rotors* are usually given more descriptive names, depending on the specific pump type. All rotary pumps employ a drive shaft to accept driving torque from the pump driver. The majority of rotary pumps are mechanically coupled to their driver with various types of couplings, but sealless magnetically driven pumps have become more common in recent years.

The cavity through which the drive shaft protrudes is called the *seal chamber*, and leakage through it is controlled by a *mechanical seal* or *packing*. In a mechanical seal, two faces with opposing axial loads are maintained in close contact with each other. When compressible packing and a stuffing box are used in place of a mechanical seal and seal chamber, the packing is compressed in the stuffing box by a gland that keeps it in intimate contact with the stationary and rotating elements. A *lantern ring* or *seal cage* is often placed between two of the packing rings to enable cooling and lubrication from an external source.

A number of other auxiliary devices and arrangements can be found in vane, gear, and lobe pumps, but two are especially characteristic of these pump types. Given the positive displacing nature of these pumps, and the potentially high pressures that can result at the outlet of the pump if there is an obstruction or blockage, *safety relief valves* must be used with all positive displacement pumps. They limit the pressure by opening an auxiliary passage at a predetermined set pressure and relieving flow back to the inlet side of the pump or, better, to the fluid's original source. They can be installed externally or serve as an integral part of the pump. One exception is flexible member pumps, which, by the nature of their design, usually do not require one, due to the resiliency and expandability of their elastomeric components.

Nonetheless, most rotary pumps are available with integral relief valves. Although these devices are a necessary investment for safety, they are not a substitute for an external system relief valve. They are a secondary safety device at best and are not intended for continuous duty, flow control, or system pressure modulation.

To reduce fluid viscosity in the pump body to facilitate a successful startup and maintain event-free operation, *heating jackets* are used. They can be either integral to the pump (either welded on or part of the body casting) or a separate bolt-on type. They are common with asphalt, gelatins, paraffin, molasses, greases, and similar fluids where, without heating, the power and torque required to drive the pump could easily overload the driver. If viscous enough, a cold startup could actually destroy the mechanical seal, shear a shaft coupling, break a drive shaft, or cause damage to the equipment some other way. Heating jackets are not intended to be the main source of heating in the system. If the system requires the fluid to be heated, some other means, such as heat tracing the system piping, must be used.

Materials of Construction When selecting materials for rotary pumps, consideration must be given to the following material properties:

- The modulus of elasticity (for deformation purposes)
- The coefficient of thermal expansion (for varying temperatures)
- The coefficient of friction (for resistance to galling when in sliding contact)

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For rotary pumps with flexible members, further consideration must be given to the materials' bulk modulus for recovery from deformation.

The close running clearances of rotary pumps require that their materials resist deformation and deflection by the various forces present when the pump is operating. If they do not, then such deformation or deflection could open the clearances and lower the operating efficiency dramatically or close the clearances and cause high mechanical loading and seizing between the moving and stationary parts.

The materials must also have compatible coefficients of thermal expansion. With the potential for deflection of the rotating parts always present, the materials selected must also have good bearing characteristics or, in particular, a good so-called "PV-value" (defined below) to resist galling and wear. This is especially important when pumping low-lubricity fluids.

Furthermore, materials used for corrosion resistance in noncontacting surfaces of centrifugal pumps may be unusable in rotary pumps where the continuous sliding contact between parts can wear away their passivating or protective layers. In general, rotary material restrictions become more severe when handling low-viscosity fluids at higher pressures and/or low lubricity fluids with abrasives. Most rotary pumps do not handle solids well, except for small trace amounts. Progressing cavity pumps, however, (see Section 3.7) are an exception and can handle very high solids content. In addition, even where there is no load-bearing contact between the rotating and stationary parts under normal conditions, the high transient forces generated at startup, shutdown, or any other unusual operating conditions (such as cavitation) must be considered when selecting pump materials.

The performance of flexible member pumps heavily depends on the material of the flexible member. Its bulk modulus must be high enough to keep distortion under pressure within functional limits and it must be resilient enough to spring back to its original shape after flexing or compressing. For instance, if, once deflected, the vanes in a flexible vane pump stayed that way, the pump could no longer operate. That is, these materials must be chosen not only to satisfy the desired hydraulic conditions, but also for resistance to deterioration from fatigue, chemicals, and the temperatures to which they may be exposed.

Vane Pumps Two basic types of vane pumps exist. The most common is the rigid sliding metal vane type, and the other is the flexible or elastomeric vane used for low viscosity (e.g., gasoline) or chemically aggressive fluids. Both are based around external sliding vanes rotating about a nonconcentric cam.

All rigid vane pumps have moveable sealing elements in the form of nonflexing blades, rollers, buckets, scoops, and so on. These elements move radially inward and outward by cam surfaces to maintain a fluid seal between the OTI and OTO sectors during pump operation. When the cam surface is internal to the pump body and the vanes are mounted in or on the rotor, the pump is called an *internal vane pump*. The OTI volume is defined by the body walls, the rotor walls, the fluid seal contact between the vanes, and the body. The body wall surface, the rotor surfaces, and the vane-to-rotor and vane-to-body fluid seal points define the CTIO volume. The body surface, the rotor surface, the vane-to-body fluid seal points, and the vane-to-rotor fluid seal points define the OTO volume.

In internal vane pumps, the volume behind the vanes must always be either a composite constant volume or else be vented, because of the piston-like pumping action of the vanes on the fluids trapped there. However, no such venting is required when the vanes are in the form of rocking slippers.

When the cam surface is external to the radial surface of the rotor and the vane, or the vanes are mounted in the body or stator, the pump is called an *external vane pump* and is illustrated in Figure 3. The OTI, CTIO, and OTO volumes are defined the same as for internal vane pumps when multiple external vanes are used. In this case, the rotor surface, the body surface, and the fluid seal points between them define the CTIO volume.

In addition to rigid, sliding metal vane pumps, flexible or elastomeric vane type pumps also exist. This kind of pump, illustrated in Figure 4, has a pumping action similar to that of an internal vane pump with the OTI, CTIO, and OTO volumes defined by the rotor surfaces, the body surfaces, the fluid seal contacts between the rotor flexible vanes, and the body surfaces.

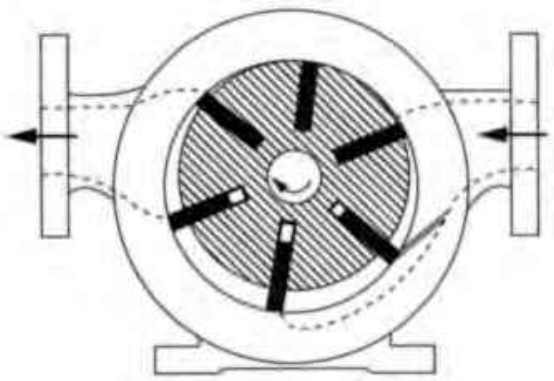


FIGURE 3 A typical external vane pump

The flexible liner pump in Figure 5 is similar in pumping action to the external vane pump, and all three chamber volumes of it are defined by the inner surface of the body, the outer surface of the liner, and the liquid seal contact between the liner and body bore. Most flexible liner pumps, unlike other rotary pumps, have at least one position of the rotor in which no fluid seal exists between the OTI and OTO volumes. The pump depends only on fluid velocity and inertia to limit backflow during this phase of rotation.

Vane pumps offer flows at up to 1,000 gpm (3,785 l/min) and pressures at up to 125 lb/in^2 (8.6 bar). They are commonly used for low-pressure transfers of gasoline, kerosene, and similar light hydrocarbons.

Gear Pumps The gear pump is the most common of all rotary pumps due to the wide variety of applications it can be used in.

Gear pumps have two gears that mesh to provide its pumping action, with one gear driving the other. The physical contact between the gears forms a part of the moving fluid seal between the inlet and outlet ports. The outer radial tips of the gears and the sides of the gears form part of the moving fluid seal between the inlet and outlet ports. The gear contact locus moves along the tooth surfaces and jumps discontinuously from tooth to tooth as the gears mesh and unmesh during rotation. These two characteristics distinguish gear pumps from lobe pumps where the rotors (lobes) are incapable of driving each other, (timing gears doing that instead,) and the fluid seal contact locus between lobes moves continuously across all the radial surfaces of the lobes.

Gear pumps are classified as *external* or *internal* (see Figure 6). External gear pumps have their gear teeth cut on their external or outside diameter and mesh about their outside diameters. Bearings (which are often product-lubricated bushings) support the shafts at both ends with the gears located between the bearings. This resists shaft deflection and contact between the gears and casing wall.

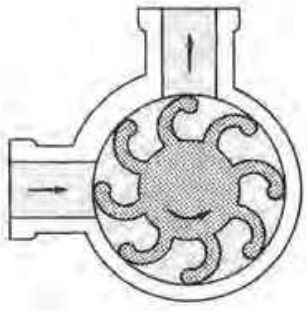


FIGURE 4 Flexible vane pump



FIGURE 5 Flexible liner pump

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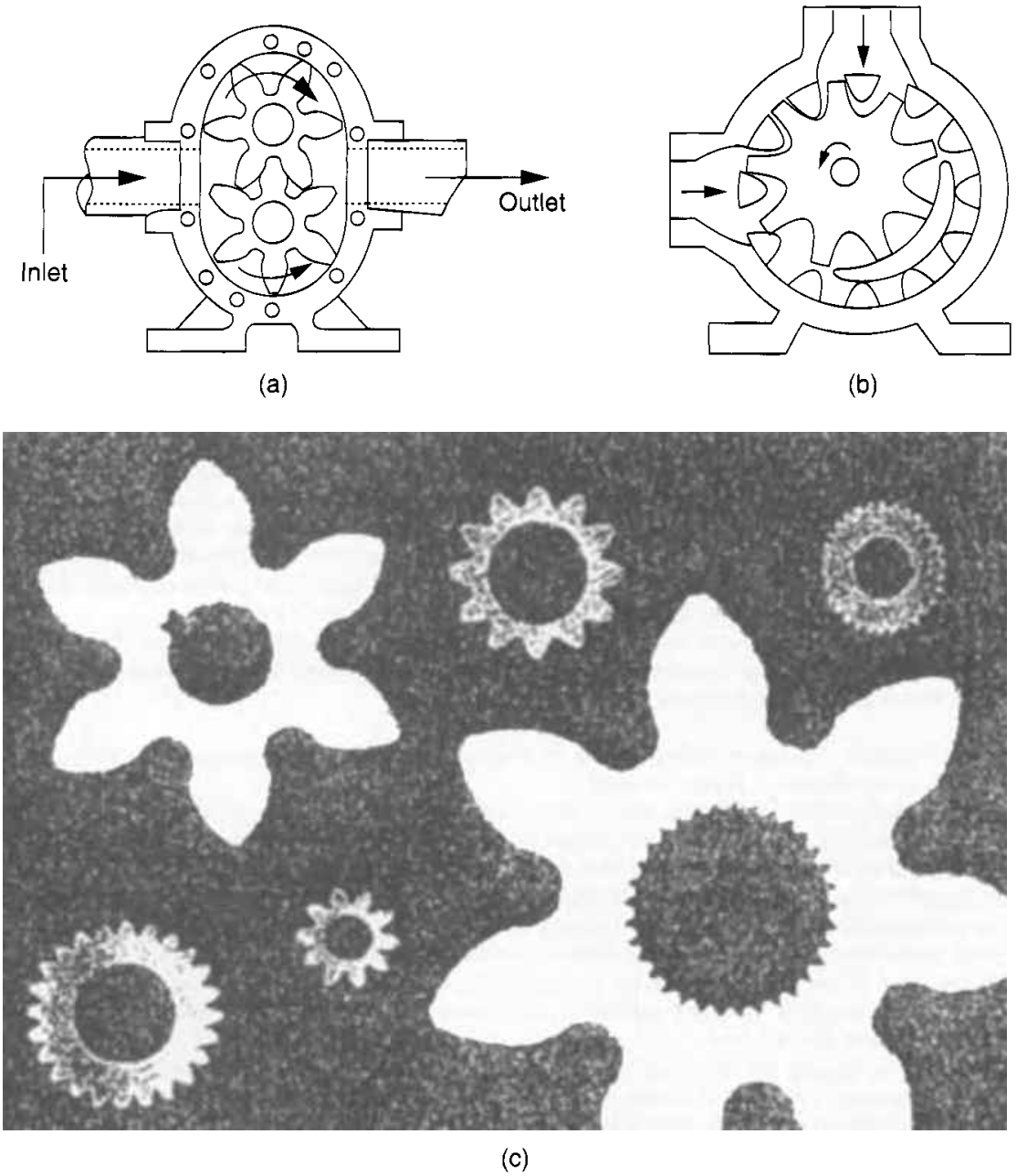


FIGURE 6 (a) external gear pump (Roper), (b) internal gear pump (Viking), and (c) various types of gears used (Roper)

Internal gear pumps, on the other hand, have one larger gear (rotor) with gear teeth cut internally on the major diameter meshing with and driving a smaller externally cut gear (idler). Pumps of this type can be with or without a crescent-shaped partition to define the OIT, CTIO, and OTO zones.

The OTI volume of the pump chamber in gear pumps is defined by the body walls and by where each tooth tip meets and seals with the body walls as it leaves the OTI volume. The fluid trapped between the gear teeth and the body walls is sealed from both inlet and outlet chambers and is the CTIO volume. The OTO volume is defined by the body walls and the gear tooth surfaces between the fluid seal points where each tooth tip leaves the body wall and enters the OTO volume and fluid seal points where the gears mesh.

A part or all of the side (or axial) surfaces of the gears run in small-clearance contact with the axial end faces of the pumping chamber. The gear teeth run in small-clearance

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contact with each other where they mesh. The tips run in small-clearance contact with the radial surfaces of the pumping chambers in their travel from the OTI to OTO volume. Load-bearing contact between the rotors or between the rotors and the stator may exist in all three of these zones, and the apertures defined by the running clearances in these zones determine the amount of slip between the OTO and OTI volumes for any given pressure difference and viscosity between them.

Both gears share pumping torque, and the proportional amount of the total torque experienced by each gear at any instant is determined by the locus of the fluid seal point between the gear teeth. As this fluid seal point moves toward the center of gear rotation, the pumping torque on that gear increases, and as the seal point moves away from the center, the torque decreases. When external timing gears are used, they transfer torque from one rotating assembly to another to safeguard against accelerated wear when dry running or handling low-lubricity or somewhat more abrasive fluids (e.g., foodstuffs). Here, bearing (bushing) life depends on projected loading and journal velocity. A product of projected loading (P, psi) times journal velocity (V, ft/min) is used to determine the ability of internally (product) lubricated bearings to operate without significant wear. In the example given with Figure 7, the idler shaft of the gear pump is shown supporting its gear and is subject to the differential pressure, resulting in the projected bearing loading as illustrated in the figure. Data exist for various material pairs running against each other. Bronze, iron, carbon, and silicon carbide are some examples of popular materials used for the internally lubricated bushings of rotary pumps. For bronze, the PV-value

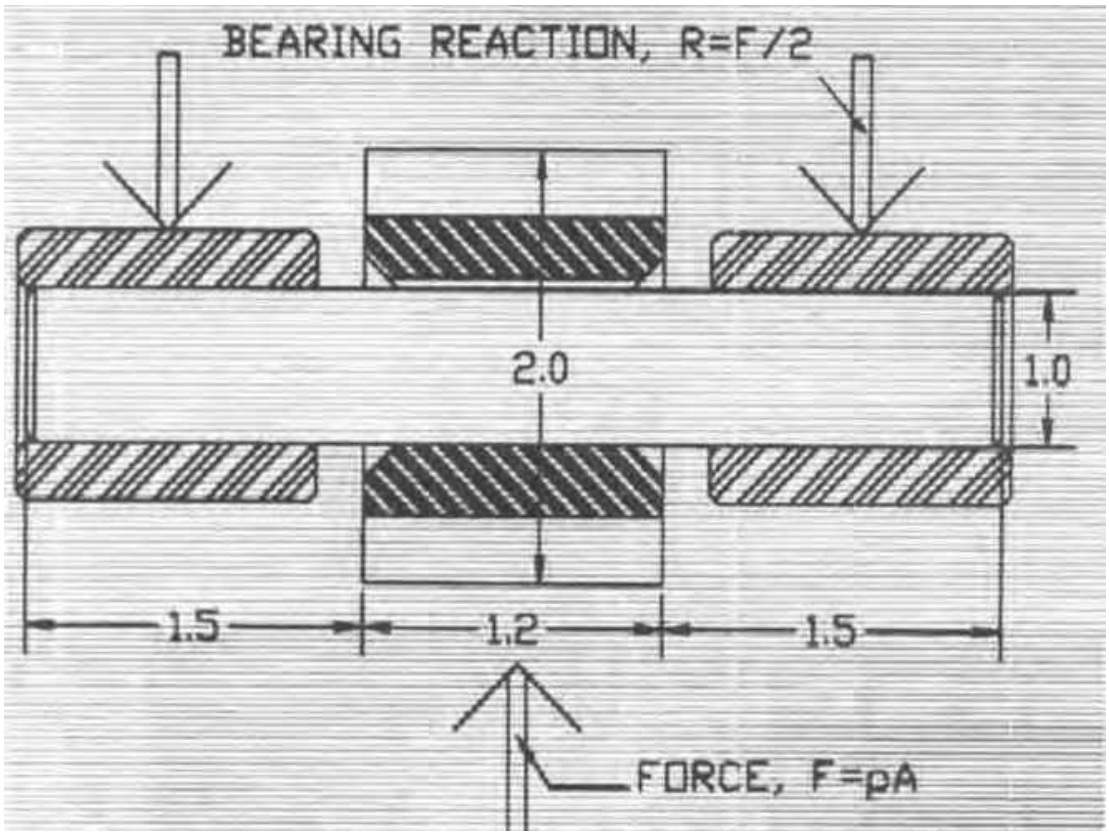


FIGURE 7 Example of dry-running PV-evaluation for a gear pump bushing. Given: rpm = 1800; suction pressure = 6 psia (0.414 bara); discharge pressure = 128 psig (8.825 barg); and dimensions are shown in inches (mm = in × 25.4). Thus, $\Delta p = (128 + 14.7) - 6 = 136.7$ psi (9.425 bar); and force on gear = 136.7 × (2 in × 1.2 in) = 328 lbf (1459 N). Half of this force is shared by each bearing, namely 164 lbf (730 N). The unit load on each bearing P = 164/(1.5 in × 1 in) = 109 psi (7.54 bar). Since the journal velocity V (ft/min) = 2π × rpm × d (ft)/2 = 2π × 1800 × [(1 in/12)/2] = 471 ft/min (2.39 m/s), PV = 109 × 471 = 51,400 psi-ft/min (18 bar-m/s). (Although the result is given here also in SI units, the PV-loading of a bearing is generally correlated in terms of psi-ft/min.)

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is 60,000; for iron it is under 30,000; and carbon is rated at approximately 120,000. Some grades of silicon carbide reach PV-values exceeding 200,000. On the other hand, plastics have very low values of PV, typically under 3000.

External gear pumps are capable of flows up to 1,500 gpm (5,680 l/min), pressures up to 500 lb/in^2 (34.5 bar), and viscosities up to 1,000,000 SSU (216,000 centistokes). They are found in both clean and dirty services serving the *original equipment manufacturer* (OEM), refinery, tank farm, marine, and API-related industries. Internal gear pumps are capable of flows up to 1,100 gpm (4,165 l/min), pressures up to 225 lb/in^2 (15.5 bar), and viscosities up to 1,000,000 SSU (216,000 centistokes). They are typically used for lower pressure transfers of fuel oils, paints, and various chemicals in the chemical processing and OEM industries.

Lobe Pumps The lobe pump (Figure 2) receives its name from the rounded shape of the rotor radial surfaces that permits the rotors to continuously overlap each other as they rotate. Lobe pumps can be either single- or multiple-lobe pumps and carry fluid between their rotor lobes much in the same way a gear pump does.

Unlike gear pumps, however, neither the number of lobes nor their shape permits one rotor to drive the other, and so all true lobe pumps require timing gears. The body surfaces, rotor surfaces, the contact between rotors, and the contact between rotor lobe ends and the pump body define the OTI volume of a pump. The contact between the lobe ends and the body wall and the adjoining body wall and lobe surfaces define the CTIO volume. The body walls, rotor surfaces, lobe-to-body wall contacts, and the lobe-to-lobe contacts define the OTO volume.

In the two-rotor lobe pumps shown in Figure 8, the torque is shared by both rotors with the proportional amount of torque dependent on the position of the rotor-to-rotor contact point on the rotor contact locus. When the contact point is at the major locus radius (maximum lobe radius of one rotor in contact with the minimum lobe radius of an adjoining rotor), one rotor sees the full pumping torque, while the other rotor feels a balanced torque. The transfer of a full pumping torque from one rotor to the other takes place as many times in each complete revolution of a rotor as there are lobes on the rotor.

Lobe pumps are capable of flows up to about 1,000 gpm (3,785 l/min) and pressures up to 225 lb/in^2 (15.5 bar). They are commonly used to pump sludge in wastewater treatment plants and in stainless steel systems for handling foodstuffs in the food, beverage, dairy, and pharmaceutical industries.

A summary of the main application parameters of vane, gear, and lobe pumps is shown in Table 1 together with similar data for other rotary pumps; e.g., screw pumps (Section 3.7).

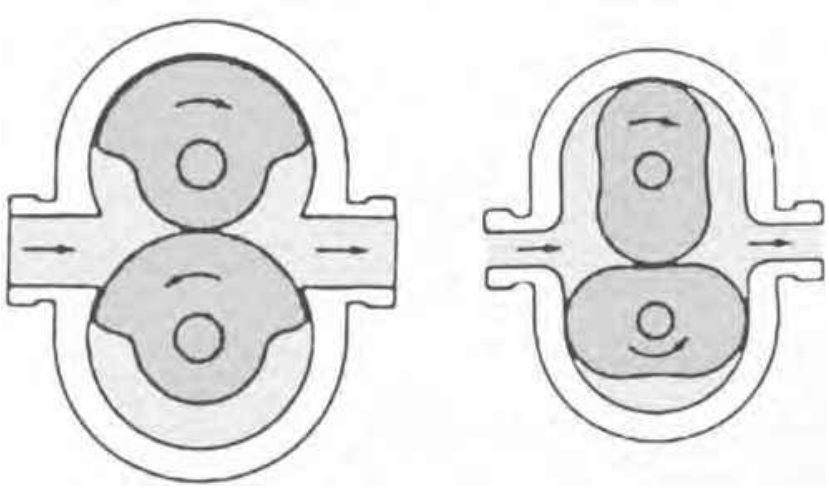


FIGURE 8 Typical single-lobe (left) and multiple-lobe (right) pumps

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TABLE 1 Application Matrix for Rotary Pumps

Pump Type	Flow, gpm	Pressure, psi	Temperature, °F	Viscocity, cP	Solids, %	Dry-run	Corrosion resistance	Pulsations	Lift ft	Price index
External gear	1 to 1000	3000	400	20 to 10,000	<1%	N	G	G	15′	110%
Internal gear	1 to 1000	3000	400	10 to 10,000	<1%	N	G	G	15′	110%
Lobe	50 to 1000	400	400	100 to 500,000	<20%	Y	G	G	10′	150%
Vane	50 to 1000	200	250	0.3 to 1,000	<1%	N	M	M	10′	100%
Diaphragm	50 to 1000	100	200	1,000 to 100,000	<20%	Y	E	B	20′	130%
Progressing cavity	100 to 1000	500	200	1,000 to 100,000	<80%	N	B	E	15′	200
Two-screw	100 to 8000	1500	650	1,000,000	<1%	N	M	G	25′	400%
Three-screw	10 to 3500	4500	550	1,000,000	<1%	N	M	E	25′	300%
Metering (diaphragm)	<1	200	400	1 to 1,000	<1%	N	E	B	5′	100%

Note: m^3/h = gpm/4.40; bar = psi/14.5; deg C = (deg F − 32)/1.8; Pa-s = cP/1000; m = 0.3048 × ft.) Values are approximate, for rough guidance only.
Code Letters: Excellent; Good; Marginal; Bad

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COMPARISONS TO CENTRIFUGAL PUMPS

A rotary pump uses mechanical and hydraulic forces to create a flow against a system backpressure. Centrifugal pumps, on the other hand, create pressure by imparting a velocity to the fluid and converting the velocity energy to a pressure energy as the fluid flows around the casing and out the discharge nozzle. A comparison of the resulting performance of these two different pump types is shown in Figure 9.

The conditions of service will usually determine the best pump for an application. For instance, for varying flow rates, a centrifugal pump would be a good choice. An example of this is a municipal water system where consistent head is maintained. By contrast, for a constant flow in the presence of varying back pressures, a rotary pump would be better. An example of this is an oil pipeline, where system economics dictate constant flow rates, regardless of any system pressure variations from changes in viscosity or the pipe diameter or condition.

Other differences exist between centrifugal and rotary pumps as well. The performance curves, affinity laws, and terminology used to describe rotary pumps are all different. And since rotary pumps are primarily for viscous fluids, the applications and markets served by these two pumps are also different. Other factors, such as vertical versus horizontal mounting, metal versus nonmetallic materials, sealless and magnetically driven versus dual-containment mechanical seals and conventional drivers, are similar whether considering a centrifugal or rotary pump.

Rotary Pump Curves Centrifugal pump curves plot the flow on the X-axis with the discharge head on the Y-axis. However, rotary pumps develop the flow against a system backpressure, rather than developing head with a corresponding flow rate. Their performance curves therefore show the flow on the Y-axis with differential pressure along the X-axis, as shown in Figure 10.

The influence of differential pressure on the flow is greatest with lower viscosity. Conversely, the maximum viscosity represents the worst (most) case from a power standpoint and is therefore the point around which the driver is sized.

Instead, a rotary pump is selected for a given differential pressure and viscosity at the nearest commercially acceptable speed (with a tendency toward the lower speeds). For viscosities under 3,000 SSU (650 centistokes), this means at synchronous motor speeds with the flow rate falling at the design point or (ideally) slightly above it. For higher viscosities,

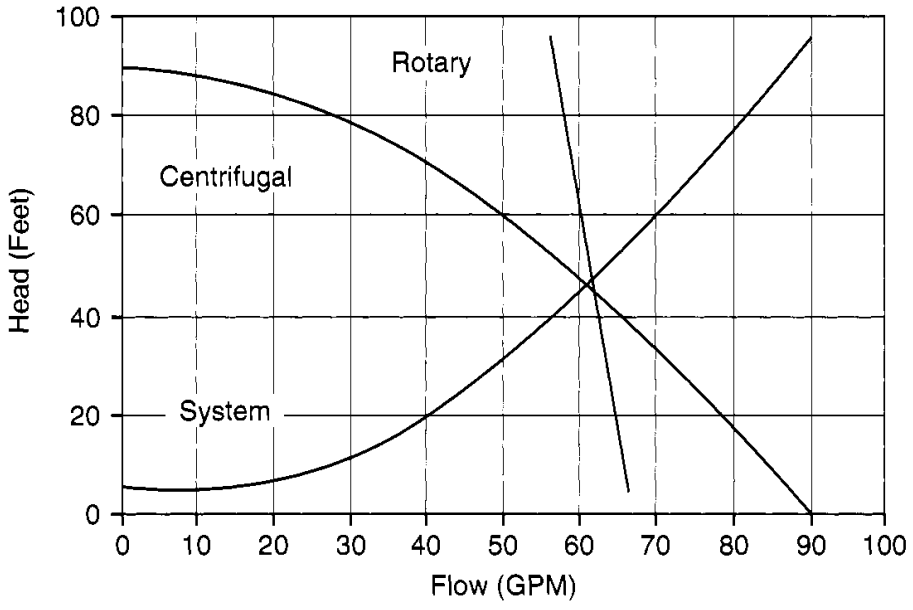


FIGURE 9 Typical performance curves (m = ft × 0.3048; m^3/h = gpm/4.403)

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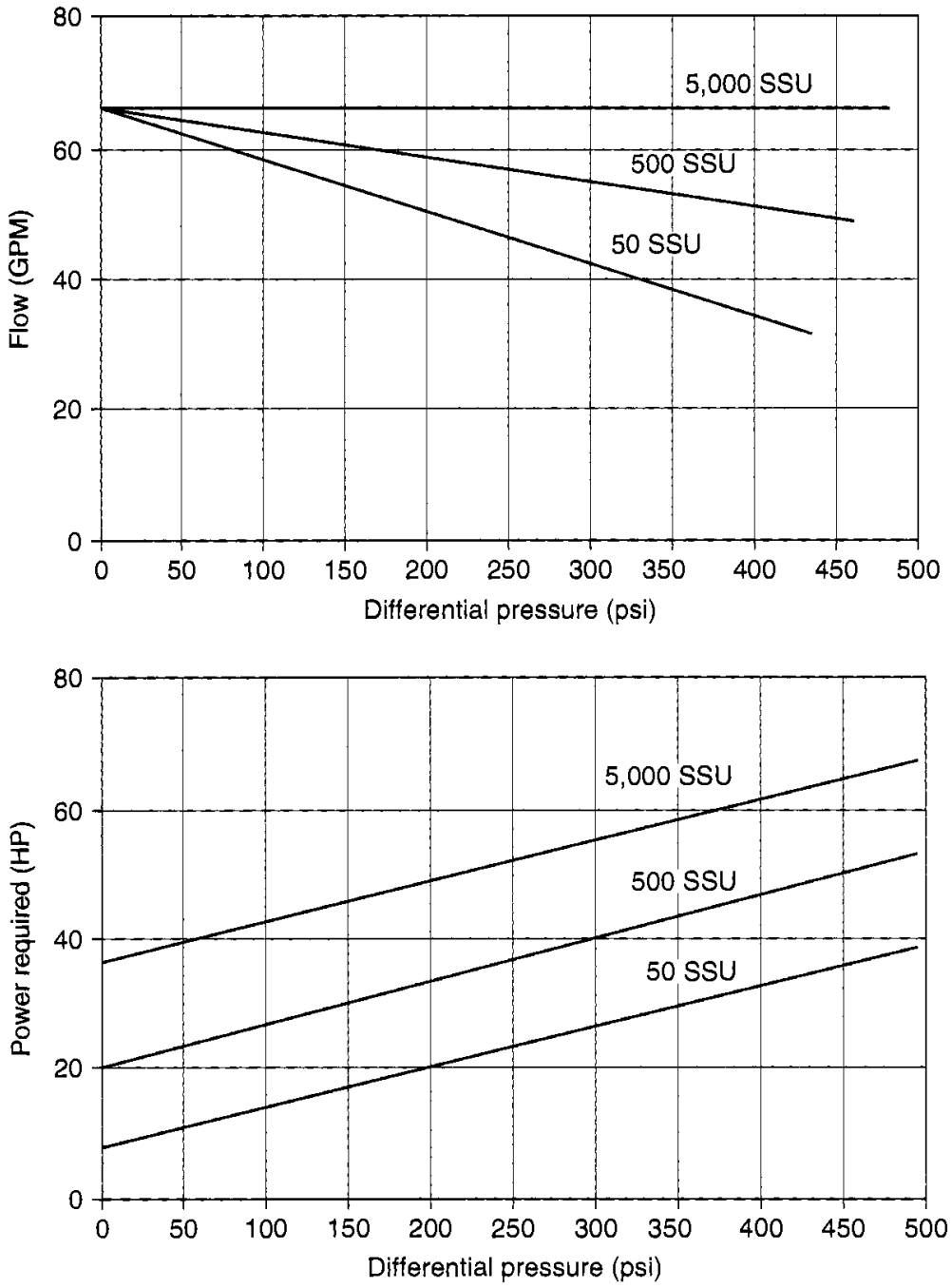


FIGURE 10 Typical rotary pump flow and power curves (bar = psi/14.5; kW = hp × 0.746; 50, 500, 5000 SSU = 7, 108, and 1080 centistokes respectively.)

rotary pumps will be run at reduced speeds, which in some cases can go well below 100 revolutions per minute (rpm).

Performance with Viscous Fluids Viscos drag can be considerable with higher viscosities. With an increasing viscosity, an increasing amount of energy must be expended to overcome these forces and produce the same amount of hydraulic work. At some point, viscosity will simply overtake the centrifugal pump, and it can no longer overcome the inertia and viscous drag losses of the fluid. The effect of this can be seen in Figure 11.

Since rotary pumps do not work this way, they are better suited for high-viscosity fluids. Certain rotary pumps, such as multiple screw pumps, can even go as high as 1,000,000 SSU

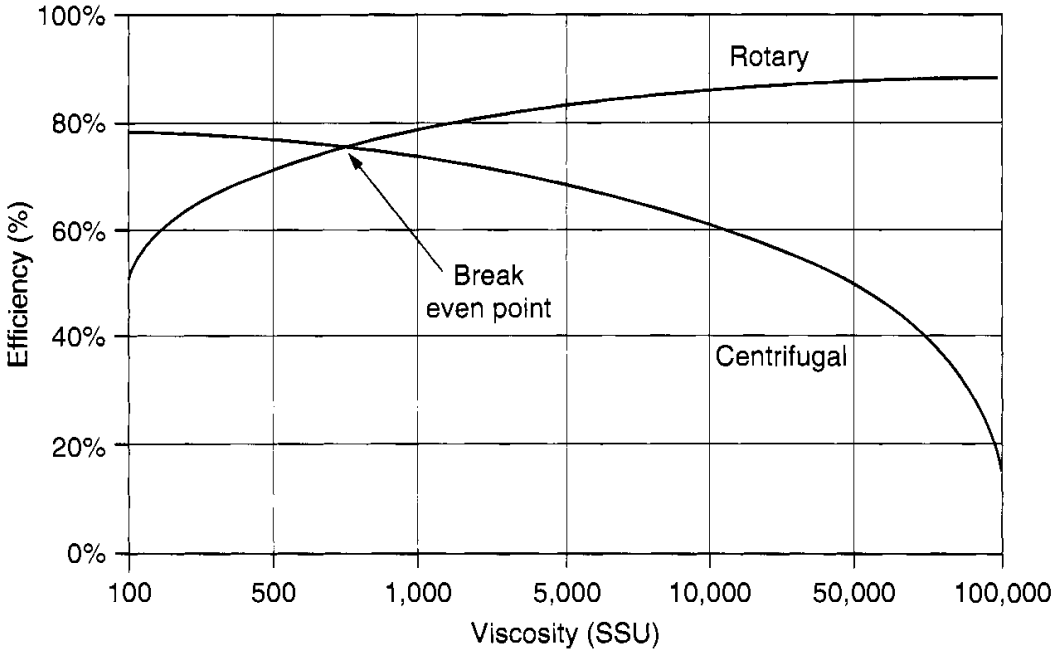


FIGURE 11 Typical efficiences as a function of viscosity. (In this range, centistokes ≈ SSU/5.)

(216,000 centistokes) without a significant deterioration of performance or efficiency. Compared to centrifugal pumps, rotary pumps are less efficient at lower viscosities but more efficient at higher viscosities. From an applications standpoint, this crossover point will vary from pump to pump and depends on many factors, but as a rule of thumb, it usually falls between 500 and 1,000 SSU (108 and 216 centistokes).

Mechanical Installation All rotary pumps, particularly rigid rotor pumps, must be installed so no mechanical forces other than those imposed by pump-generated pressures can act to warp or distort the pump chamber or rotating assembly. This is important because relatively small distortions of a few thousandths of an inch can cause interference between rotating and stationary parts and generate high wear rates or pump damage.

To avoid such distortions, the pump must not be installed with overly long, rigid fittings or in such a way that the pump body supports the weight of the piping system. Large mechanical forces that distort the pump body may also cause distortions of the mechanical seals and accelerate wear of the bearings. A cardinal rule, then, is that a rotary pump must not be rigidly coupled to the piping system in a way that causes it to support the weight of the piping or otherwise exposes it to any forces from thermal expansion. Poor mechanical installation is the cause of many field problems.

Dry Running Most rotary pumps will be damaged if allowed to run dry. As long as the system provides a positive static pressure on fluid present at the inlet, the pump should prime and not run dry. If a negative inlet gage pressure, or suction lift, is present at the pump inlet, the installation should be checked to ensure airtight seals throughout the entire upstream piping.

The sealing arrangement on the pump must also be checked for any leakage of air or gas into the pump through the seal. If these conditions are satisfied, fluid will enter the pump soon after pump operation starts because of the vacuum generated by pump operation. If these conditions are not met, air or other gases will flow through leaks in the inlet system or seal to satisfy the pump flow rate requirements and the pump will run dry or partially dry.

Suction Strainers Like most other pumps, rotary pumps last longer when handling clean fluids. Nonetheless, this is an ideal scenario, and the pump will more realistically encounter dirty or abrasive-laden fluids of varying degrees. Fine particles and abrasives will cause wear in the close clearances of the pump, which eventually reduces pump flow rate by increasing the slip through the increased clearances.

As such, all rotary pumps should have a suction strainer to exclude larger materials such as welding slag, scale, rust, chips, rags, bolts, nuts, and so on. Since a suction strainer contributes to suction line losses, this reduces the net inlet pressure available. The finer the filtration, the greater the restriction and the more frequently it must be maintained. This leads to a trade-off between the cost of the added maintenance versus the cost of replacing the worn pump parts earlier than they would be otherwise.

When pumping fluids over 5,000 SSU (1080 centistokes), the finest strainer screen practical is a $\frac{1}{16}$-inch (1.5 mm) perforation. Strainers and filters not only require periodic maintenance, but should also be instrumented accordingly. It is important for the user to provide some means of monitoring, such as a differential pressure gauge or switch, since a clogged strainer will cause the pump to cavitate or even run dry.

Entrained Air and Dissolved Gases An important consideration with rotary pumps is the amount of entrained air or gas in the fluid. It is generally neglected, since a rotary pump cannot become vapor-bound the way a centrifugal pump can. Nonetheless, with entrained air present, there can be a loss in the outlet flow. If the entrained air is a large enough percentage, there may be unacceptable noise and vibration levels as well.

For example, if a fluid contains 5% entrained gas by volume and the suction pressure is atmospheric, the mixture is 95% liquid and 5% gas. This mixture fills up the moving voids on the inlet side, with 5% of the space filled with gas and the remainder with liquid. Therefore, in terms of the amount of liquid handled, the output is reduced directly by the amount of gas present, or 5%. Unless this is understood up front, it could lead to a less than satisfactory output flow rate through no direct fault of the pump itself.

Entrained air is common in systems where the liquid is cycled frequently. In many cases, the foaming or air entrainment cannot be avoided, such as with the lubrication system on a large reduction gearbox. Instead, the condition must be known and well understood before selecting a pump for the application.

If dissolved gases (gases different than the fluid's own vapor) are present in the fluid, the effect on the output flow is the same as with entrained gases. This is because the dissolved gases will come out of solution when the pressure is lowered, just as the fluid's own vapor will. This will have the same net effect as the entrained gas and will occupy the available displacement capacity. Although the fluid mass transfer rate will not be affected, this is likely to be small comfort since the measured liquid displacement will be reduced.

Noise Pumps are often the most offensive noise sources in hydraulic machinery. High-pressure pulsations and heavily loaded sliding elements within the pump produce broadband, high-energy airborne noise. Vane, gear, and lobe pumps (as well as the screw pumps of Section 3.7), however, are among the lowest noise producers of any fixed displacement-type pump. Flow is delivered continuously without the variations that produce noise in conventional hydraulic pumps. Pumping elements utilize a fluid film, reducing the sliding contact, and the visco-elastic properties of the fluids they pump help dampen whatever fluid-borne pressure pulsations are present. These design features are responsible for the wide use of these pumps wherever noise is critical. For instance, they are widely installed on die-casting machines, plastics equipment, presses, and an enormous variety of machine drives and machine tools.

Inlet Pressure The absolute pressure above the vapor pressure *available* at the pump inlet must always exceed the absolute pressure above the vapor pressure *required* by the pump.

Rotary pumps are not effective at stopping the flow through them. In applications where the flow cannot be permitted through an idle pump, or where inlet or outlet static

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pressure heads exist, check valves in the system must be used to stop the flow. For example, in intermittent deliveries where the pump is "lifting" liquid from a source below its inlet without any valving in place to address this when the pump is stopped, the fluid will gradually drain backward through the pump and back to its source.

Cavitation The creation of vapors and the subsequent collapse of the vapor bubbles upon reaching the higher-pressure discharge side of the pump is known as *cavitation*. Cavitation forces the liquid into the vapor voids at high velocities and produces local pressure surges of high intensities impinging on the pump surfaces. These forces can exceed the tensile strength of the metal, eroding it in the process. Left uncorrected, cavitation can cause pitting of the vanes, gears, or lobes and interior casing walls; bearing failures; and even shaft breakage. In addition, cavitation causes noise, vibration, and a loss of output flow. The bigger the pump, the greater the noise and vibration can be.

Lowering the static suction lift of a system, increasing the suction pipe diameter, and simplifying the suction piping layout can reduce cavitation by raising the net inlet pressure or NPSH available. In reality though, the net inlet pressure available is not something that can easily be altered. Instead, the pump vendor will usually be asked to select a pump with a lower net inlet pressure required. The best way to do this is to select a larger pump running slower. This will give the fluid more residence time to fill the void on the suction side, and with larger internal passages and ports, pump entry losses will be reduced as well.

FLUID CONSIDERATIONS

Very few commercially handled fluids are homogenous liquids. In actual systems, most fluids have air or other gases dissolved or entrained in them. In other cases, solids (abrasive or nonabrasive) may be in the fluid. The variety of fluids handled by rotary pumps requires that each pump application be uniquely considered in terms of the effects of the pumped fluid on pump performance. General summaries of the effects of various fluid characteristics on pump performance are covered in the following paragraphs.

Temperature The temperature of the pumped fluid affects pump performance in three main ways. If the pump is to handle fluid at temperatures considerably different from ambient ones, the materials of construction, both in the pump and in the seals, and the operating clearances of the pump must be selected to provide the desired operating characteristics at the temperature of operation. The selection process becomes even more stringent when the pump is operated over a wide range of temperatures. Such a use may preclude the use of pump construction materials having high thermal coefficients of expansion. Furthermore, in the actual application, it may be necessary to preheat or precool the pump to the operating temperature before the pump is started in order to avoid thermal shocking the internal components when the fluid enters the pump. If this is not done, the resulting rapid heating or cooling of the pump members from the inside out can damage the pump. Preheating or pre-cooling can usually be accomplished with a secondary medium such as a heating jacket.

Viscosity Another effect of fluid temperature is on the viscosity of the fluid. For the majority of the commercial liquids, the viscosity increases with decreasing temperature and decreases with increasing temperature. If the pump application requires usage over a wide temperature range, the highest viscosity at this temperature range must be known to determine whether the pump is operating below the upper speed limit imposed by this fluid viscosity. The effects of viscosity on speed and the net inlet pressure (above the vapor pressure of the liquid) required is shown in Figure 12.

Lubricity Still another important fluid characteristic is lubricity. Many rotary pumps are not designed to operate on liquids with no lubricity. Either the wear rate or the pump mechanical friction may be inefficiently high if these pumps are used on nonlubricating

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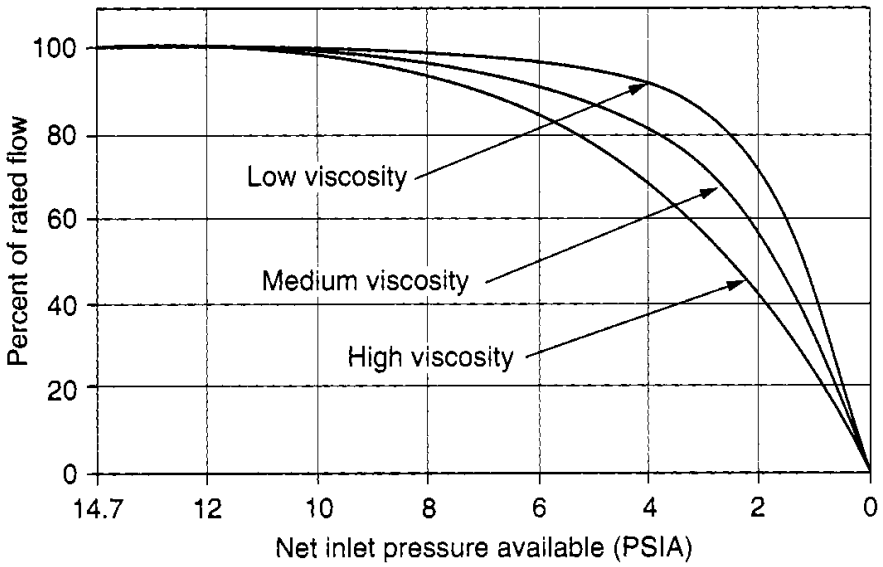


FIGURE 12 How viscosity and net inlet pressure available influence flow (bar = psi/14.5. Expressed as head of pumped liquid, the net inlet pressure is called NPSH.)

fluids. Vane pumps are the exception here; for they can handle even liquid CO_2, a very poor lubricating substance.

Non-Newtonian Fluids The viscosity of most fluids is unaffected by agitation or shearing as long as the temperature remains constant. They are known as *true* or *Newtonian fluids*. Certain fluids, however, change viscosity as a shear is applied at a constant temperature. The viscosity of these fluids will depend upon the shear rate at which it is measured, and these fluids are termed *non-Newtonian*. Some examples are cellulose compounds, glues, greases, paints, starches, slurries, and candy compounds.

If a fluid is non-Newtonian, the viscosity under actual pumping conditions must be determined and can vary quite a bit from the viscosity under static conditions. An example is grease, where the static viscosity is around 20,000 SSU (4300 centistokes). Under actual pumping conditions, however, the viscosity is closer to 500 SSU (108 centistokes). Without realizing this, a larger, slower pump may have been offered when a smaller, less expensive pump running at a higher speed would have been acceptable.

Tests or computations should determine the effective fluid viscosity under actual operating conditions. If the pump is to operate over a range of speed and pressures, the maximum and minimum effective viscosities over this range should be known to allow for the additional slip or horsepower that may be required and to ensure the correct operating speed and pump driver power selections. If these data are unavailable, as is often the case, then data from a similar installation can be helpful. Ideally, the information will include the pump size, flow rate, and speed as well as the NPSH available and the pressure drop over a specific length of piping. With this information, assumptions can be made regarding the actual viscosity.

Corrosiveness Knowledge of the corrosiveness of the pumped fluid on the pump materials in contact with the fluid is important to satisfactory pump application. The clearances in rotary pumps are small, and corrosion rates of only a few thousandths of an inch per year may seriously affect the efficiency of the pump, particularly when it is handling low-viscosity fluids. In addition, general compatibility of the fluid with the materials of pump construction must be considered. For example, a certain solvent to be pumped may soften, dissolve, or destroy the elasticity of flexible members in the pump, including those flexible members used in the sealing arrangements. Hence, the effect of the pumped fluid on the physical and chemical state of materials used in pump construction should be considered when choosing a particular application.

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Abrasiveness Abrasives present a problem for every pump type, and rotary pumps are particularly vulnerable. With small operating clearances, the tendency of the abrasive is to wear down and open the tight clearances of the pump. Most rotary pumps' internal clearances are long, narrow, rectangular cross sections that can be modeled as two parallel flat plates, with one plate stationary and the other moving. These clearances range from essentially zero to a few thousandths of an inch. Thus, even minor variations in manufacturing tolerances or the effects of wear over time can cause considerable variations in the percentage change of the aperture volume.

Because rotary pumps depend upon close clearances for their pumping action, whenever abrasive or dirty fluids are encountered, accelerated wear can be expected. Wear is difficult to predict because it depends on many variables that are difficult to measure (such as particle size, size distribution, or whether smooth or jagged shaped).

Classic theory tells us that laminar slip flow Q_s between two plates follows the general equation:

$$Q_s = k\left\{\frac{P_d \times w \times d^3}{v \times l}\right\} \tag{1}$$

where Q_s = slip through the clearance, gpm (m^3/hr)
k = a constant
P_d = differential pressure, lb/in^2 (bar)
w = width of clearance, in (mm)
d = clearance gap or depth, in (mm)
v = absolute viscosity, lb-sec/ft^2 (centipoises) [Note: (absolute viscosity in centipoises) = (kinemetic viscosity in centistokes) × sp. gr.]
l = length of clearance, in (mm)

Rotary pumps rely on close running clearances to seal between the suction and discharge pressures. This presents certain manufacturing trade-offs. For instance, the clearances must be tight enough to yield an acceptable volumetric efficiency with lower viscosities, yet large enough to facilitate internal lubrication with higher viscosities. For a given pump with a fixed viscosity and differential pressure, the slip flow is a function of the cube of the clearances only.

Unlike a centrifugal pump impeller, which can be trimmed to about 85% of its maximum diameter, rotary pump gears, screws, or lobes cannot be trimmed. As Figure 13 shows, up to a certain critical point, wear has little effect on the flow. Beyond this point, which varies from case to case, performance starts to deteriorate rapidly.

Much progress has been made in the use of harder and more abrasive-resistant materials for the pumping elements. However, rotary pumps should not be used for abrasive fluids, unless a decreased pump life and/or increased maintenance intervals are acceptable.

Other fluid characteristics could be important in the selection and application of vane, gear, and lobe pumps. Some liquids are sensitive to shear and must be handled in pumps with low shear rates. Others cannot tolerate exposure to the atmosphere, either because they can explode, crystallize, and damage the seal arrangement or otherwise interfere with the pump operation. Rotary pumps for these applications should have only static seals or multiple-section rotary seals with a protective liquid or gas in the seal zones between the fluid and outside atmosphere. In aseptic pumps, for example, stationary and moving seals can be designed with multiple seals to enable the use of steam under pressure as a sterile barrier between the pumping chamber and the atmosphere. Pumps in industrial applications where it could be dangerous if the fluid were exposed to the atmosphere may use multiple sealing arrangements with an inert liquid under pressure acting as a barrier between the pumping chamber and the outside atmosphere.

For the effective application of rotary pumps, the temperature, viscosity, lubricity, corrosiveness, and nonliquid content of the pumped fluid must be known, along with any special characteristics of the liquid, such as shear or atmospheric sensitivity. With these known, the proper pump type, seal arrangements, horsepower requirements, and speed can be determined.

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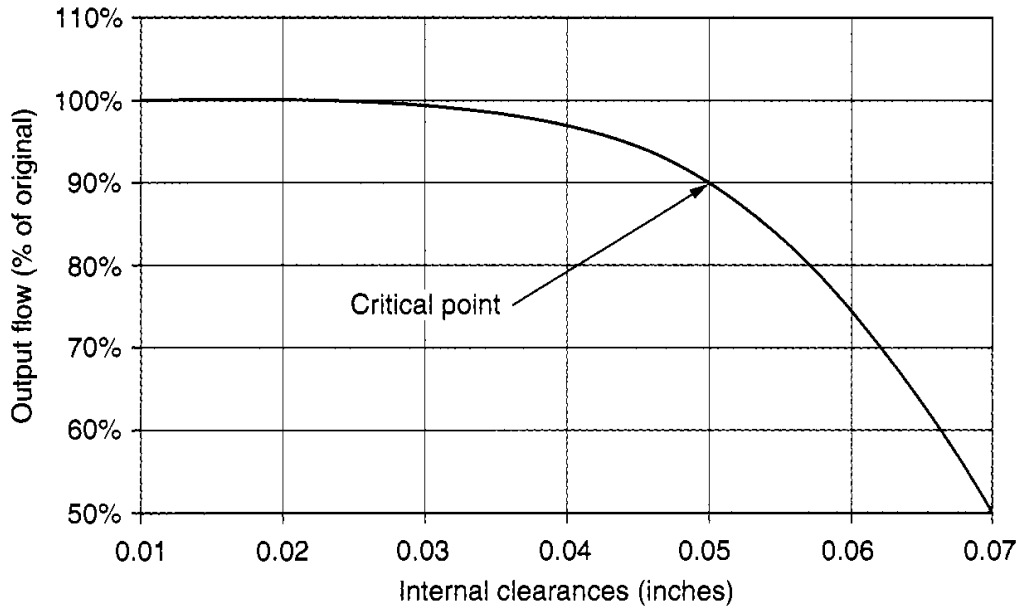


FIGURE 13 Output flow rate as a function of clearance size and wear

Vapor Pressure A second effect of fluid temperature is on the vapor pressure of the fluid. Vapor pressure is the pressure that must be maintained at a given temperature to prevent the liquid from partially vaporizing into a gaseous state. Vapor pressure increases with temperature, and if the local pressure falls below the vapor pressure, vaporization and cavitation will occur.

Vapor pressure is particularly important when handling hydrocarbons and petrochemicals, which can have very high vapor pressures. For instance, low-sulfur crude oil vapor pressures can be as high as 100 lb/in^2 absolute (6.89 absolute) under summer ambient temperatures. Unless this is known when selecting a pump, the results could be a pump that works in cold weather but cavitates in warm weather.

OPERATING CHARACTERISTICS

The operating characteristics covered in this section assume the fluid is a true incompressible liquid with a viscosity independent of the rate of shear (shear strain). Common fluids used in testing are either a light lubricating oil or cool water with small amounts of soluble oil added for lubricity.

Displacement The *theoretical* or *geometric displacement* q_0 of a rotary pump is the fluid volume transferred during each revolution. For any given pump, the displacement depends only upon the physical dimensions of the pump elements and the pump geometry and is independent of other operating conditions. In those pumps designed for variable displacement, the pump usually is rated at its maximum displacement. Thus, neglecting slip, the *theoretical displacement volume flow rate* of the pump Q_d is the *theoretical displacement per revolution* q_0 multiplied by the rotative speed or rpm; namely, $Q_d = q_0 \times rpm$.

Slip Flow slip, Q_s, is an important aspect of rotary pump performance. It is defined as that quantity of fluid that slips from the OTO volume to the OTI volume per the unit of time. Slip is a function of the clearances between the rotating and stationary members, the differential pressure between the OTO volume, the OTI volume, and the fluid viscosity. Hydraulically, it is equivalent to a bypass line from the pump outlet back to the inlet. The most common unit of measuring slip is U.S. gallons per minute (cubic meters per hour). Slip is essentially independent of *RPM*.

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The major slip paths through the pump in the presence of a positive differential pressure are the clearances between the end faces of the rotors and the endplates of the pump chamber, and those between the outer radial surfaces of the rotors and the inner radial surfaces of the chamber. If the differential pressure across the pump remains constant during a revolution, then the instantaneous slip rate usually varies throughout the revolution. This variation in the slip is caused by the same effect that would be produced if the physical dimensions of the equivalent bypass around the pump were varied as a function of the angular rotation of the drive shaft. This is also one of the common causes of flow pulsation in rotary pumps. It is particularly dominant when pumping low-viscosity fluids at high pressures.

The average slip for any set of operating conditions can be found by measuring the flow rate from the outlet port (assuming an incompressible liquid) and subtracting that flow rate from the theoretical displacement flow rate Q_d that would otherwise be expected at those operating conditions. The paths can also vary considerably in length with the changing positions of the rotary and body-sealing surfaces during rotation.

The effect of pressure on the slip is complex. The primary effect is direct in that slip increases in direct proportion to pressure. However, several secondary effects should be considered as well. The first is the effect of pressure differences across the pump on the dimensions of the slip path. This occurs because of the deflection of pump elements as a function of pressure. This is relatively small in rigid element pumps but can be significant in flexible vane pumps where the pressure can cause the vanes to flatten out and move away from the body walls. In addition, while the slip may increase in flexible member pumps at high pressures, in rigid rotor pumps it can actually decrease as clearances close, due to the high-pressure deflection of the rotors.

Another secondary consideration is the indirect effect of pressure on the fluid velocity through the slip paths. At any given viscosity, the flow through these paths can have the characteristics of turbulent, laminar, or slug flows. The majority of practical applications would require that the slip be a minor percentage of the pump displacement. To remain so, the velocity of fluid flow through slip paths would normally be in the laminar flow region, and the slip would then be directly proportional to the pressure difference. A pressure increase could cause a change to turbulent flow and a corresponding change in the slip as a function of pressure.

Also, an indirect effect of pressure exists on the effective compression ratio of compressible fluids. The compression ratio reduces the amount of net volume flow through the outlet port relative to the displacement of the pump. Although not a true slip in the sense discussed up to this point, the type of slip caused by this effect reduces the net volume delivered through the outlet port and consequently affects the volumetric efficiency. This effect is a secondary effect in most liquids but can become a large component of the slip in aerated or compressible liquids. An increase in the compression ratio caused by an increase in the pressure difference causes an increase in slip from this effect.

Flow Rate and Displacement The flow rate or *capacity* Q_c of a rotary pump is the net quantity of fluid delivered by the pump per the unit of time through its outlet port or ports under any given operating condition. When the fluid is incompressible, the flow rate is numerically equal to the total volume of liquid displaced by the pump per the unit of time minus the slip, all expressed in the same units. When a rotary pump is operating with zero slip, the theoretical or geometrical displacement Q_d of the pump becomes the flow rate Q_c. A common unit of flow rate is U.S. gallons per minute (cubic meters per hour):

$$Q_c = Q_d - Q_s \tag{2}$$

The theoretical displacement q_0 per revolution represents the fluid confined within the gear spaces and can be found by integrating the differential rate of a net volume transfer over one shaft revolution with respect to the angular displacement of the drive shaft through any complete planar segment taken through the pump chamber between the inlet and outlet ports. This integration depends on the gear shape, which is defined by the involute; and so the exact calculation of this volume is mathematically involved. Therefore, in most practical applications, an approximation can be made with reasonable accuracy, as the following example illustrates.

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Gear pump displacement calculation (approximate method). Assume an external gear pump with two gears measuring 3 in (7.62 cm) OD, and 2 in (5.08 cm) ID (root), and 4 in (10.16 cm) gear width W. Assuming that the liquid occupies half of the volume between the OD and ID cylinders (the other half being the solid material of the teeth), the volume of fluid between OD and ID for the two gears is

$$2 \times \frac{1}{2} \times \frac{\pi}{4} \times (\text{OD}^2 - \text{ID}^2) \times W = 15.71 \text{ in}^3(257.4 \text{ cm}^3) \text{ per revolution.}$$

Therefore,

$$q_0 = \frac{15.71}{231 \text{ in}^3/\text{gal}} = 0.068 \text{ gal/rev.} = \frac{257.4}{1{,}000{,}000 \text{ cm}^3/\text{m}^3} = 0.0002574 \text{ m}^3/\text{rev.}$$

For a rotative speed of 1000 *rpm*, the *theoretical displacement flow rate* Q_d (which does not account for slip) thus becomes

$$Q_d = q_0 \times \text{rpm} = 0.068 \times 1000 = 68 \text{ gpm} = 0.0002574 \times 1000 \times 60 \text{ min/h} = 15.44 \text{ m}^3/\text{h}$$

From a practical standpoint, pump displacement often cannot be computed precisely from the geometry of the pump and is instead established by testing. In either case, the manufacturer will usually state this because it is needed to determine the efficiency of the pump under various operating conditions.

Speed Considerations Centrifugal pumps typically run at synchronous motor speeds. The flow and head are fine-tuned by such things as trimming the impeller or underfiling the impeller vanes. Rotary pumps, on the other hand, operate with their internal dimensions fixed. Performance is usually fine-tuned externally by speed adjustments where the operating speed N is the number of revolutions of the driving or main rotor per unit time. When no gear reduction or increase exists between the drive shaft and the main rotor, the speed is measured or set at the drive shaft.

The most common unit of speed is rpm, and its direction is usually described as either clockwise or counterclockwise when viewed from the front or drive end of the pump looking from the driver.

Usually, no minimum speed exists for a rotary pump, but certain vane pumps that depend on a centrifugal force to draw the vanes out of their slots do have minimum speed requirements. Another exception is with low-viscosity fluids at high pressures where the slow speed enables slip to equal the theoretical displacement, resulting in no net output flow. To avoid this situation and assure the pump is running at a suitable volumetric efficiency, the pump should always run within its permitted speed range. Figure 14 shows a typical speed versus viscosity curve for a rotary pump.

Since flow is directly proportional to speed, it can be tempting to increase the pump speed above its maximum speed to obtain more flow. However, besides all the aforementioned concerns, one other item must be considered. With any rotary pump, as the cavities rotate, they present a void for the incoming fluid to fill. This void is available for a fixed amount of time, and the fluid in the suction chamber must accelerate to fill this void in the available time. The higher the fluid viscosity, the more energy is required to accelerate the fluid to fill the void. If the fluid cannot fill the void in the time available, a partial void and cavitation will occur. Recognizing a need to limit pump speed, most manufacturers provide charts showing the maximum speed, which decreases with increasing viscosity of the liquid. This speed is related to the required excess of the inlet total pressure P_1 above the vapor pressure p_v of the liquid (called NPSH when expressed in terms of the equivalent head of the liquid) increases with both the pitch line velocity V_p of the gears and the viscosity. Thus, if the available level of P_1 is limited, the pump speed is also limited accordingly. For external gear pumps, this relationship can be approximated by the formula:

$$P_1 - p_v = V_p^{0.8} \times \nu^{0.1}/61 \text{ psi (Multiply by 6.895 to obtain the answer in kPa)}$$

where ν is the kinematic viscosity in SSU, and

$$V_p = D_p \times \text{rpm} \times 3.1/n_t$$

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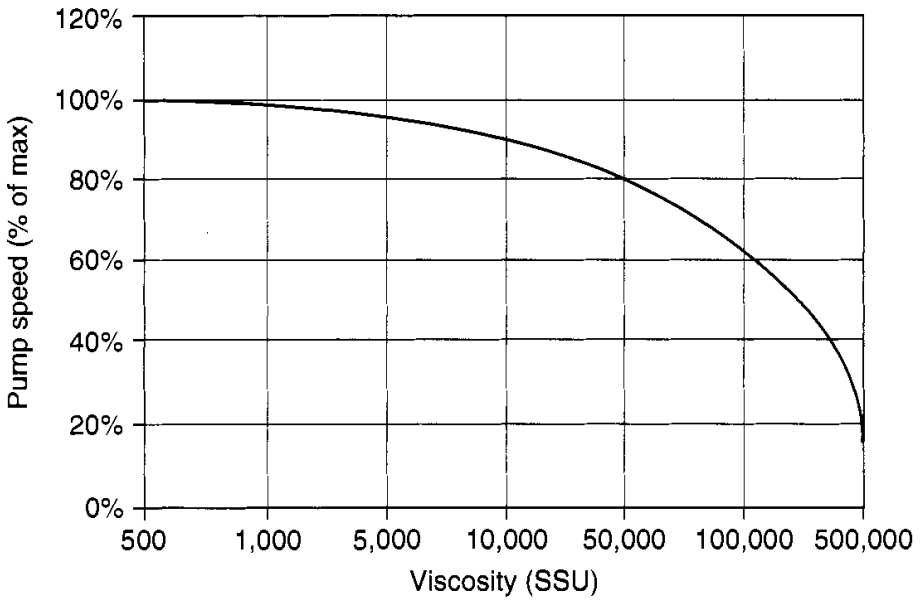


FIGURE 14 Typical speed reduction as a function of viscosity (In this range, centistokes = SSU/4.6)

and D_p is the pitch diameter of the gears in inches (= mm/25.4) and n_t is the number of teeth on a gear.[1]

Pressure The absolute pressure of the fluid at any location in the pump, expressed in lb/in^2(bar), is the total pressure there and the basis for all other pressure definitions associated with pump operation. Helping to simplify matters, the velocity pressure component of the fluid is usually small enough relative to the total pressure to be neglected.

The *inlet* or *suction pressure* p_{in} is the total pressure at the inlet to the pump. In common practice, the inlet pressure can be variously expressed as absolute pressure (lb/in^2 absolute or bar), as positive or negative gage pressure (lb/in^2 gage or bar), or as vacuum (inches or millimeters of mercury).

The total *differential pressure* Δp (or P_d) is the algebraic difference between the outlet pressure and the inlet pressure, with both expressed in the same units. This differential pressure is used in the determination of power input and in evaluating the slip characteristics of the pump:

$$\Delta p = p_{out} - p_{in} \tag{3}$$

The *net inlet pressure* $p_{net\,in}$ of a rotary pump is the difference between the inlet pressure expressed in absolute units and the vapor pressure of the fluid expressed in absolute units:

$$p_{net\,in} = p_{in} - p_{vapor} \tag{4}$$

Expressed as head of the pumped liquid, this is the net positive suction head, NPSH. The required NPSH or *net inlet pressure required* p_{nipr} is the minimum net inlet pressure that can exist without the creation of enough vapor in the inlet to interfere with proper operation of the pump. The inlet pressure usually is not the true minimum pressure in the OTI volume. The flow of fluid in its passage through the OTI volume causes a fluid friction pressure loss, which causes the pressure to drop below the inlet pressure at some point in the OTI volume or inlet chamber. This loss increases with fluid velocity and hence with pump speed and fluid viscosity. It is a function of the pump geometry, which determines the fluid path lengths and local velocities where the fluid changes direction in flowing around curves or corners in the pump OTI volume boundary.

The required net inlet pressure is established by the manufacturer for particular speeds, pressures, and fluid characteristics. In practice, the pump user is warned that the

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net inlet pressure is near or at the required net inlet pressure by noisy and rough operation of the pump caused by the incomplete filling of the CTIO volume with an accompanying reduction in pump flow rate.

The two other pressure ratings used are the maximum allowable working pressure, which is the maximum gage pressure at the outlet port permitted for safe operation. It is usually determined by the stiffness and strength of the pump body and the type of seals used. The maximum differential pressure is the maximum allowable difference between the outlet pressure and the inlet pressure, measured in the same units, and is determined by the capability of the rotating assembly and its fluid seal contact zones to withstand the pressure difference between the OTO and OTI volumes.

Efficiency and Power Overall efficiency (E_{tot}) for any type of pump is the ratio of the output fluid power P_{out} to the input shaft power P_s. In a development similar to that for centrifugal pumps in Section 2.1.1, E_{tot} can then be broken into three main component efficiencies; namely, volumetric, hydraulic, and mechanical. Volumetric efficiency (E_{vol}) is the ratio of ingested flow rate Q_{net} to the theoretical or displacement flow rate Q_d, where $Q_{net} = Q_d - S$, where S is the internal leakage flow rate (slip) through clearances back to the inlet. Thus the volumetric efficiency is defined in the same way as it is for screw pumps (Section 3.7). Hydraulic efficiency (E_{hyd}) is the ratio of the total fluid power P_{tf} of both the net flow and the slip flow to the actual power P_{fl} delivered to the fluid, where $P_{tf} = P_{out}/E_{vol}$. P_{fl} in turn is the sum of P_{tf} and the additional power required to overcome internal hydrodynamic friction P_{fric}. Mechanical efficiency (E_{mech}) is the ratio of the P_{fl} to P_s, where the shaft power $P_s = P_{fl}$ plus the mechanical losses P_{mech}, namely the power due to bearing friction, seal drag, internal rubbing, and so on. For incompressible flow, $P_{out} = [\Delta p \text{ (psi)} \times Q_{net} \text{ (gpm)}/1714]$ hp or $[\Delta p \text{ (bar)} \times Q_{net} \text{ (m}^3\text{/h)}/36]$ kW, where Δp is the pressure rise of the pump, and Q_{net} is also the flow rate leaving the pump.

As in the case of centrifugal pumps, the above definitions yield the following relationship of the component efficiencies with the overall efficiency:

$$E_{tot} = E_{vol} \times E_{hyd} \times E_{mech} \tag{5}$$

For rotary pumps, efficiency evaluation requires a significantly different perspective in comparison to centrifugal pumps. There are several reasons for that. Centrifugal pumps handle primarily low-viscosity liquids, typically under 100 cSt, while rotary pumps operate at much wider range of viscosities, often above 10,000 cSt and at times reaching even to 1,000,000 cSt. The Hydraulic Institute developed a chart (Figure 11 in Section 2.1.1) to correct centrifugal pump flow, head and power, when viscosities exceed normally very low, water-like values, but such cases are very rare, and thus a need to apply those corrections is rare. Also, centrifugal pumps can reach much higher power levels, as compared to rotary pumps, and thus overall efficiency (E_{tot}) evaluation, as it impacts energy consumption, is more important for centrifugal pumps. In contrast, rotary pumps are typically smaller (lower power), and their ability to deliver a constant (or nearly constant) flow rate, with little dependence on system restriction (relative insensitivity of flow to system pressure), is a consideration in the selection and application of rotary pumps. For these reasons, volumetric efficiency (E_{vol}) has historically played an important role in the world of rotary pumps, while overall (E_{tot}) efficiency is of primary importance in the world of centrifugal pumps.

The above efficiency definitions for rotary pumps are stated as follows in equation form. Volumetric efficiency is given by

$$E_{vol} = Q_{net}/Q_d = (Q_d - S)/Q_d \tag{6}$$

where

$$Q_d = \text{theoretical (ideal) flow, in the absence of slip, } = q_0 \times \text{rpm} \tag{7}$$

and where Q_{net} = net flow rate leaving a rotary pump

S = slip (internal recirculation back to inlet)

q_0 = unit flow rate (constant for a given pump) per revolution

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Overall efficiency is formally stated as

$$E_{tot} = P_{out}/P_s \tag{8}$$

where, assuming incompressible flow, useful fluid power is given by

$$P_{out} = [Q_{net}\,(\text{gpm}) \times \Delta p\,(\text{psi})/1714]\ \text{hp} = Q_{net}\,(\text{m}^3/\text{h}) \times \Delta p\,(\text{bar})/36\ \text{kW} \tag{9}$$

and total fluid power (useful plus that which drives the slip) is

$$P_{tf} = [Q_d \times \Delta p/1714]\ \text{hp} = [Q_d \times \Delta p/36]\ \text{kW} \tag{10}$$

The actual power P_{fl} delivered to the fluid is P_{tf} augmented by the internal hydrodynamic friction P_{frict}:

$$P_{fl} = P_{tf} + P_{frict} \tag{11}$$

and the input shaft power delivered to the pump is the total fluid power plus the mechanical losses:

$$P_s = P_{fl} + P_{mech} \tag{12}$$

Slip S and the viscous drag, P_{visc}, where,

$$P_{visc} = P_{frict} + P_{mech} \tag{13}$$

are determined (usually by testing) by pump manufacturers. These are specific to each design type and size and are published in pump catalogues as functions of pump speed, differential pressure, and viscosity.

Volumetric efficiency has numerical significance only when comparing similar designs (their comparative/competitive ability to maintain a constant flow rate with minimum slip), operating at the same speed. The reason for this is that, for rotary pumps, the ideal, theoretical, or displacement flow rate Q_d is directly proportional to pump speed, but slip does not depend on pump speed, and only depends on differential pressure, viscosity, and internal clearances. Consider, for example, a rotary pump delivering a required flow rate of 90 gpm (20.4 m³/h) at 1000 rpm, with 10 gpm (2.3 m³/h) internal slip (the ideal flow rate thus being 100 gpm or 22.7 m³/h). The volumetric efficiency of this pump is 90/100 = 90%. If this pump speeds up to 2000 rpm, its ideal flow will double (200 gpm or 45.4 m³/h), but the slip remains the same 10 gpm (2.3 m³/h), with a net flow rate of 200 − 10 = 190 gpm (43.2 m³/h), thereby yielding higher volumetric efficiency 190/200 = 95%, but too great a flow rate as compared to the required amount.

If a competitive pump were presented for evaluation, operating at the same speed (1000 rpm), same unit flow (q_0) but with smaller clearances and a slip of only 5 gpm (1.1 m³/h), then its volumetric efficiency would be (100 − 5)/100 = 95%, which makes it a more efficient design with regard to volumetric capability.

However, if the width of the gears of the 2000-rpm pump were reduced by half, its ideal flow would be reduced to 100 gpm (22.7 m³/h) with no change in slip, because end and radial clearances remain the same. Volumetric efficiency would then become (100 − 10)/100 = 90%. This altered 2000-rpm pump now has lower volumetric efficiency (a greater proportion of slip), but it is also less expensive due to the halved gear length. These facts would need to be evaluated together by the end user.

NUMERICAL EXAMPLE

The trends of the above example and the distinctions discussed between rotary pumps (gear, vane, or lobe) and centrifugal pumps become clearer in the following series of performance computations pertaining to a single rotary pump operating at two greatly differing viscosities (Cases 1 and 2) and at two greatly differing speeds at each of these two viscosities (subcases of Cases 1 and 2). Most rotary pump manufacturers publish pump performance curves showing four main relationships that enable these computations to proceed. Specifically, Figures 15 through 18 illustrate these relationships for the pump of this example.

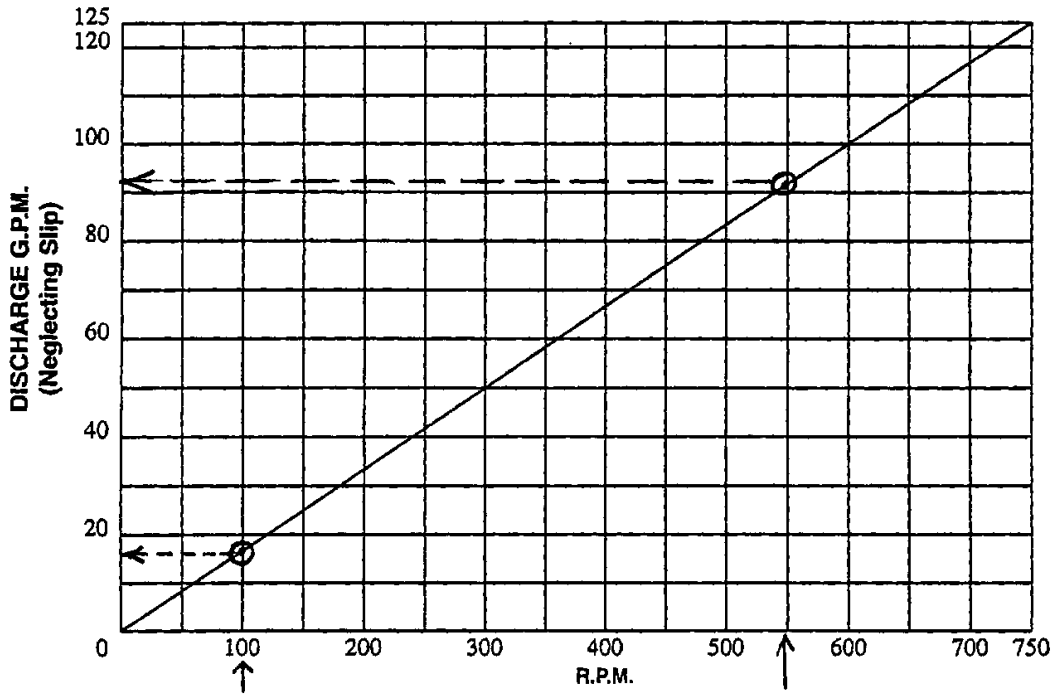


FIGURE 15 Ideal (theoretical) flow rate versus speed. This is typically a straight line, with a slope equal to pump unit flow q_0. ($m^3/h = 0.227 \times$ gpm) (Ranger, Inc.)

Figures 15 and 17 are plotted for convenience, because they can be alternatively calculated from Eqs. 7 and 10; but Figures 16 and 18 can be obtained only from the manufacturer's curves (internally tested values).

Case 1: Viscosity = 30 SSU (~ 1 cSt)

Subcase 1.1: Speed = 100 rpm. Displacement flow rate: $Q_d = q_0 \times 100 = 0.17 \times 100 =$ 17.0 gpm (3.86 m^3/h) (or read from Figure 15).

Results for Δp = 25 psi (1.72 bar)

Slip flow rate, S = 4.2 gpm (0.95 m^3/h) from Figure 16

Net flow rate, Q_{net} = 17.0 − 4.2 = 12.8 gpm (2.91 m^3/h)

Total fluid power $P_{tf} = \Delta p \times Q_d/1714 = 25 \times 17.0/1714 = 0.25$ hp (= 0.185 kW) (or read from Figure 17)

Useful output power, $P_{out} = \Delta p \times Q_{net}/1714 = 25 \times 12.8/1714 = 0.187$ hp (0.139 kW)

Viscous power, $P_{visc} \approx 0.1$ hp (0.075 kW) from Figure 18

Shaft power, $P_s = P_{tf} + P_{visc} = 0.25 + 0.1 = 0.35$ hp (0.26 kW)

Volumetric efficiency, $E_{vol} = Q_{net}/Q_d = 12.8/17.0 = 75.2\%$

Overall efficiency $E_{tot} = P_{out}/P_s = 0.187/0.35 = 53.4\%$

Results for Δp = 50 psi (3.45 bar)

S = 8.4 gpm (1.91 m^3/h)

$Q_{net} = 17.0 - 8.4 = 8.6$ gpm (1.95 m^3/h)

$P_{tf} = \Delta p \times Q_d/1714 = 50 \times 17.0/1714 = 0.496$ hp (0.370 kW)

$P_{out} = \Delta p \times Q_{net}/1714 = 50 \times 8.6/1714 = 0.251$ hp (0.187 kW)

$P_{visc} \approx 0.1$ hp (0.075 kW)

$P_s = P_{tf} + P_{visc} = 0.496 + 0.1 = 0.596$ hp (0.445 kW)

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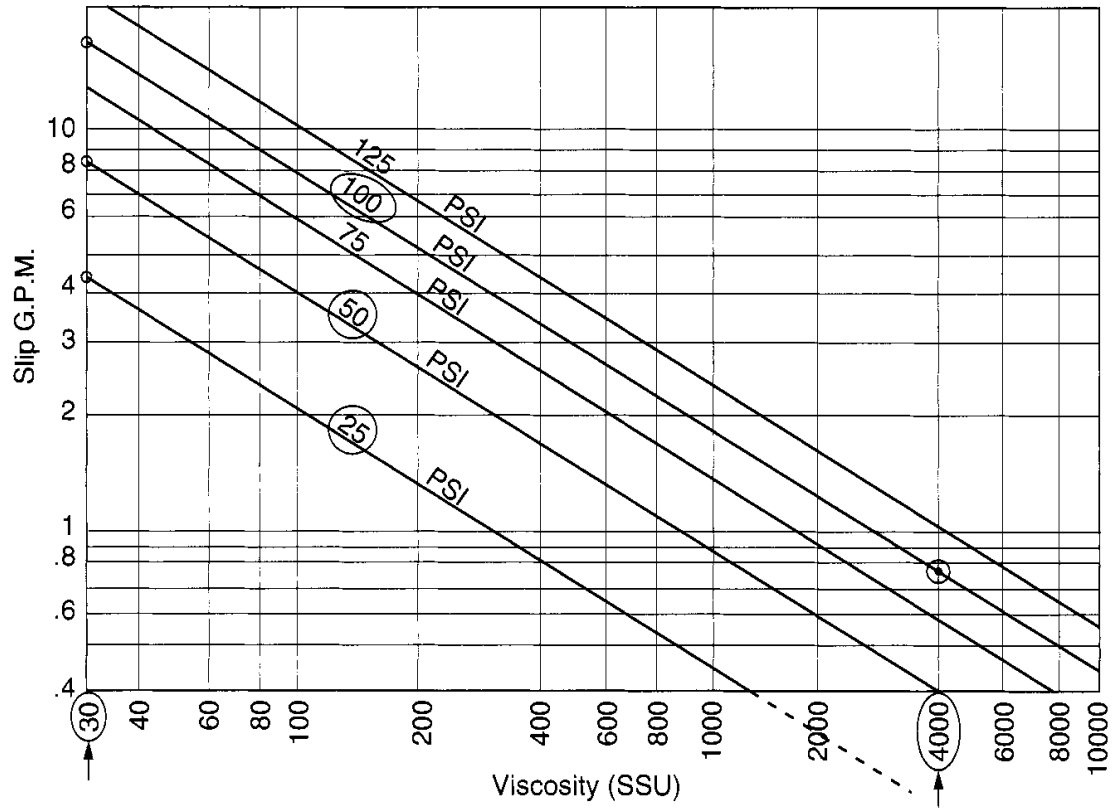


FIGURE 16 Relationship between internal slip and viscosity at several values of differential pressure. (m^3/h = 0.227 × gpm and bar = psi/14.5) (Ranger, Inc.)

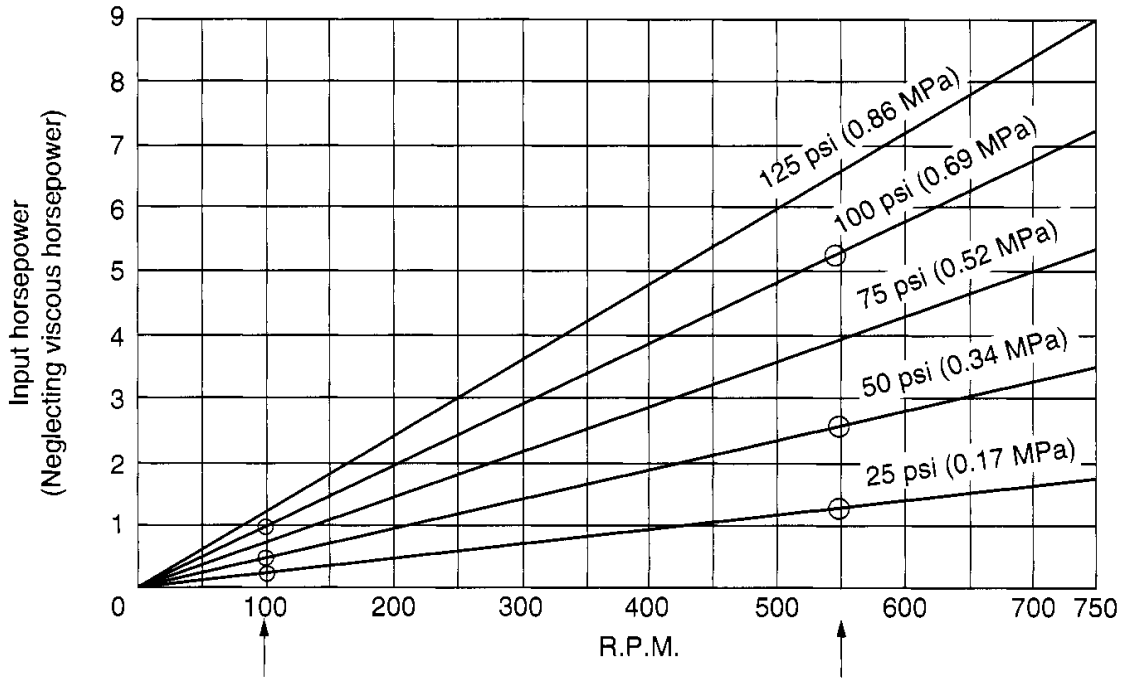


FIGURE 17 Total fluid power (minus power due to internal hydrodynamic friction or "viscous power") versus speed at several values of differential pressure (kW = hp/1.34, and bar = psi/14.5) (Ranger, Inc.)

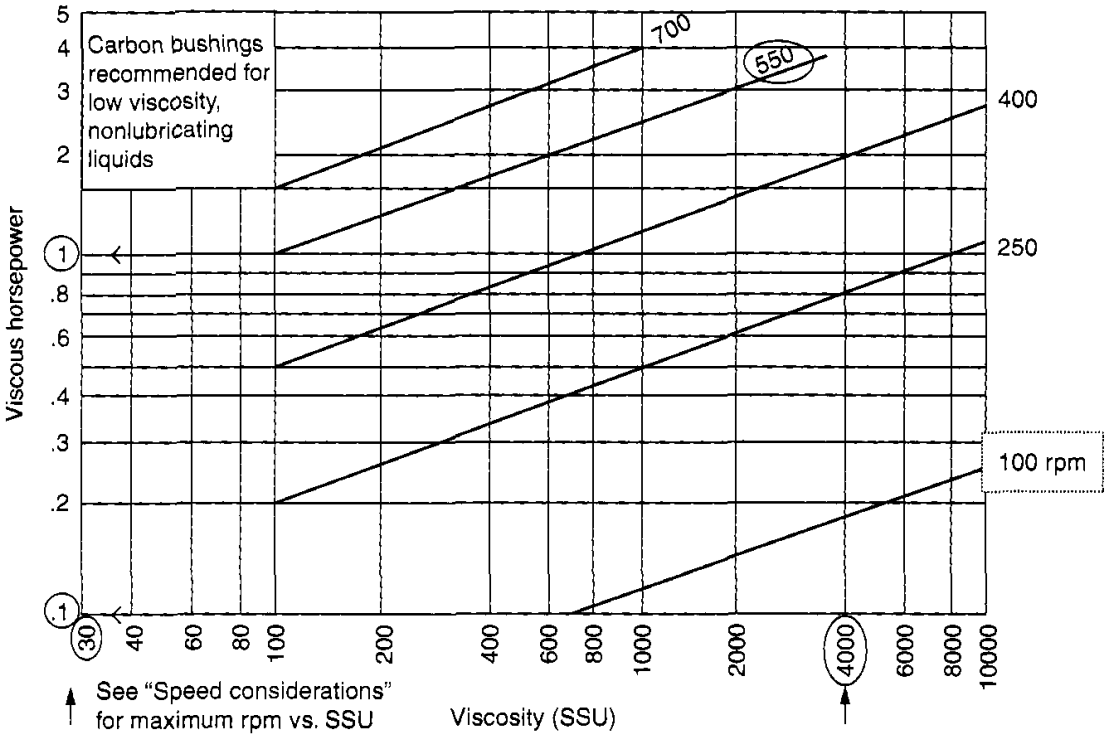


FIGURE 18 Viscous power loss versus viscosity, at several values of speed (kW = hp/1.34) (Ranger, Inc.)

$E_{vol} = Q_{net}/Q_d = 8.6/17.0 = 50.6\%$

$E_{tot} = P_{out}/P_s = 0.251/0.6 = 41.8\%$

Results for $\Delta p = 100$ psi (6.89 bar)

$S = 16.8$ gpm (3.82 m³/h)

$Q_{net} = 17.0 - 16.8 = 0.2$ gpm (0.045 m³/h)

$P_{tf} = \Delta p \times Q_d/1714 = 100 \times 17.0/1714 = 0.992$ hp (0.740 kW)

$P_{out} = \Delta p \times Q_{net}/1714 = 100 \times 0.2/1714 = 0.012$ hp (0.009 kW)

$P_{visc} \approx 0.1$ hp (0.075 kW)

$P_s = P_{tf} + P_{frict} = 0.992 + 0.1 = 1.092$ hp (0.815 kW)

$E_{vol} = Q_{net}/Q_d = 0.2/17.0 = 1.2\%$

$E_{tot} = P_{out}/P_s = 0.012/1.092 = 1.1\%$

Observation: The slip S appears to vary linearly with differential pressure. This is not always the case for all gear pumps and is strictly coincidental in this particular example.

Subcase 1.2: Speed = 550 rpm. Displacement flow rate: $Q_d = q_0 \times 550 = 0.17 \times 550 = 93.5$ gpm (21.24 m³/h) (or read from Figure 15).

Results for $\Delta p = 25$ psi (1.72 bar)

$S = 4.2$ gpm (0.95 m³/h)

$Q_{net} = 93.5 - 4.2 = 89.3$ gpm (20.28 m³/h)

$P_{tf} = \Delta p \times Q_d/1714 = 25 \times 93.5/1714 = 1.364$ hp (1.017 kW)

$P_{out} = \Delta p \times Q_{net}/1714 = 25 \times 89.3/1714 = 1.303$ hp (0.972 kW)

P_{visc} = 1.0 hp (0.746 kW)
$P_s = P_{tf} + P_{frict}$ = 1.364 + 1.0 = 2.364 hp (1.764 kW)
$E_{vol} = Q_{net}/Q_d$ = 89.3/93.5 = 95.5%
$E_{tot} = P_{out}/P_s$ = 1.303/2.364 = 55.1%

Results for Δp = 50 psi (3.45 bar)
S = 8.4 gpm (1.91 m³/h)
Q_{net} = 93.5 − 8.4 = 85.1 gpm (19.33 m³/h)
$P_{tf} = \Delta p \times Q_d/1714$ = 50 × 93.5/1714 = 2.728 hp (2.035 kW)
$P_{out} = \Delta p \times Q_{net}/1714$ = 50 × 85.1/1714 = 2.482 hp (1.852 kW)
P_{visc} = 1.0 hp (0.746 kW)
$P_s = P_{tf} + P_{frict}$ = 2.728 + 1.0 = 3.728 hp (2.781 kW)
$E_{vol} = Q_{net}/Q_d$ = 85.1/93.5 = 91.0%
$E_{tot} = P_{out}/P_s$ = 2.482/3.728 = 66.6%

Results for Δp = 100 psi (6.89 bar)
S = 16.8 gpm (3.82 m³/h)
Q_{net} = 93.5 − 16.8 = 76.7 gpm (17.42 m³/h)
$P_{tf} = \Delta p \times Q_d/1714$ = 100 × 93.5/1714 = 5.455 hp (4.069 kW)
$P_{out} = \Delta p \times Q_{net}/1714$ = 100 × 76.7/1714 = 4.475 hp (3.338 kW)
P_{visc} = 1.0 hp (0.746 kW)
$P_s = P_{tf} + P_{frict}$ = 5.455 + 1.0 = 6.455 hp (4.815 kW)
$E_{vol} = Q_{net}/Q_d$ = 76.7/93.5 = 82.0%
$E_{tot} = P_{out}/P_s$ = 4.475/6.455 = 69.3%

Case 2: Viscosity = 4000 SSU (~ 900 cSt)

Subcase 2.1: Speed 100 rpm. Displacement flow rate: $Q_d = q_0 \times 100$ = 0.17 × 100 = 17.0 gpm (3.86 m³/h) (or read from Figure 15).

Results for Δp = 25 psi (1.72 bar)

Slip flow rate, $S \approx$ 0.1 gpm (0.023 m³/h) from Figure 16

Net flow rate, Q_{net} = 17.0 − 0.1 = 16.9 gpm (3.84 m³/h)

Total fluid power $P_{tf} = \Delta p \times Q_d/1714$ = 25 × 17.0/1714 = 0.25 hp (= 0.185 kW) (or read from Figure 17)

Useful output power, $P_{out} = \Delta p \times Q_{net}/1714$ = 25 × 16.9/1714 = 0.247 hp (0.184 kW)

Viscous power, P_{visc} = 0.18 hp (0.134 kW) from Figure 18

Shaft power, $P_s = P_{tf} + P_{frict}$ = 0.25 + 0.18 = 0.43 hp (0.32 kW)

Volumetric efficiency, $E_{vol} = Q_{net}/Q_d$ = 16.9/17.0 = 99.4%

Overall efficiency $E_{tot} = P_{out}/P_s$ = 0.247/0.43 = 57.4%

Results for Δp = 50 psi (3.45 bar)
S = 0.4 gpm (0.091 m³/h)
Q_{net} = 17.0 − 0.4 = 16.6 gpm (3.77 m³/h)
$P_{tf} = \Delta p \times Q_d/1714$ = 50 × 17.0/1714 = 0.496 hp (0.370 kW)
$P_{out} = \Delta p \times Q_{net}/1714$ = 50 × 16.6/1714 = 0.484 hp (0.361 kW)
P_{visc} = 0.18 hp (0.134 kW)

$P_s = P_{tf} + P_{frict} = 0.496 + 0.18 = 0.676$ hp (0.504 kW)
$E_{vol} = Q_{net}/Q_d = 16.6/17.0 = 97.6\%$
$E_{tot} = P_{out}/P_s = 0.484/0.676 = 71.6\%$

Results for $\Delta p = 100$ psi (6.89 bar)
$S = 0.75$ gpm (0.170 m³/h)
$Q_{net} = 17.0 - 0.75 = 16.25$ gpm (3.69 m³/h)
$P_{tf} = \Delta p \times Q_d/1714 = 100 \times 17.0/1714 = 0.992$ hp (0.740 kW)
$P_{out} = \Delta p \times Q_{net}/1714 = 100 \times 16.2/1714 = 0.945$ hp (0.705 kW)
$P_{visc} = 0.18$ hp (0.134 kW)
$P_s = P_{tf} + P_{frict} = 0.992 + 0.18 = 1.172$ hp (0.874 kW)
$E_{vol} = Q_{net}/Q_d = 16.25/17.0 = 95.6\%$
$E_{tot} = P_{out}/P_s = 0.945/1.172 = 80.6\%$

Subcase 2.2: Speed = 550 rpm. Displacement flow rate: $Q_d = q_0 \times 550 = 0.17 \times 550 = 93.5$ gpm (21.24 m³/h) (or read from Figure 15).

Results for $\Delta p = 25$ psi (1.72 bar)
$S \approx 0.1$ gpm (0.023 m³/h)
$Q_{net} = 93.5 - 0.1 = 93.4$ gpm (21.21 m³/h)
$P_{tf} = \Delta p \times Q_d/1714 = 25 \times 93.5/1714 = 1.364$ hp (1.017 kW)
$P_{out} = \Delta p \times Q_{net}/1714 = 25 \times 93.4/1714 = 1.362$ hp (1.016 kW)
$P_{visc} = 3.9$ hp (2.909 kW)
$P_s = P_{tf} + P_{frict} = 1.364 + 3.9 = 5.264$ hp (3.927 kW)
$E_{vol} = Q_{net}/Q_d = 93.4/93.5 = 99.9\%$
$E_{tot} = P_{out}/P_s = 1.362/5.264 = 25.9\%$

Results for $\Delta p = 50$ psi (3.45 bar)
$S = 0.4$ gpm (0.091 m³/h)
$Q_{net} = 93.5 - 0.4 = 93.1$ gpm (21.14 m³/h)
$P_{tf} = \Delta p \times Q_d/1714 = 50 \times 93.5/1714 = 2.728$ hp (2.035 kW)
$P_{out} = \Delta p \times Q_{net}/1714 = 50 \times 93.1/1714 = 2.716$ hp (2.026 kW)
$P_{visc} = 3.9$ hp (2.909 kW)
$P_s = P_{tf} + P_{frict} = 2.728 + 3.9 = 6.628$ hp (4.944 kW)
$E_{vol} = Q_{net}/Q_d = 93.1/93.5 = 99.6\%$
$E_{tot} = P_{out}/P_s = 2.716/6.628 = 41.0\%$

Results for $\Delta p = 100$ psi (6.89 bar)
$S = 0.75$ gpm (0.170 m³/h)
$Q_{net} = 93.5 - 0.75 = 92.8$ gpm (21.08 m³/h)
$P_{tf} = \Delta p \times Q_d/1714 = 100 \times 93.5/1714 = 5.455$ hp (4.069 kW)
$P_{out} = \Delta p \times Q_{net}/1714 = 100 \times 92.8/1714 = 5.414$ hp (4.039 kW)
$P_{visc} = 3.9$ hp (2.909 kW)
$P_s = P_{tf} + P_{frict} = 5.455 + 3.9 = 9.355$ hp (6.979 kW)
$E_{vol} = Q_{net}/Q_d = 92.8/93.5 = 99.3\%$
$E_{tot} = P_{out}/P_s = 5.414/9.355 = 57.9\%$

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SUMMARY

1. Results

Speed		100 rpm			550 rpm	
Δp	25 psi	50 psi	100 psi	25 psi	50 psi	100 psi
			Viscosity = 30 SSU			
Eff_{vol}	75.2%	50.6%	1.2%	95.5%	91.0%	82.0%
Eff_{tot}	53.4%	41.8%	1.1%	55.1%	66.6%	69.3%
			Viscosity = 4000 SSU			
Eff_{vol}	99.4%	97.6%	95.6%	99.9%	99.6%	99.3%
Eff_{tot}	57.4%	71.6%	80.6%	25.9%	41.0%	55.9%

2. Observations
   - *Volumetric* efficiency is very high at high viscosity due to low slip, regardless of pressure and speed.
   - At low viscosity and high pressure *total* efficiency tends to be very low.
   - Higher speed helps *total* efficiency (due to low drag) at low viscosity but reduces it at higher viscosity due to high viscous drag.

Pump Performance Figures 19, 20, and 21 show the change in displacement capacity Q_d, flow rate or capacity Q_c, and slip Q_s as the differential pressure across the pump P_d (= Δp), the viscosity v, and the pump speed N are varied. Several assumptions are made in these graphs. It is assumed that inlet conditions are satisfactory and that there is no inlet effect on the pump capacity over the charted range. It is assumed that the fluid is Newtonian and the liquid is incompressible. In Figures 19 and 20, it is assumed that viscosity is constant and relatively low, and that the speed is within the normal range of the pump. In Figure 20, it is assumed that both pressure and speed are within the normal ratings of the pump.

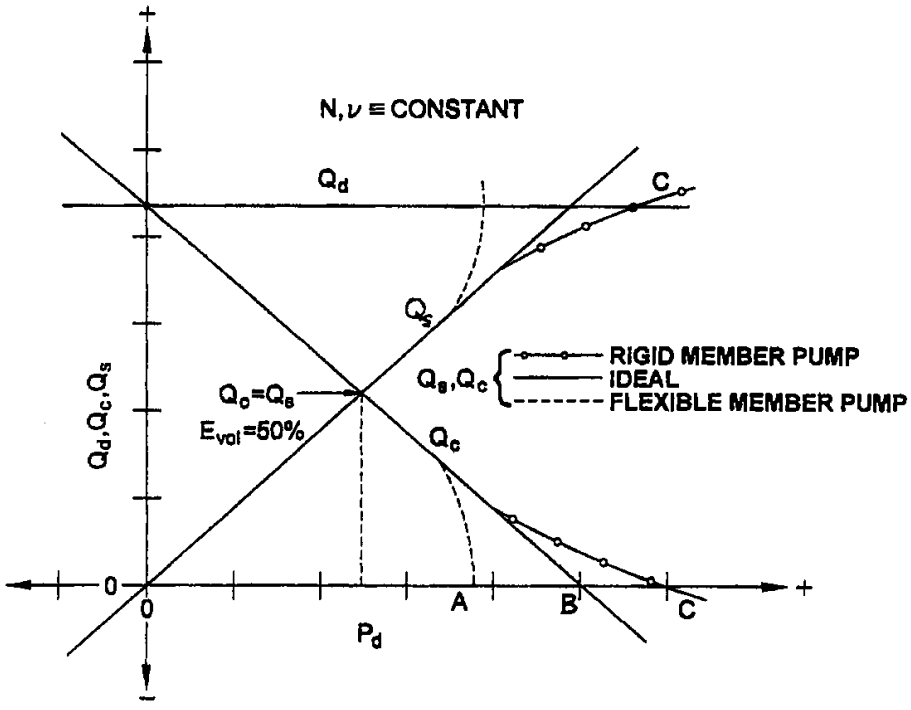


FIGURE 19 Varation of Q_d, Q_c, and Q_s with P_d, N and v constant

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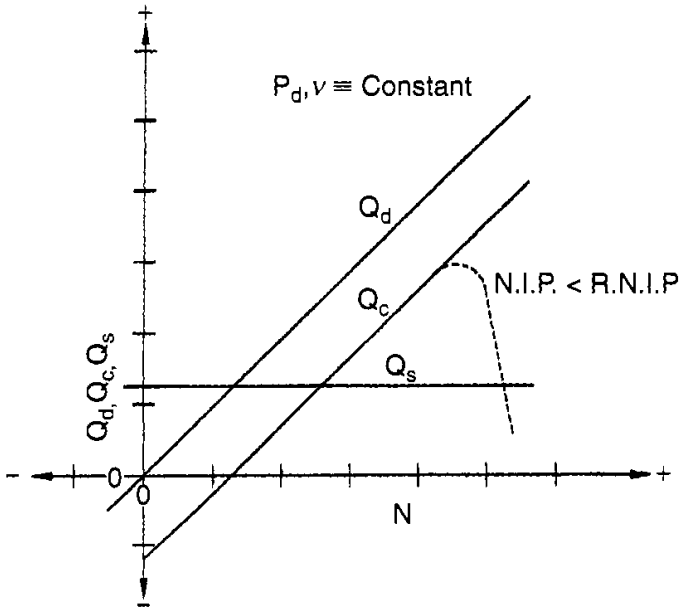


FIGURE 20 Varations of Q_d, Q_c, and Q_s with N, P_d, and υ constant

The graph in Figure 19 is plotted with the flow rate, slip, and displacement flow rate on a linear scale as the ordinate (Y-axis) and pressure on a linear scale as the abscissa (X-axis). A further assumption is that the clearances and viscosity are such that the slip increases proportionately with pressure. The solid lines are the ideal characteristics when secondary effects are neglected. It may be noted that at zero pressure, slip Q_s is zero and Q_c equals Q_d. As the pressure increases, Q_s increases until it equals Q_d at pressure B. If the

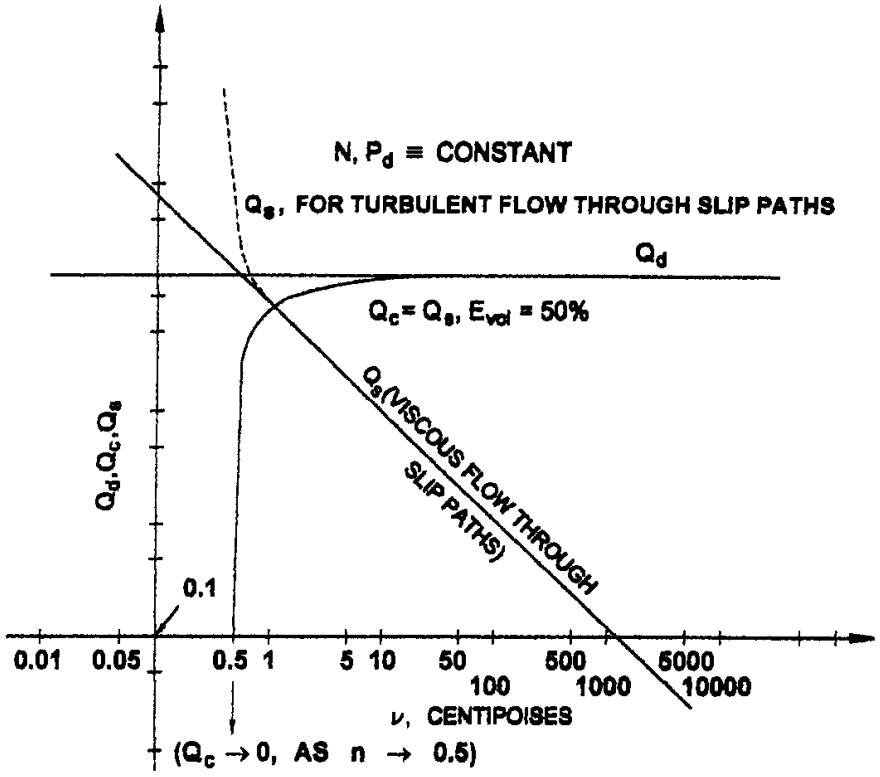


FIGURE 21 Varations of Q_d, Q_o, and Q_s with υ, N and P_d constant

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pressure imposed on the pump were to increase past this point, Q_s would exceed Q_d and the flow through the pump would be from outlet to inlet, causing a negative Q_c.

Although rotary pumps do not operate in this end of the pressure range, the condition of pressure B can be reached when a valve is closed, blocking the outlet of the pump. Pressure B, then, represents dead-ended pressure developed by a rotary pump when its outlet line is blocked. Should Q_d be numerically equal to 100 in whichever units are used, then the plot of Q_c is numerically equal at each point to the volumetric efficiency E_{vol} of the pump and Q_c becomes zero at pressure B.

Pressure B data is usually not available from pump manufacturers since these values are usually far beyond the normal safe pressure rating of the pump. However, they can be estimated by extrapolating the data given. Cautions should be observed, however. A flexible member can quickly reach a pressure at which flexible member deflection becomes excessive, and the pressure at which Q_c equals zero is very quickly reached. This is illustrated in Figure 19 by the dashed line breaking away from the solid Q line and intersecting the abscissa at pressure point A. This characteristic of flexible member pumps provides a self-limited maximum pressure should the pump be dead-ended. In rigid rotor pumps, as the pressure increases beyond the normal operating range, deflections of the shafts and rotors bring the rotors into heavy bearing contact with the body chamber walls, reducing the dimensions of the slip clearance path. The dashed line leaving the Q_c and Q_s curves and intersecting the abscissa and Q_d line at pressure point C illustrates this. Consequently, the zero flow pressure, which is ideally at point B, may be considerably different from this extrapolated value, and it should be measured if it is important to the application.

Under the same assumptions given for Figure 19, Figure 20 shows the relative independence of the slip with speed when differential pressure is constant. Here the chart of Q intersects the abscissa at the point where the speed is low enough for the displacement flow rate to be equal to the slip at the pressure of operation. The speed at which Q_d equals $2 \times Q_s$ is the speed at which Q_s equals Q_c and the volumetric efficiency E_{vol} becomes 50%.

As the speed increases, the pumping action of the shear stress in the clearances tends to reduce the slip below the ideal line. However, for most rotary pumps, the detrimental effects of increasing the speed above the normal operating range is usually caused by inlet losses in the pump. This effect is cavitation and is illustrated by the dashed line, for which the net inlet pressure (N. I. P.) or NPSH is less than that required.

Figure 21 shows the effect of viscosity on the slip and flow rate in a rotary pump. The graphs are on a log-log scale. In this chart, it is assumed that the pressure, speed, and viscosity combine to keep the flow through clearances of the pump in the laminar or "viscous" flow region.

As viscosity increases, slip Q_s becomes very small and the flow rate of the pump Q_c approaches the displacement flow rate Q_d. As the viscosity decreases, the slip very rapidly approaches the displacement flow rate and the flow rate of the pump drops rapidly to zero or to a negative quantity. For any given pump with speed N and differential pressure P_d, there is a viscosity below which the slip flow, through clearances of the pump, will change to a turbulent flow. It is unlikely that this change will occur simultaneously through all slip paths. However, once it begins to occur, the slip will increase much more rapidly with further reductions in viscosity because of the turbulent flow relationship of the slip, pressure, and viscosity expressed in Eq. 14:

$$Q_s = \frac{KP_d^{1/2}}{v^{1/x}} \tag{14}$$

where x usually is in the range of 4 to 10, and K is a constant.

Volumetric efficiency E_{vol} drops very rapidly with the viscosity if the viscosity is lower than that required for 50% volumetric efficiency (represented by the crossover point of the slip and capacity curves). This crossover point occurs for most rotary pumps operating at rated differential pressure P_d for viscosities between 0.1 and 10 centipoise. For the majority of commercially available models, this point usually falls in the viscosity range of 0.3 to 3.0 centipoise at the maximum rated differential pressure.

The effect of inlet pressure on the capacity can be seen clearly if all secondary effects are eliminated. This assumes the pump is operating within its normal pressure limits and speed

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range and that viscosity is high enough to reduce the slip to a negligible value. A graph of the capacity as a function of inlet pressure is shown in Figure 22 with these assumptions.

There is no change in capacity or flow rate as inlet pressure is lowered until the pressure reaches pressure *A* on the graph. If the inlet pressure were lowered further, the flow rate would drop as shown. The cause of this drop is complex in detail but simple in concept. The liquid flow from the inlet port through the inlet chamber of the pump causes a pressure drop, which causes a minimum pressure point somewhere in the inlet chamber. When the pressure in the liquid at this minimum pressure point approaches the vapor pressure of the liquid, vapor begins to form there. When the amount and time persistence of this vapor cause the vapor to be swept into the CTIO volume of the pump, the amount of liquid in this volume is reduced, leaving a deficiency of liquid volume. For example, if half the fluid volume swept from the inlet chamber is vapor, then only half the normal-capacity liquid volume is available at the outlet chamber, and the flow rate is reduced accordingly.

An increase in speed would mean an increase in flow rate, all other things being equal. This would increase the pressure drop between the inlet port and the inlet chamber and correspondingly increase the absolute inlet pressure (which is measured at the inlet port) at which the flow rate would begin to drop (pressure *A*). Correspondingly, if the speed and flow rate were constant and the viscosity increased, the pressure drop between the inlet port and the minimum pressure point in the inlet chamber would increase with viscosity. This also would cause pressure *A* to move to higher inlet absolute pressures. Operation with absolute inlet pressures below pressure *A* for any given speed and viscosity is usually unsatisfactory both because of the drop in capacity (and hence in volumetric efficiency) and because of the noisy and rough operation caused by the formation and collapse of the vapor.

For lower viscosity liquids where the collapse of the vapor bubbles may be quite rapid, cavitation may cause a significant amount of damage to the body or rotor surfaces. It is important to understand that cavitation damage may occur even though the absolute inlet pressure is above pressure *A*. Locations may be found in the inlet chamber, particularly where the flow direction changes rapidly, as around sharp corners, where vapor formation in the form of very fine bubbles, and hence cavitation, may occur. However, these vapor cavities may be swept

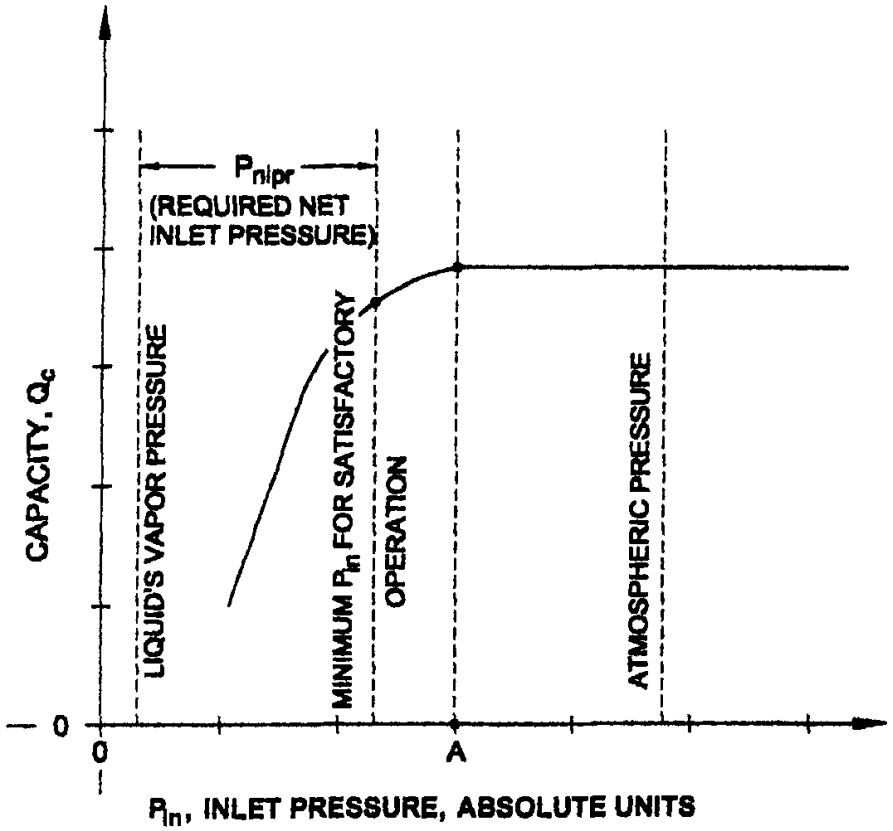


FIGURE 22 Varations of Q_c and P_{in} with other operating conditions constant

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into higher pressure regions of the inlet chamber and collapse near body or rotor surfaces to cause cavitation damage, although the capacity of the pump is not affected. For any given viscosity then, there is an upper limit to the speed at which the pump can be operated.

The inlet pressure may be allowed to drop slightly below pressure *A* without a significant deterioration in pump performance. However, there is a point at which the pressure becomes too low for satisfactory pump operation. This pressure determines the required net inlet pressure P_{nipr} or NPSHR for the particular pump and the particular set of operation conditions. For any given set of operating conditions, satisfying the required net inlet pressure or NPSHR is a main limitation on the pump operating speed.

The other main limit on the pump operating speed is the pump outlet pressure. In every application, some fluid frictional losses occur in the outlet system of the pump. Even if the pump outlet is opened to the atmosphere, a pressure drop exists between some maximum pressure point in the pump outlet chamber and the pump outlet itself. However, by far, the most common situation is one in which a significant fluid friction pressure is developed in the system external to the outlet of the pump. This fluid pressure usually is a function of pump capacity and therefore of pump speed. If the inlet conditions are maintained to keep the inlet pressure above pressure *A* as the pump speed increases, a speed will be reached at which the pump outlet pressure equals the outlet pressure rating of the pump. Operating at speeds higher than this will cause the pump outlet pressure to exceed the rated pressure and this could result in damage to the pump.

In most applications, one of the two limits described determines the maximum permitted speed of pump operation. However, if neither of these conditions limits the speed and the speed is continuously increased, a speed will be reached at which the peripheral velocity of the rotors will exceed the cavitation velocity of the liquid. A further increase in speed beyond this point would be limited by the cavitation occurring at the rotor outer radial surfaces.

SPECIFICATIONS, INDUSTRIES, AND APPLICATIONS

In addition to the many application-oriented classifications for vane, gear, or lobe pumps, many technical classifications are also issued by various industrial, governmental, or even military organizations. It is also important for both the manufacturer and the user to know of any voluntary or regulatory specifications governing the construction and materials of the pumps destined for use in a specific industry and application.

Many industry-based specifications, such as the International Association of Milk, Food, and Environmental Sanitarians, the United States Public Health Service, and the Dairy Industry Committee, have jointly issued specifications called "The 3-A Sanitary Standards for Pumps for Milk and Milk Products." These standards govern the materials of construction of the pump, the surface finish and shape details of the contact surfaces, the finish and shape of external surfaces, the method of mounting, and restrictions on gaskets, seals, and other pump auxiliary features. Pumps constructed to meet 3-A Sanitary Standards are called *sanitary* pumps and carry a 3-A seal of approval mounted on the pump body. Similar specifications have been developed by the International Association of Milk, Food, and Environmental Sanitarians, the United States Public Health Service, the United States Department of Agriculture, the American Poultry Industries, and the Dairy and Food Industries Supply Association to cover design features used in the handling of cracked eggs.

Standards for pumps used in various processes in the petroleum industry are concerned with the materials and design features that are intended to prevent catastrophic failures when explosive, flammable, or toxic fluids are being handled. Others generated by governmental agencies and professional societies, such as the American Society of Mechanical Engineers and the American Petroleum Institute Standard 676, establish manufacturing procedures and design limitations on pumps for refineries, nuclear power stations, and so on. In these cases, the terms *sanitary, aseptic, explosion proof, API pump* or *N stamp* are not merely generic terms and cannot be casually applied. They must instead be carefully applied and used only when the manufacturer warrants that the pumps actually meet these specifications. In addition, as an unavoidable consequence, the cost of manufacturing pumps to meet any of these standards makes them more expensive than pumps built for general service applications.

TABLE 2 Markets and applications

Animal Renderings Industry

General transferring to burners of:
- animal feed
- beef by-products
- bone meal
- molten fats
- poultry by-products

Automobile Industry

Distribution and assembly line supply of:
- grease
- lubricating oil
- motor fuel
- paint, lacquers and thinners
- test stand systems

Chemical Industry

Any process or distribution system handling of:
- acids
- adhesives
- bleaches
- caustics
- cellulose acetate
- glycerin
- nitrates

Cosmetics and Soap Industry

Nonemulsifying supply to packaging machinery of:
- detergent
- lanolin
- liquefied soap
- shampoo

Food Products Industry

General process transferring of:
- coconut oil
- corn oil
- fish oils
- glucose
- glycerin
- lard
- milk and milk by products
- molasses
- palm oil
- sucrose solutions
- vegetable shortening

General Industry and OEMs

Various fluid handling requirements for:
- burners and heater sets
- centrifuge separator systems
- cooking fat filtration units
- filtering and metering units
- fluid reclamation systems
- hydraulic auto lifts and elevators
- lube oil supply units for rotating equipment
- machinery coolant and lubricant circulation
- paint spray booths
- roofing tar systems
- textile dyeing and printing machinery
- wood preservation machinery

Marine Industry

Unloading, tank stripping and engine room supply of:
- asphalt
- heating oil
- heavy fuel oil
- marine diesel oil
- lubricating oil
- molasses

Paint and Lacquer Indus

Various process handling applications for:
- dyes and pigments
- lacquer
- latex
- paint
- titanium dioxide slurry
- thinners and solvents

Petroleum Industry

Blending, packaging, and unloading of:
- asphalt
- bitumen
- creosote
- grease
- fuel oil
- heavy fuel oil
- lubricating oils

Pulp and Paper Industry

Various general mill and coating applications for:
- adhesives
- black liquor
- de-inking solvents
- fuel oil
- printing ink
- sulfate soap
- tall oil
- titanium dioxide slurry
- turpentine
- viscose
- waste pond recycling
- white liquor

Plastics and Petrochemical Industry

Process transfer of:
- cellophane
- epoxy hardener
- film dope
- insulation coatings
- isocyanate
- polyol
- polyester
- rayon

Public Utilities and Power Stations

General transfer and supply of:
- central lubrication systems
- fuel oil supply to burners
- oil pressure for damper controls
- transfer of gashouse tars, creosote, etc.

Steel Industry

Circulation and supply of:
- coolants
- quench oils
- lube oil to rolling mills
- hydraulic oil to hydraulic lifters

Rotary pumps are used for both metering and transfer applications; although, small diaphragm metering pumps are now more common. Metering pumps tend to be smaller and operate with varying flow rates. Transfer pumps typically have a fixed speed, although variable speed units are becoming more common as the cost for this equipment falls.

In general, the three most basic applications for these pumps are as follows:

1. **Liquid handling** In this class, well over 2,000 fluids are pumped and performance is judged on the pump's capability to handle the specific liquids, and the hydraulic power generated by the pump is a secondary consideration.
2. **Hydraulic fluid power** In this class, the hydraulic power generated by the pump is used to actuate valves, pistons, cylinders, rotary actuators, and similar devices. Pumps in this class are usually special vane, gear, piston, or screw types and are designed for pressures up to 9,000 lb/in^2 (620 bar) on selected hydraulic power fluids.

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3. **Commercial and retail** In this class are the pumps designed for a single specific use, application, or even fluid. Pumps in this miscellaneous class include everything from miniature pumps for home aquariums to well water pumps for agricultural use to fuel injection pumps for automobiles.

Among these three application areas, the most common is liquid handling tasks, and they make up the largest percentage of uses for rotary vane, gear, and lobe pumps.

Markets and Applications The markets and applications served by vane, gear, and lobe pumps are among the broadest of all rotary pumps. Although it would not be possible to list every application served by these pumps, a broad sampling of the more common ones is shown in Table 2.

REFERENCES

1. Budris, A. "Preventing Cavitation in Rotary Gear Pumps." *Chemical Engineering*, May 5, 1980, pp. 109–112.

FURTHER READING

Nelik, Lev. *Centrifugal and Rotary Pumps: Fundamentals and Applications*, CRC Press, Boca Raton, FL, 1999.

C • H • A • P • T • E • R • 4

SOLIDS PUMPING

SECTION 4.1
HYDRAULIC TRANSPORT OF SOLIDS

KENNETH C. WILSON
ANDERS SELLGREN

INTRODUCTION

Applications of Slurry Transport Vast tonnages are pumped every year in the form of solid-liquid mixtures, known as slurries. The majority of applications are short-distance in-plant installations, mainly in the minerals industry. A large plant may contain several hundred centrifugal pumps handling various types of slurries.

The application that involves the largest quantities is the dredging industry, continually maintaining navigation in harbors and rivers, altering coastlines and winning material for landfill and construction purposes. As a single dredge may be required to maintain a throughput of 7000 tons of slurry per hour or more, very large centrifugal pumps are used with impeller diameters of over 100 in. (2.5 m). (See Wilson et al., 1997, and Section 4.2.)

The manufacture of fertilizer is another process involving massive slurry-transport operations. In Florida phosphate matrix is recovered by huge draglines in open-pit mining operations. It is then slurried, and pumped to the wash plants through pipelines with a typical length of about 6 miles (10 km). The total drive capacity can be in excess of 13,400 hp (10 MW).

Recent decades have seen a great increase in the transport of waste materials, in slurry form, to suitable deposit sites. Waste-disposal environmental problems require that wastes be conveyed to dedicated and monitored disposal sites, either underground or on the surface. This requirement can often be satisfied by backfilling mines (either deep or open-pit), and slurry transport is the favored placing method.

Large slurry pumping operations are found in Alberta, Canada, where oil sands obtained by open-pit mining operations are directly pumped for processing. The mixing and exposure time during the slurry transportation has eliminated some of the processing carried out at the processing plant, resulting in large savings. After the extraction operations are completed, the sand is used as backfill in areas previously mined.

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Coarse-particle slurries with maximum particle sizes of 4 in (100 mm) or more can often be pumped cost-effectively by combining them with smaller particles, to produce a broad and even particle size distribution (Sellgren & Addie, 1998).

Slurry transportation may play an important role in the development of integrated mining systems of tomorrow. In mines there is sometimes a considerable inflow of groundwater that has to be removed. When the mine dewatering installations are integrated with a hydraulic hoisting system, the cost of power needed to pump out the groundwater can be excluded from the cost of hoisting in making cost comparisons with other modes of transporting the solids to the surface. The economic effectiveness of hydraulic hoisting, together with hydraulic design considerations, have been discussed by Kostuik (1965) and Sellgren et al. (1989) for both small shallow mines and large deep underground mines.

Long-distance slurry pipeline transport is a thoroughly tested and cost-effective mode of transportation of ores, coal, and industrial minerals. For example, in Brazil up to 13 million tons (12 million tonnes) of iron ore concentrate (0.15 mm) have been pumped per year since 1977 from an inland mine to a pellet plant at the coast 250 miles (400 km) away. The well-known Black Mesa pipeline in southwestern United States transports a partially processed slurry coal (1.5 mm) about the same distance to an electrical generating station.

Long distance slurry pipelines involve heads of 75 atmospheres or more at each pumping section, the pumps employed are therefore usually of the positive-displacement type (see Section 4.3). Descriptions of long-distance pipelines and associated slurry pumps can be found, for example, in Brown & Heywood (1991).

Principles of Slurry Flow Slurries are mixtures of solid particles and a liquid (typically water). The interaction of the solids and the liquid can produce a large variety of flow behaviors. The major types will be outlined here and described in more detail in subsequent sections. The simplest type of slurry behavior occurs, for example, when silt or fine sand is dredged. It is represented by the "equivalent fluid" model. This evaluates the friction loss (pressure gradient) in the pipeline on the basis of an equivalent fluid with the density of the mixture and a friction factor near that of water at the same flow rate. This simple case is not particularly common in practice. On one hand, there are slurries of finer particles, for example natural clays or industrial materials like red mud. Typically, these are non-Newtonian in nature, and require rheological techniques to evaluate the pressure gradient. These will be outlined in the following section on homogeneous slurries.

On the other hand, as particle size or density increases, then settling of particles becomes significant. In turbulent flow the fluctuating velocity of the turbulent eddies (v') acts against the particle settling velocity (terminal fall velocity v_t). The importance of particle settling increases as the ratio v_t/v' becomes larger. When settling is significant the slurry is no longer homogeneous. It is particle-lean near the top of the pipe and particle-rich near the bottom. In the lower area, contact between particles and the bottom of the pipe add a granular (Coulombic) shear stress to that produced by the fluid, thus increasing the pressure gradient (for a given flow rate). The solids contributing to this Coulombic stress are known as "contact load" solids. The stratification ratio, representing the fraction of total solids that travels as contact load, tends to increase with v_t/v'. The turbulent fluctuating velocity v' varies with the mean flow velocity V_m (flow rate divided by $\pi D^2/4$ where D is the internal pipe diameter). Thus, decreasing V_m from an initially large value increases the stratification ratio, and hence the solids effect on the pressure gradient. With further reductions in V_m the liquid effect on pressure gradient decreases, combining with the increasing solids effect to produce a minimum in the pressure gradient (for constant delivered solids concentration). This minimum, shown schematically on Figure 1, was observed long ago by Blatch (1906).

Calculations for this type of pressure-gradient behavior are outlined in the next section on settling slurries, as are other features of this type of flow. It is worth noting here that, as V_m is decreased below the minimum-gradient value, there comes a lesser velocity at which particles at the bottom of the pipe cease moving and form a stationary deposit, also shown on Figure 1. This deposit leads to a further increase in pressure gradient (and hence in the power required) and may cause system instability. Beginning with Durand (1951a, 1951b), deposition has been studied extensively (see the settling slurry section), primarily so that it can be predicted and hence avoided in engineering design.

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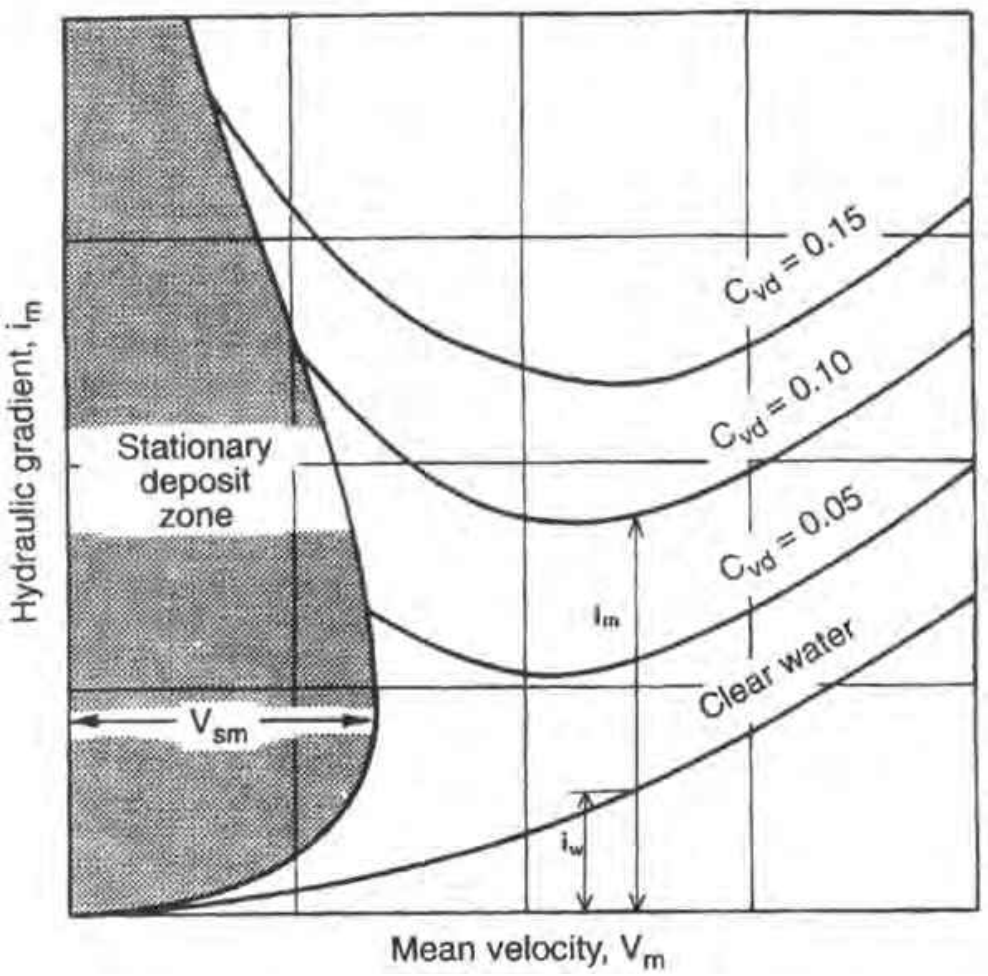


FIGURE 1 Definition sketch

Newitt et al. (1955) divided slurry flows into three types. These include the equivalent-fluid type and the partially stratified type, which have been previously described. The remaining type is known as fully stratified. In this case, either the particle fall velocity is large or the particle size is a significant fraction of the pipe diameter. These factors preclude any support of the solids by the fluid turbulence, so that the submerged weight of all the particles must be carried to the pipe by granular contacts. Calculations (outlined on the section on settling slurries) and experiments show that pressure gradients can be very high for this type of flow. As a result, it is not suitable for long-distance or medium-distance transport; but it has found some applications for short hauls.

HOMOGENEOUS SLURRIES

Equivalent-Fluid Calculations As mentioned in the previous section, pseudo-homogeneous flow (e.g., aqueous slurries of silt or fine sand) shares with truly homogeneous flow the property that the pressure gradient increases with throughput velocity in a fluid-like fashion. An increase of this type can be expressed, at least to a reasonable approximation, by the statement that the pressure drop for turbulent flow of a homogeneous or pseudo-homogeneous mixture is proportional to that obtained for an equal discharge of carrier fluid alone.

Before this statement can be put into mathematical form, it is necessary to consider various ways of expressing the pressure drop due to friction. The change of pressure per

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unit length of pipe, $\Delta p/\Delta x$, is not commonly employed in practice, in part because it includes the effect of differences in elevation as well as friction losses. For liquids, especially water, the hydraulic gradient, i, is normally used. This quantity is the slope of the hydraulic grade line, which is based on the levels to which the fluid would rise in a series of imaginary tubes tapped into the pipeline. As the slope of the hydraulic grade line, the hydraulic gradient represents the drop in level per unit length of pipe. For water (density ρ_w) flowing in a horizontal pipe, i is related to the pressure gradient as follows

$$i = \left(- \frac{\Delta p}{\Delta x} \right) / \rho_w g \tag{1}$$

where g is gravitational acceleration.

As water forms the carrier liquid in the majority of slurry flows, it is convenient to use it as a standard for comparison of frictional pressure losses. Thus, throughout this chapter the usual expression for frictional losses is in terms of i (height of water per unit length of pipe). The value for i for clear-water flow is written i_w, and that for a mixture (i.e., frictional loss in height of water per unit length) is written i_m. The statement made previously for homogeneous and pseudo-homogeneous flows amounts to a direct proportionality between i_m and i_w.

Calculation of i_w uses the Stanton-Moody friction factor, f, in the equation

$$i_w = f \frac{V^2}{2gD} \tag{2}$$

For turbulent flow, the friction factor depends on both the Reynolds number ($Re = \rho V_m D/\mu$) and the relative roughness ϵ/D. These two quantities can be entered on Figure 2, which is

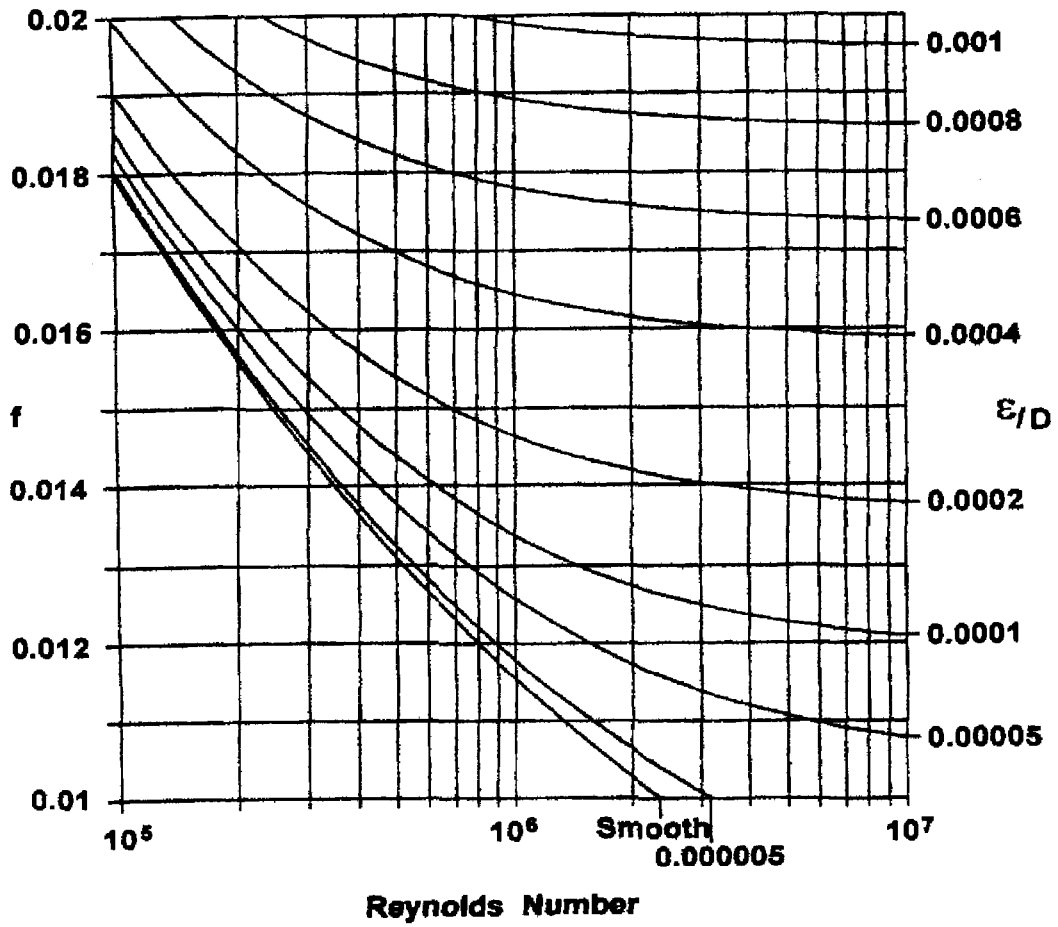


FIGURE 2 Friction factor in normal operating range

used to obtain f. For example, with $Re = 1.5 \times 10^6$ and $\varepsilon/D = 0.0001$, $f = 0.013$ (a typical value). As f is approximately constant for turbulent flow in a given pipe, Eq. 2 shows that the hydraulic gradient varies roughly as V^2 (or as Q^2).

In nonslurry applications, if a liquid other than water is conveyed, the hydraulic gradient concept is applied by substituting the density of this liquid for that of water in Eq. 1, so that the hydraulic gradient is expressed in height of flowing liquid per unit length. In considering, say, a pseudo-homogeneous slurry of fine sand, it may be of interest to compare its behavior to that of a liquid with density equal to that of the flowing mixture ρ_m, which is given by

$$\rho_m = \rho_w(1 + (S_s - 1)C_v) \tag{3}$$

Here S_s is the relative density of the solids (compared to water) and C_v is the volumetric concentration of solids.

Substituting ρ_m for ρ_w in Eq. 1 gives the mixture-height gradient (i.e., measured in height of mixture rather than of water). For clarity, a different symbol must be used, and j is employed for this purpose. For a mixture flowing in a horizontal pipe:

$$j = \left(-\frac{\Delta p}{\Delta x}\right)/\rho_m g \tag{4}$$

Although j is expressed in height of mixture per unit length of pipe, actual measurements based on columns of mixture in vertical tubes would not be feasible, and j would have to be obtained indirectly from pressure measurements. It can be seen from Eqs. 1 and 4 that the ratio of i to j equals the relative density of the mixture S_m (i.e., ρ_m/ρ_w).

The equivalent-fluid model of slurry flow assumes that the solids have little effect on friction factor, and that the mixture acts as a liquid as far as the relative-density effect is concerned. The resulting hydraulic gradient for homogeneous mixture flow, i_{mh}, is equivalent to the product of S_m and i_w. Although the equivalent-fluid model has been widely employed in the past, it is not generally supported by the experimental evidence. For example, sand-water experiments by Carstens & Addie (1981) show that for some pseudo-homogeneous flows i_{mh} does not exceed i_w at all. An appropriate equation for the hydraulic gradient is:

$$i_{mh} = [1 + A'(S_m - 1)]i_w \tag{5}$$

Setting the coefficient A' equal to unity gives the relative-density effect of an equivalent-fluid model, whereas $A' = 0$ gives the behavior observed by Carstens & Addie (1981). Intermediate types can be represented by values of A' between zero and unity. Equation 5 will be referred to as the "homogeneous flow" equation, with the specific case of $A' = 1.0$ called the "equivalent fluid" model.

Modeling Non-Newtonian Flows In pipeline transport of non-Newtonian materials, the variation of pressure drop with velocity is typically rather flat for laminar flow, while it is much steeper for turbulent flow. Figure 3 shows these features. This figure is a plot, on logarithmic axes, of hydraulic gradient j_m (i.e., frictional losses in height of equivalent fluid per unit length of pipe) versus mean velocity V_m (volumetric discharge/cross-sectional area). The data shown on this plot refer to various concentrations of a red mud tested at the GIW Hydraulic Laboratory in a pipeline with internal diameter of 3.19 in. (81 mm). The plotted data are for mixtures with relative densities from 1.14 to 1.28. Note that where laminar flow occurs the slope of the lines (referred to the logarithmic coordinates) is rather small and does not vary significantly with particle concentration. However, the vertical position of the laminar lines increases strongly with increasing concentration. Conversely, for turbulent flow, concentration has very little effect; all the points fall close to a single line with slope (for logarithmic coordinates) close to 2.0. This behavior implies that the friction factor f has a near-constant value for the observed turbulent flows. Note that the transitions between laminar and turbulent flow are rather abrupt.

Rheograms, that is, curves of shear stress versus strain rate, are obtained from tests under laminar conditions, using either rotary viscometers or tube viscometers. The latter

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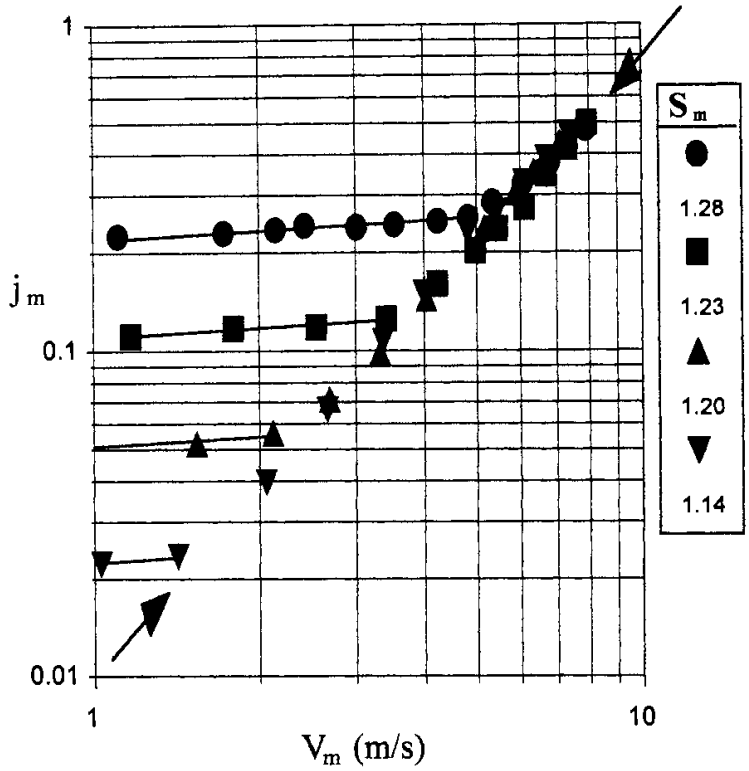


FIGURE 3 Friction gradients of a red-mud slurry

are preferable, as they are geometrically similar to the pipeline configuration. For a Newtonian fluid of viscosity μ, shear stress τ is related to strain rate du/dy by the classic expression

$$\tau = \mu(du/dy) \tag{6}$$

Rheograms for non-Newtonian materials are not so simple. For example, Figure 4 shows a rheogram for a fine-particle slurry. This rheogram does not pass through the origin (the strain rate remains zero until a certain yield stress τ_y is exceeded) and is not straight (although in this case the behavior is approximately linear at large values of strain rate). The definition of viscosity given by Eq. 6 can be retained, although this viscosity no longer represents the slope of the rheogram itself, but rather that of the secant line shown on the figure. Hence μ as defined by Eq. 6 can also be called the "secant" viscosity. The tangent to the rheogram, though less meaningful physically, is often referred to. This "tangent" viscosity will be denoted η_t. Both μ and η_t vary with position, and hence they both depend on du/dy (or, alternatively, on τ). The stippling of Figure 4 indicates the area beneath the leftward portion of the rheogram. The area ratio α (which will be used below) is the ratio between the stippled area and the traingular area below the secant line. As with μ and η_t, both the area beneath the rheogram and the area ratio α depend on du/dy (or on τ).

It is common practice to represent rheograms by simple mathematical functions (i.e., functions having only two or three parameters). For materials exhibiting a yield stress, the most common engineering choice is the Bingham model, a straight line given by

$$\tau = \tau_B + \eta_B(du/dy) \tag{7}$$

Here the Bingham viscosity η_B is the tangent viscosity given by the slope of the fit line, and the Bingham yield stress τ_B is the intercept of the fit line with the shear-stress axis. For the data of Figure 4, τ_B is larger than the observed stress at zero strain rate, a situation that is commonly encountered.

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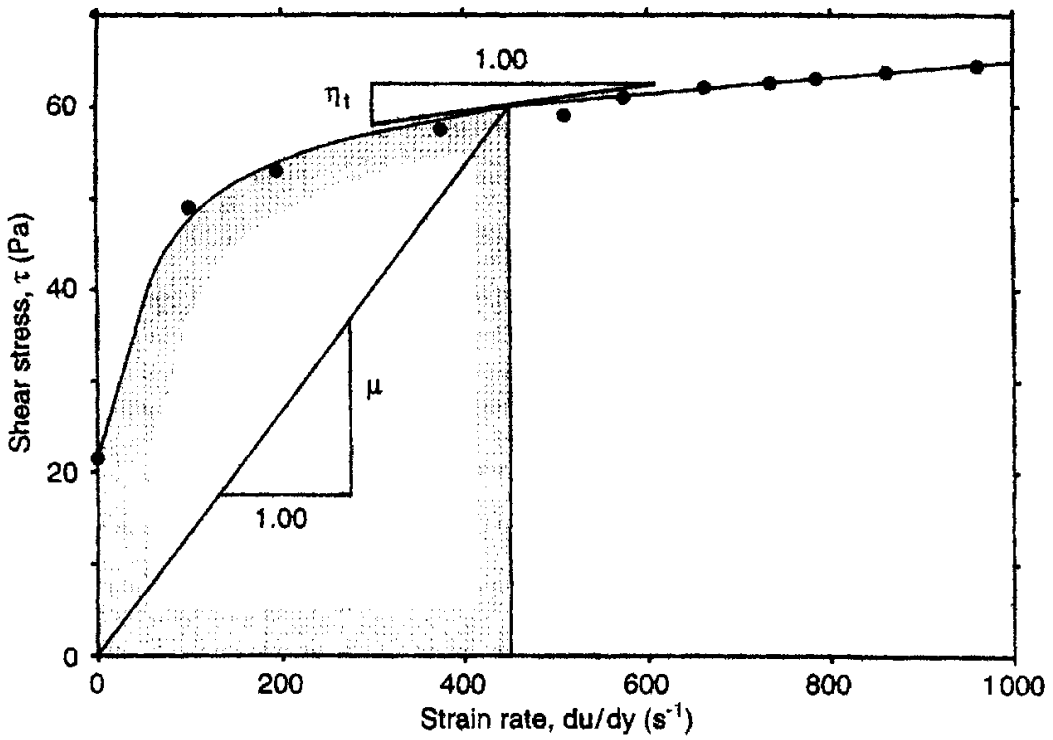


FIGURE 4 Rheogram for phosphate slimes tested at GIW Hydraulic Laboratory (after Wilson, 1986). [Pa × 0.000145 = lb/in²]

The dimensionless ratio τ/τ_B (denoted θ and called the stress ratio) is a useful parameter when dealing with Bingham physics. For example, in laminar flow, the secant viscosity ("apparent viscosity") μ is related to the stress ratio by the equation

$$\mu/\eta_B = \theta/(\theta - 1) \tag{8}$$

This equation indicates a gradual decrease of μ with increasing stress ratio. Another parameter appropriate for Bingham plastics is the Hedström number, *He*

$$He = D^2\rho\tau_B/\eta_B^2 \tag{9}$$

where ρ is the mixture density.

The friction factor for laminar flow of a Bingham plastic can be expressed in terms of these parameters as

$$\sqrt{\frac{8}{f}} = \sqrt{He}\cdot\left(1 - \frac{4\theta^{-1}}{3} + \frac{\theta^{-4}}{3}\right)\cdot\frac{\sqrt{\theta}}{8} \tag{10}$$

As noted previously, turbulent flows of non-Newtonians are often characterized by a roughly constant value of friction factor, in contrast with the large variations typical of laminar flow. An old way of estimating f for turbulent flow, proposed by Hedström (1952), is to use the Bingham viscosity η_B in place of μ in the relationship for Newtonian turbulent flow. This approach should be considered as only a rough approximation. A better method is to begin with, say, the smooth-wall Newtonian equation

$$V_m = 2.5U_*\ell n(\rho DU_*/\mu) \tag{11}$$

where ℓn denotes the natural logarithm and the shear velocity U_*, that is, $\sqrt{\tau/\rho}$, is evaluated at the pipe wall. For turbulent pipe-flow tests of a non-Newtonian material, all quantities except μ will be available for each test run. Therefore, in each case Eq. 11 can be

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solved for the viscosity, which is now denoted μ_{eq}, the equivalent turbulent-flow viscosity. This viscosity is not constant (as would be the case for a Newtonian fluid, or for Hedström's postulate, i.e., $\mu_{eq} = \eta_B$) but varies with wall shear stress, and hence with shear velocity, as described by Wilson et al. (1997).

If turbulent-flow tests are available, scale-up to prototype size can be accomplished directly (as shown in the following section). Otherwise, μ_{eq} can be predicted on the basis of the laminar-flow rheogram. The prediction method follows the model of Wilson & Thomas (1985), which in turn is based on a proposal of Lumley (1973, 1978) regarding the viscous sublayer. Increasing the size of the dissipative micro-eddies leads to an increase in the thickness of this sublayer. The thickened sublayer, in turn, produces a higher mean velocity for the same wall shear stress. The model uses the ratio of the integrals under the non-Newtonian and Newtonian rheograms (denoted by α, and defined earlier in connection with Figure 4) to estimate the size increase of the turbulent micro-eddies. It is found that both the micro-eddy size and the thickness of the viscous sublayer should be multiplied by a factor equal to α. This results in a simple expression for the equivalent turbulent-flow viscosity. For a Bingham plastic α can be expressed in terms of the stress ratio as $(\theta + 1)/\theta$, and the expression for μ_{eq} becomes

$$\mu_{eq}/\eta_B = [(\theta + 1)/(\theta - 1)]\exp(-4.64/\theta) \tag{12}$$

A plot of this function (Wilson, 1997) shows that, as θ increases above 1.0, μ_{eq}/η_B begins by dropping below unity, descends to a minimum (0.216 at $\theta = 1.32$) and then rises gradually toward unity. The resulting expression for turbulent-flow friction factor depends only on the parameters *He* and θ. The equivalent expression for laminar flow (Eq. 10) involves the same two parameters. The intercept of turbulent and laminar lines occurs where these two equations give the same value of f. At this intercept it is possible to eliminate θ using an iterative technique, thus producing a plot of f (at intercept) versus Hedström number. On logarithmic coordinates the plot is not far from a straight line in the region $200 \leq \sqrt{He} \leq 10000$, and a power-law approximation can be used in this range, as follows:

$$f \approx 0.0877(He)^{-0.113} \tag{13}$$

Experimental evidence from pipe-flow tests of fine-particle slurries typically shows abrupt laminar-turbulent transitions, followed in the turbulent range by a plateau of virtually constant friction factor. Equation 13 gives a valuable tool for predicing f for this plateau. This friction factor falls below the traditional curve of Hedström, by which the equivalent turbulent-flow viscosity is equated to the Bingham tangent viscosity, η_B.

This technique can be extended to predict the transition between laminar and turbulent flow. For a Newtonian fluid, with a single rheological parameter (μ), the transition is defined by a specific value of a single dimensionless ratio (the Reynolds number $\rho V_m D/\mu$). A second rheological parameter, required for a non-Newtonian material such as a Bingham plastic, introduces the need for a second dimensionless ratio. In this case the two ratios are the "Bingham" Reynolds number, that is, $Re_B = \rho V_m D/\eta_B$ and Hedström number which has been previously defined. As pointed out by D. G. Thomas (1963) the laminar-turbulent transition must now be defined by a functional relationship linking Re_B at transition with *He*.

Following the method used in obtaining Eq. 13, the laminar and turbulent relations can be equated at the intercept and f (as well as θ) can be eliminated to obtain a plot of Re_B at intercept versus *He*. It was found that the result could be closely approximated by the simple relationship that Re_B equals $25\sqrt{He}$. On cancelling common quantities, it is seen that the flow velocity at transition, say V_T, is given by:

$$V_T \cong 25\sqrt{\tau_B/\rho} \tag{14}$$

The approach of D. G. Thomas (1963) produced a similar form in the limit, but with a coefficient of 19 rather than 25.

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Scale-up of Laminar and Turbulent Flows The laminar flow of any given fluid can be scaled from one pipe size to another by use of the Rabinowitsch-Mooney technique provided that the flow remains laminar in both pipes (turbulent flow will be dealt with separately below). The basis of the Rabinowitsch-Mooney technique is the plot of wall shear stress τ_0 versus $8V_m/D$. Rabinowitsch (1929) and Mooney (1931) proved that for steady uniform laminar flows in a pipe τ_0 and $8V_m/D$ are linked in a functional relationship. In other words, for a given material the values of both τ_0 and $8V_m/D$ can be determined from experiments in a single pipe. The experimentally determined plot or function linking these two variables can then be applied to all laminar flows of the material in question, whatever the pipe diameter.

The application of the method is based on a logarithmic plot of τ_0 versus $8V_m/D$ obtained from a pipe of internal diameter D_1. For some other pipe, of diameter D_2, the pressure gradient dp/dx can be obtained from the plotted values of τ_0 by multiplying them by $(4/D_2)$. Since the factors by which each of the coordinates are multiplied depend only on D_2, a simple rescaling of the axes of the figure based on D_1 gives a plot of dp/dx versus V_m for a pipe of diameter D_2. For plots on logarithmic coordinates, this type of scaling amounts to a pair of linear translations, which do not affect the shape of the plot. In this regard it should be noted that a Newtonian fluid shows a slope of unity on a logarithmic plot of pressure gradient versus mean velocity, whereas the slopes of logarithmic plots for non-Newtonian laminar flow (such as those on Figure 3) are much flatter.

An alternative way of dealing with scale-up to a larger pipe is to transform the data from the experimental pipe (diameter D_1) to the prototype (diameter D_2) on a point-by-point basis. This process is similar to using the affinity laws for scaling head and discharge data for a centrifugal pump. For the pipe, the analogous quantities are the frictional gradient i_m (ft of water/ft or m of water/m) and the mean velocity V_m. With the subscripts 1 and 2 denoting experimental and prototype conditions, respectively, the scaling relations (affinity laws) for laminar flows in pipes are:

$$(i_m)_2 = (i_m)_1 \left(\frac{D_1}{D_2}\right) \tag{15}$$

and

$$(V_m)_2 = (V_m)_1 \left(\frac{D_2}{D_1}\right) \tag{16}$$

If it is foreseen that flow in the prototype pipeline may be turbulent, it is important to extend the small-scale experimental tests into the turbulent-flow region, and then to scale these test results up to prototype size. For turbulent scaling, as for the laminar case, the quantity that remains unchanged is the wall shear stress τ_0. As in laminar flow, the wall shear stress fully determines the stress distribution within the pipe. For example, with a material having a yield stress τ_y a specified value of τ_0 is sufficient to give the fraction of the pipe area where $\tau < \tau_y$, a factor which can affect the mean velocity (Wilson & Thomas, 1985; Thomas & Wilson, 1987). Likewise, a given value of τ_0, and hence of U_*, determines conditions within the viscous sublayer and hence establishes both μ and the area ratio α. The result is that τ_0 (or U_*) is sufficient to determine the equivalent turbulent-flow viscosity. For each value of τ_0 two scaling laws apply to turbulent pipe flow. The first, for pressure gradient, is the same as that established for laminar flow, that is, Eq. 15. The second scaling law, for mean velocity, is given by

$$(V_m)_2 = (V_m)_1 + 2.5U_* \ell n(D_2/D_1) \tag{17}$$

As with Eqs. 15 and 16 for laminar flow, the combined use of Eqs. 15 and 17 allow scale-up of data for turbulent non-Newtonian flow from one pipe size to another. Wilson (1986) gives an example where scaled-up values from a small pipe are compared to measured data in a larger pipe, with very satisfactory agreement. Equation 17 is also valid for cases where μ is constant, that is, for Newtonian fluids, and for turbulent flow of dilute polymer solutions exhibiting drag reduction (Wilson, 1989).

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TABLE 1 Data for phosphate slimes slurry in 8 in. (203 mm) pipe

	V_m				τ_0		U_*	
Run	ft/s	m/s	$8V_m/D(s^{-1})$	i_m	lb/ft^2	Pa	ft/s	m/s
1	1.7	0.53	21.0	0.1004	1.04	49.9	0.69	0.210
2	5.0	1.52	60.1	0.1130	1.17	56.2	0.73	0.223
3	6.6	2.00	78.8	0.1150	1.19	57.2	0.74	0.225
4	8.5	2.59	102.1	0.1189	1.24	59.1	0.75	0.229
5	10.6	3.24	127.9	0.1218	1.27	60.1	0.76	0.231
6	12.5	3.81	150.3	0.1237	1.29	61.5	0.77	0.233
7	14.5	4.43	174.7	0.1273	1.32	63.4	0.78	0.237
8	16.8	5.12	202.0	0.1348	1.40	67.0	0.80	0.243
9	18.5	5.64	222.6	0.1472	1.53	73.2	0.83	0.254

EXAMPLE 1 A clay-water slurry of phosphate slimes is to be pumped over a horizontal distance of 2300 ft. (700 m), using a pipe of internal diameter 12 in. (305 mm). The slurry will be taken from a pond in which the relative density of the mixture is 1 · 13. Tests have been carried out with this material using a pipe of internal diameter 8 in. (203 mm). Test data for i_m and V_m appear in Table 1 together with values of $8V_m/D$ and τ_0 (the wall shear stress equals $\rho_w g i_m D/4$). Figure 5 shows the values of i_m plotted versus V_m for the test pipe.

The preliminary design of the pumping system calls for a discharge Q_m of 4770 gpm (0.30 m^3/s) in the prototype pipe. However, this value is not yet definite, and Q_m values of 3180 gpm (0.20 m^3/s) and 6360 gpm (0.40 m^3/s) are also to be considered.

a. Find the values of i_m for the three values of Q_m just noted. First, the pipe area (0.785 ft^2 or 0.0731 m^2) is used to obtain the required velocities, that is, 9.0, 13.5 and 18.0 ft/s (2.74, 4.11 and 5.48 m/s).

The next step is to scale the data points from the test pipe to the prototype pipe. For the laminar-flow points the appropriate scaling laws are given by Eqs. 15 and 16, which show that i_m scales inversely with the diameter ratio while V_m scales directly with this ratio. Points from test runs 1 to 7 have been scaled on this basis and the first six are shown on Figure 5. For test runs 8 and 9 the flow is turbulent. The hydraulic gradient can still be scaled by Eq. 15, but V_m must now be scaled by Eq. 17. This requires evaluation of U_*, that is, $\sqrt{\tau_0/\rho}$. For example, run 8 with τ_0 = 0.01 lb/ft^2 (67.0 Pa) has U_* = 0.8 ft/s (0.243 m/s), for which Eq. 17 gives a scaled-up value of V_m equal to 17.6 ft/s (5.37 m/s).

The laminar-turbulent transition point is obtained by projecting the turbulent line back to intercept the laminar line. For the data in the test pipe this intercept occurred to the right of point 7, at a velocity of about 16 ft/s (say 5 m/s). For the prototype pipe the intercept lies between points 4 and 5, with a slightly lower velocity. Note that the scaled values for points 5 to 7 are not physically meaningful, because the equivalent flows in the prototype pipe are turbulent, not laminar.

Figure 5 shows that conditions will be laminar for the two lower flows to be investigated, and turbulent for the highest flow. The values of i_m can be taken directly from the figure, and are listed in Table 2.

The flat curve for laminar flow, plotted on Figure 5, shows a very small rise in i_m as Q_m goes from 7.1 to 10.6 ft^3/s (0.20 to 0.30 m^3/s). On the other hand, the equal increase of Q_m from 10.6 to 14.1 ft^3/s (0.30 to 0.40 m^3/s) requires a much larger increment in i_m, as a result of the shift from laminar to turbulent flow.

b. Suppose now that testing had stopped after run 6, so that no turbulent-data flow points were available.

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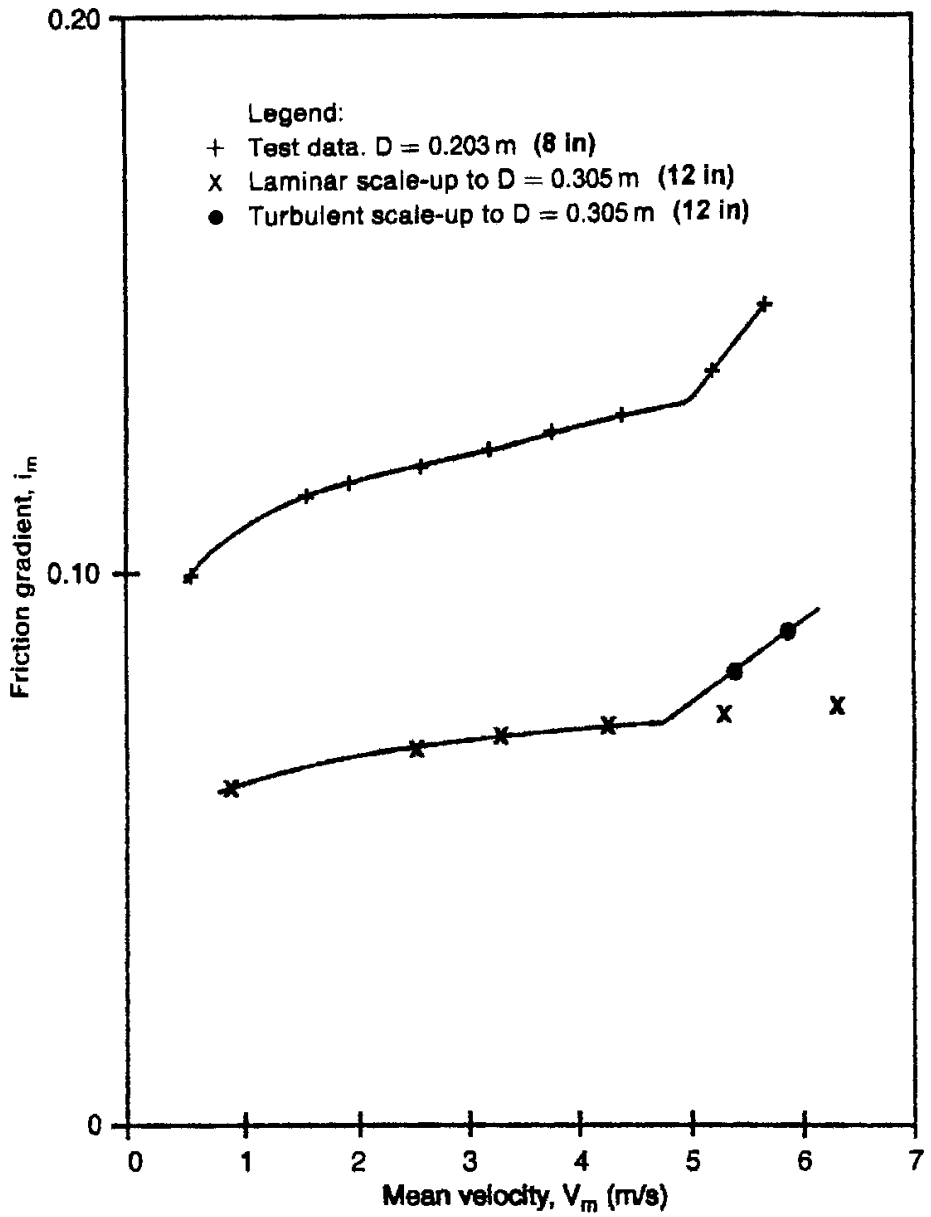


FIGURE 5 Test data and scale-up for Example 1 [m/s × 3.28 = ft/s]

In this case the laminar line would be scaled as before, but the transition to turbulent flow must be predicted from the parameters of the laminar-flow rheogram. It was found that all laminar points except the lowest one can be considered as obeying a Bingham formulation of the rheogram, with $\tau_y = 0.0076$ lb/in^2 (52.7 Pa) and $\eta_B = 0.020$ Pa · s. Substitution of these quantities into Eq. 14 yields $V_T = 17.7$ ft/s

TABLE 2 Hydraulic gradients and pump heads from scaled-up values

Q_m (m^3/s)	0.20	0.30	0.40
Q_m (US gpm)	3180	4760	6360
V_m (m/s)	2.74	4.11	5.48
V_m (ft/s)	9.0	13.5	18.0
i_m	0.0760	0.0795	0.0915
Pump Head (m water)	53.2	55.6	64.0
Pump Head (ft water)	174	183	210

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(5.4 m/s) which is somewhat above the observed transition. The similar expression of D. G. Thomas (1963) gives V_T of 13.4 ft/s (4.1 m/s), significantly below the observed transition point. The use of Eq. 13 to evaluate turbulent friction produces a turbulent line that passes through the transition point predicted by Eq. 14.

SETTLING SLURRIES

Velocity at Limit of Stationary Deposition The deposition limit will be discussed in this section, and the modelling of fully stratified coarse-particle flow at velocities above the deposition limit in the following section.

The results of a detailed force-balance computer model for the limit of stationary deposition showed that the throughput velocity V_m at this limit is concentration-dependent, having small values at low concentration, rising to a maximum (denoted V_{sm}) at some intermediate concentration (which depends on pipe size, particle size, and density) and then dropping off again as the delivered solids concentration approaches the loose-poured value, C_{vb}. This behavior was shown on Figure 1. It should be noted that the velocities used here are obtained simply by dividing the mixture flow rate (Q_m) by the pipe area ($\pi D^2/4$).

The computer output is unwieldy for a designer concerned with many alternative proposals. Moreover, the conservative designer may be content to know only the maximum velocity at the limit of deposition, V_{sm}, since maintaining the operating velocity above this value ensures that deposition will not occur. The value of V_{sm} depends on internal pipe diameter, particle diameter and relative density, and the effect of these variables is expressed concisely by a nomographic chart which was developed at Queen's University (Wilson & Judge, 1978; Wilson, 1979) with the help of the late Professor F. M. Wood's expertise in nomography (Wood, 1935). This chart, reproduced here as Figure 6, is recommended as a practical design aid.

It should be noted, by way of explanation of the chart, that the left-hand panel deals with sand-weight materials (S_s = 2.65). The internal pipe diameter appears on the left vertical axis, with V_{sm} on the central vertical axis. The particle diameter is plotted on a curve known, on the basis of its shape, as the "demi McDonald." This shape illustrates that for large particles, but not for small ones, the shear stress at the interface between the upper and lower layers increases with increasing particle size. Thus, for coarse-particle transport the velocity at the limit of deposition (i.e., the velocity beyond which no bed can remain stationary) will decrease with increasing particle size. This finding, which is amply supported by experimental evidence, shows how the fully stratified mode of transport can display behavior quite different from that of the heterogeneous mode.

To demonstrate the particle size effect consider particles of S_s = 2.65 in a pipe 12 inches (0.30 m) in diameter. This diameter is located on the vertical scale on the left-hand side of the chart, and connected by straight-edge to any desired particle size on the curved scale. V_{sm} is then obtained by projected to the central vertical scale. For instance, a particle size of 0.025 in. (0.6 mm) gives V_{sm} of almost 13 ft/s (4 m/s), which is the largest value found for this pipe diameter and solids density. For a larger particle of, say 0.2 in. (5 mm), the deposition-limit velocity is diminished to about 9 ft/s (2.7 m/s).

When operating with centrifugal pumps it may be difficult to take advantage of the decrease of V_{sm} with increasing particle size. The question is one of obtaining a stable intercept of pump and pipeline characteristics, and will be discussed later in this section. For applications where control of particle size is limited, the conservative designer may wish simply to assume particles of "Murphian" size, i.e. those which give the largest value of V_{sm} for the pipe under consideration. In this case, it should be noted that the values of V_{sm} obtained from Figure 6 tend to be conservatively high, especially for large pipe diameters. Thus, these values can sometimes be used as operating velocities.

A particularly useful feature of nomographic presentation of results is that it gives an immediate indication of the sensitivity of the output to variations in the input. Thus on Figure 6 it is seen that the value of V_{sm} for sandweight solids in a 12-in. (0.30-m) pipe is

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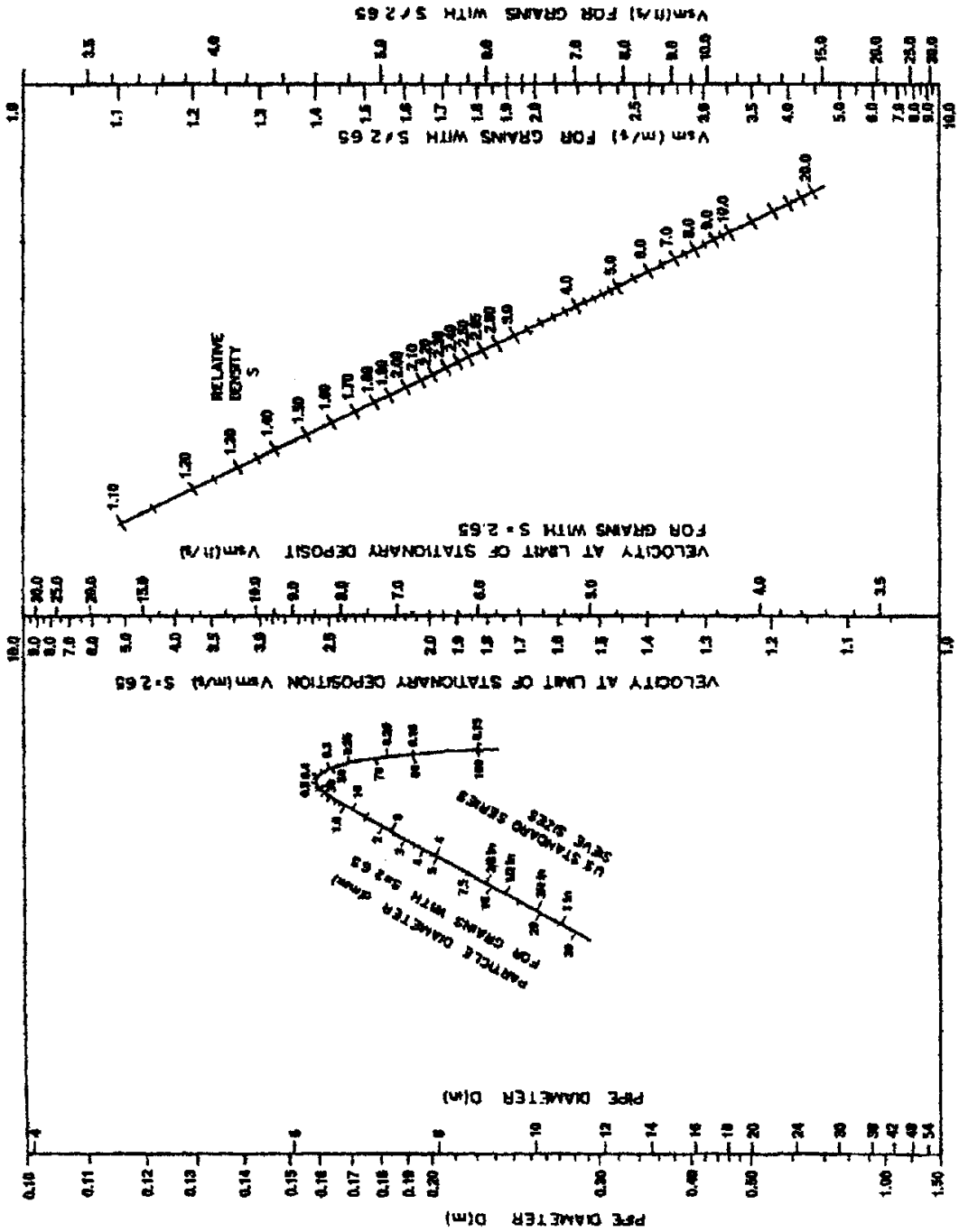


FIGURE 6 Nomographic chart for maximum velocity at limit of stationary deposition (from Wilson, 1979).

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virtually unaffected by a variation of the particle diameter between 0.015 and 0.04 in. (0.4 and 1.0 mm). However, the right limb of the particle-diameter curve shows greatly increased sensitivity, so that a change in d from 0.006 to 0.008 in. (0.15 to 0.20 mm) alters V_{sm} by more than 25%.

It is found that a change of the relative density does not influence the form of the relation shown on Figure 6, but does require its "recalibration." This is accomplished graphically by using the inclined relative-density axis on the right-hand side of the figure. To illustrate the use of the right-hand panel of Figure 6, consider again the pipe 12 inches (0.30 m) in diameter, now carrying particles with a diameter of 0.004 in. (1.0 mm) and relative density 1.50. With 0.004 in. (1.0 mm) entered on the particle scale, a straight line joining it to the pipe diameter can be projected to the central axis of the figure, showing that V_{sm} is 11.5 ft/s (3.5 m/s) for the equivalent sand size. In order to correct this for the relative density of 1.50, the latter value is entered on the sloping axis on the right-hand side of the figure and joined by straight-edge to the point just found on the central axis (11.5 ft/s or 3.5 m/s). The projection of this line to the vertical axis on the right of the figure gives the adjusted value of the maximum deposition velocity, approximately 6 ft/s (1.9 m/s).

Particles that differ in density from sands may also have different values of other properties, including the solids fraction in the deposit and the mechanical friction coefficient between the particles and the pipe. The values of these quantities that were employed in the computer program, and hence are reflected in the nomographic chart, apply to sands but not necessarily to other materials. Therefore the values of V_{sm} determined from Figure 6 for materials other than sands must be treated as somewhat less accurate than values for sand. However, both the solids fraction in the bed and the particle-pipe mechanical friction coefficient occur in the computer program only as multiples of the submerged relative density of the particles, and it is found that any change in these quantities merely gives rise to a multiplicative factor which must be applied to the values of V_{sm} obtained from the figure. This permits calibration of the output of the nomographic chart from a few pilot-plant tests with the material of interest.

The particle diameter scale of Figure 6 has not been extended below 0.006 in. (0.15 mm) since smaller particles tend to be influenced by mechanisms not included in the mathematical model on which the nomogram is based. As shown by Thomas (1979) the viscous sublayer can have a significant effect, and all particles which are small enough to be completely embedded in this sublayer will behave in a fashion which is no longer dependent on particle diameter. In this limiting case certain simplifications can be made in the mathematical model. Thomas found that these lead to a simple expression for the shear velocity at the limit of stationary deposition, and on employing a power law approximation for the friction factor, he then obtained a corresponding expression for V_{sm}

$$V_{sm} = 9.0[g\nu(S_s - S_f)]^{0.37}(D/v)^{0.11} \tag{18}$$

Here ν represents the kinematic viscosity of the carrier fluid (i.e., μ/ρ), and the coefficient of 9.0 applies in any consistent system of units. This equation gives the minimum value of deposition velocity for small particles, assuming turbulent flow.

For cases where the interfacial friction sets up a sheared layer several grain diameters in thickness, a new analysis has been developed (Wilson, 1988; Wilson & Nnadi, 1990). For large pipes, and particles near the Murphian size, this new analysis shows that the particle size no longer influences V_{sm} directly. In such cases the value of V_{sm} is less than that predicted by Figure 6, a point which is in accord with experimental evidence. It is then necessary to obtain the value of V_{sm} from the new analysis of Interfacial friction, for comparison with the value from the nomographic chart.

When the parameters in the new analysis are given values typical for sands, it is found that the dimensionless maximum deposition velocity (i.e., the Durand deposition variable) depends only on the fluid friction factor for the portion of the pipe wall above the deposit. A power-law approximation gives an appropriate fit function for this effect, that is,

$$\frac{(V_{sm})_{\max}}{\sqrt{2gD(S_s - S_f)}} = \left(\frac{0.018}{f_f}\right)^{0.13} \tag{19}$$

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where f_f is the friction factor for fluid alone. If the value of V_{sm} found from the nomographic chart exceeds the value of $(V_{sm})_{max}$ from Eq. 19 the latter value should be used for V_{sm}.

Fully Stratified Coarse-Particle Transport Fully stratified flow occurs where almost all of the particles travel as contact load (i.e., fluid suspension is ineffective). The ratio of particle diameter to pipe diameter is of major importance in determining the presence of this flow type, which does not normally occur for d/D ratios less than 0.015. Fully stratified flow is less likely if the particles are broadly graded, especially if there is a significant homogeneous fraction (i.e., a significant fraction of particles smaller than 200 mesh/75 μm). Calculations made for narrow-graded slurries with water as a carrier fluid indicate fully stratified behavior for values of d/D above 0.018.

Although the actual relationship of the detailed force-balance analysis cannot be expressed in closed form, simple approximating functions can be fitted to match the output of the detailed model. Wilson & Addie (1995) proposed the following expression for approximating fully stratified flow:

$$\frac{i_m - i_w}{(S_m - 1)} = B'\left(\frac{V_m}{0.55V_{sm}}\right)^{-0.25} \tag{20}$$

The ratio on the left-hand side of this equation is the same as that used by Newitt et al. (1955), and the right-hand side shows the decrease of this ratio with increasing V_m/V_{sm}. The coefficient B' has a value of 1.0 for angular particles, and decreases with the degree of rounding. For typical cases the suggested default value is 0.75. As fully stratified flow has higher energy consumption than other flow types, Eq. 20 can be thought of as representing the upper limit of excess pressure gradient (i.e., solids effect). Heterogeneous slurry flows, which are covered in the following section, will generally display smaller values of the solids effect.

Another type of stratified flow that may be encountered is flow above a stationary bed of solids. This type of operation is generally uneconomic, and thus is seldom a deliberate design choice, but it is sometimes found in existing pipelines. Flow over a stationary bed has been studied experimentally, and analyzed by a computer model (Nnadi & Wilson, 1992; Pugh, 1995). For most cases of this type of flow encountered in pipelines the following rough approximation to the computer output may be sufficient. It is based on particles near the "Murphian" size, which tend to be disproportionately represented in stationary deposits:

$$i_m = 0.32(S_s - 1)^{1.05}C_{vd}^{0.6}\left(\frac{V_m}{\sqrt{2gD}}\right)^{-0.1} \tag{21}$$

Pressure Gradients for Partially Stratified Flow As mentioned previously, two methods of support are significant in partially stratified flows: turbulent suspension and granular contact. For turbulent suspension the eddies can carry the particles with no significant additional energy consumption, but granular contacts set up Coulombic friction forces which must be overcome by an additional "solids effect" pressure gradient, represented by $(i_m - i_f)$ where i_f (or i_w) is the hydraulic gradient for fluid only. In many cases this solids effect varies with the submerged weight of solids, i.e., with $(S_s - S_f)C_v$ where S_f is the relative density of the fluid (1.0 for water) and C_v is the volumetric delivered concentration. The quantity $(S_s - S_f)C_v$ can also be expressed as $(S_m - S_f)$ where S_m is the relative density of the mixture. The relative solids effect, written $(i_m - i_f)/(S_m - S_f)$, can be obtained from pipe-loop testing. Typical results for partially stratified flow are shown on Figure 7 for various concentrations of sand with d_{50} of 0.016 in. (0.42 mm). The relative solids effect drops with increasing mean velocity V_m, which in turn produces increased turbulent fluctuating velocities, implying that a larger fraction of particles will be suspended by turbulence. The straight fit line shown on the logarithmic coordinates of Figure 7 indicates that the relation can be approximated by a power law. (Other formulations have also been proposed, see for example Shook & Roco, 1992 and Gillies et al., 1991.)

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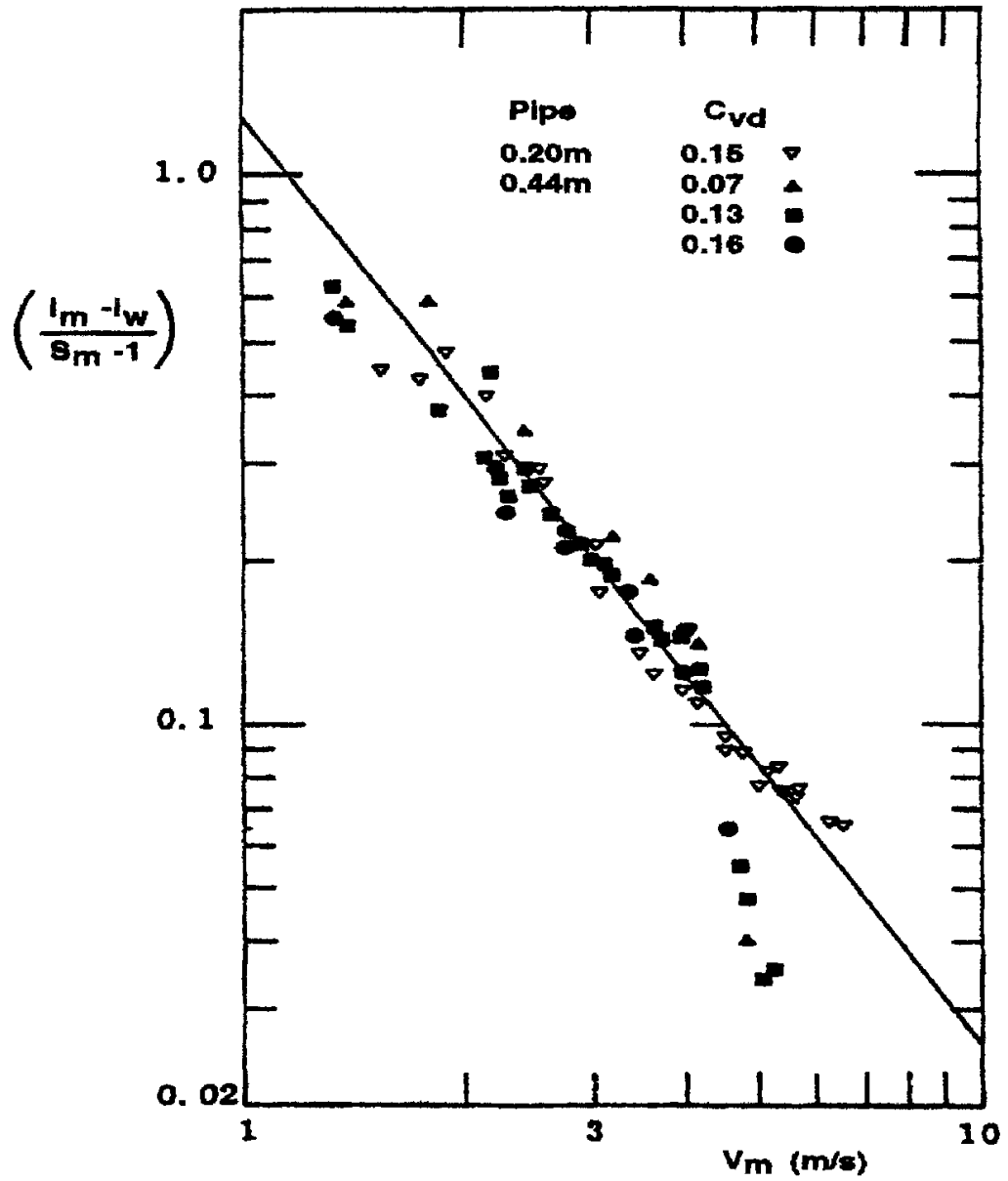


FIGURE 7 Behavior of masonry-sand slurry (d_{50} = 0.42 mm) in pipes of two sizes (after Clift et al., 1983). [m × 3.28 = ft; m/s × 3.28 = ft/s]

In addition to the slope parameter (denoted by M) a power law requires a central-value parameter, denoted here by V_{50}. Conceptually, this represents the value of V_m at which half the mass of solids is supported by granular contact and half by fluid suspension. From a practical standpoint it is necessary to match this point with a specific value of $(i_m - i_f)/(S_m - S_f)$. The mechanics of the situation were discussed by Wilson et al. (1997) who proposed a value of 0.22 and showed that this is suppoted by experimental data. The resulting equation is

$$\frac{(i_m - i_f)}{(S_m - S_f)} \cong 0.22\left(\frac{V_m}{V_{50}}\right)^{-M} \tag{22}$$

If test data are available, M and V_{50} can be obtained directly from the results. For example consider the data of Figure 7 in the 8 in. (0.20 m) pipe. In calculating the ratio $(i_m - i_w)/(S_m - 1)$, the clear water gradient i_w has been used to approximate i_f, and S_f is S_w,

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that is, 1.0. The value of i_w was calculated from the Darcy-Weisbach formula using the friction factor $f_f = 0.013$ measured previously for flows of water in this pipe. The plotted points fall on an essentially straight line on Figure 7. This behavior corresponds to Eq. 22. The slope of the line gives $M \approx 1.7$, which is typical for slurries with narrow particle grading. The point on the line where $(i_m - i_w)/(S_m - 1) = 0.22$ gives V_{50}, which is approximately 9 ft/s (2.8 m/s).

With V_{50} and M obtained for the 8-in. (0.20-m) pipe, scale-up to a larger pipe diameter can be carried out. The larger diameter of 18 in. (0.44 mm) has been selected because data are available with the same sand in this larger pipe (Clift et al., 1982) and thus the scaled-up results can be verified directly. In fact, as seen on Figure 7, the fit line for the larger pipe coincides with that for the smaller pipe in this instance. In this case the clear-water friction factor f_w was found to be the same for both pipes and in both pipes the particle size d was a small fraction of the pipe diameter. It has been found that V_{50} should vary with $(8/f_w)^{1/2}$; it should also depend on the diameter ratio d/D (Wilson & Watt, 1974; Wilson et al., 1997).

These factors can be incorporated in the scale-up procedure, which involves preparing curves of i_m versus V_m for various values of C_{vd} in the larger pipe, the relation for each line of constant C_{vd} being expressed as

$$i_m = \frac{f_w}{2gD} V_m^2 + 0.22(S_s - 1)V_{50}^M C_{vd} V_m^{-M} \tag{23}$$

It is typical to find a moderate decrease in f_w with increasing pipe size, which produces a small increase in V_{50}. Increases in D have the opposite effect, but for small ratios of d/D, as is the case for the data of Figure 7, this effect is not significant.

Other problems arise when pipeline experiments have not been carried out with the particular slurry of interest. In many cases of practical importance, information on the size and grading of the material to be pumped is limited, but estimates of the solids effect must be made. For example, consider the case of an ore that is to be crushed and then transported by pipeline. At the initial stage of the design there may be no adequate sample of the crushed ore, but estimates of the effect of the solids on the head loss must be made for feasibility studies, preliminary designs and cost estimates, and even for justifying the expenses of laboratory or pilot-plant testing.

To estimate the solids effect, two parameters are required: the power M and the velocity V_{50}. The value of M has a lower limit of 0.25 (for fully stratified flow) and approaches 1.7 for slurries with narrow particle grading. If only a rough idea of the grading is available, it may be adequate to use the following approximation, which requires only an estimate of the particle diameter ratio d_{85}/d_{50} (d_{50} is the mass-median particle diameter and d_{85} is the diameter for which 85% by mass of the particles are smaller). Using this evaluation, the approximation for M is written

$$M \approx [\ell n(d_{85}/d_{50})]^{-1} \tag{24}$$

Here ℓn is the natural logarithm. Also, M should not be allowed to exceed 1.7 or fall below 0.25.

The next step is to obtain a commensurate approximation for V_{50} (Wilson et al., 1997). This formula reads

$$V_{50} \approx 3.93 d_{50}^{0.35}[(S_s - 1)/1.65]^{0.45} \tag{25}$$

Here d_{50} is in mm. The coefficient 3.93 applies for velocities in m/s; for velocities in ft/s this coefficient becomes 12.9. With sand-weight solids $(S_s - 1)$ equals 1.65 and the bracketed portion of Eq. 25 equals 1.00. The value of V_{50} obtained from Eq. 25 is substituted into Eq. 22 to obtain the solids effect $(i_m - i_f)$, where i_f is the gradient for an equal flow of water. Note, Eq. 25 is only applicable for 0.006 in $< d_{50} <$ 0.055 in (0.15 mm $< d_{50} <$ 1.4 mm). For larger particles the value of V_{50} given by Eq. 25 should be multiplied by $\cosh(60 d_{50}/D)$.

Remarks on Complex Slurry Flows In the introduction to this section, it was mentioned that slurry flows can be divided into three types on the basis of mechanisms of particle support. These types are homogeneous, partially stratified and fully stratified, and

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the applicable methods of analysis for these flows have been outlined in the appropriate preceding sections. Quite often, the particle grading curve is sufficiently broad to span two of the flow types, or even all three. This gives rise to complex slurry flows. The larger particles, which would settle readily in water, often receive considerable support from the smaller particles and the carrier fluid, promoting efficient transport. A complete analysis of such flows is not yet available, but it is hoped that the following remarks will aid the design engineer.

Whenever some coarse particles settle, they form contact load. As shown in earlier sections of this chapter, this has an effect on pressure drop which is quite different from that of particles suspended by the fluid. The contact-load effect, analyzed previously for the case of a Newtonian carrier fluid, must eventually be combined with the scaling laws for non-Newtonian fluids presented in an earlier part of this chapter. As laminar flows which have significant particle settling are usually avoided in design, only turbulent flows will be considered here.

Maciejewski et al. (1993) compared large-diameter transportation of coarse particles of about 4 in. (100 mm) in clay suspensions and in oil-sand tailings slurries (particle size below 0.03 in./0.8 mm). They found that the sand slurry was more effective as a transport medium than a viscous, homogeneous clay slurry. The important role of particles with sizes of 0.004 to 0.020 in. (0.1 to 0.5 mm) in reducing friction was further shown in studies by Sundqvist et al. (1996a, 1996b) for products with d_{50} of 0.024 to 0.027 in. (0.6 to 0.7 mm) and various size distributions, with maximum sizes of up to 6 in. (150 mm).

In studying the behavior of complex slurries like these, it is logical to begin by dividing the total concentration of solids C_v into three components, each associated with a support mechanism. Thus C_h stands for homogeneous, C_{mi} for partly stratified (the "middlings") and C_{cl} for the coarse fully stratified particles (the "clunkers"). On the basis outlined previously the particle size of 200 mesh (75 μm) separates C_h and C_{mi} and the size $0.018D$ separates C_{mi} and C_{cl}. This point is best illustrated by an example. Take $S_s = 2.65$; and a concentration of 30% by volume ($C_v = 0.30$). From the solids grading curve, suppose that 20% of the total is slimes, 50% middlings and 30% clunkers. Thus, the concentration of slimes in the slurry is (0.30)(0.20) = 0.06, and similarly 0.15 and 0.09 for middlings and clunkers, respectively. The equivalent fluid based on the slimes has specific gravity $S_h = 1 + (S_s - 1)C_h = 1 + (1.65)(0.06) = 1.099$ and that for the combined slimes and middlings is $S_{hmi} = 1 + (1.65)(0.21) = 1.347$. Thus, for the middlings the specific gravity difference between solids and carrier fluid is (2.650 − 1.099) = 1.551 (rather than 1.650). For the clunkers, the equivalent difference is (2.650 − 1.347) = 1.303.

The homogeneous fraction now forms the carrier fluid for the rest of the slurry, and its hydraulic gradient i_h replaces i_w in equations like Eq. 20 and Eq. 22. These are used to determine the solids effect for middlings and clunkers, which may be written Δi_{mi} and Δi_{cl}. The gradient for the mixture i_m represents the sum of i_h and the solids effects for the middlings and the clunkers, i.e.

$$i_m = i_h + \Delta i_{mi} + \Delta i_{cl} \tag{26}$$

The homogeneous gradient i_h is based on appropriate equivalent-fluid or non-Newtonian calculations, as given previously. For the middlings, Δi_{mi} is effectively equivalent to $(i_m - i_f)$ in Eq. 22, when applied to the middlings only, giving

$$\Delta i_{mi} = C_{mi}(S_s - S_h)0.22\left(\frac{V_m}{V_{50}}\right)^{-M} \tag{27}$$

Here, S_h is the relative density of the homogeneous "carrier fluid" component (1.099 for the example just introduced). The evaluation of M and V_{50} will be mentioned in the following text.

For the clunkers, Δi_{cl} is based on Eq. 20, except that the carrier fluid for the clunkers now includes both the homogeneous portion and the middlings, with a relative density written S_{hmi} (1.347 for the example). The carrier fluid will also have an effect on the coefficient which will now be written B'' instead of B' (the evaluation of B'' will be mentioned next). The relation for Δi_{cl} is

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$$\Delta i_{cl} = C_{cl}(S_s - S_{hmi})B''\left(\frac{V_m}{0.55V_{sm}}\right)^{-0.25} \tag{28}$$

For coarse solids only, it was suggested above that the parameter B' in Eq. 20 should have a default value of 0.75. When considerable fractions of middlings are found, B' must be reduced to give B''. As it is expected that the reduction is associated with density differences, the appropriate variable would appear to be the weight concentration, C_w, of fines plus middlings, i.e., C_{whmi} (e.g., $C_{whmi} = 0.372$). Thus

$$B'' = B'(1 - \xi C_{whmi}) \tag{29}$$

where ξ is a coefficient with a proposed default value of unity.

Another point concerns the evaluation of V_{50} and M for the middlings. The methods of Eqs. 24 and 25 are basically applicable but d_{50} and d_{85} must now refer to the middlings fraction only, rather than the whole grading curve. For very broad well-distributed gradings the middlings will simply be a slice between 0.003 in. (0.075 mm) and $0.018D$. Here it may be possible to assume that d_{50} is the geometric mean of the two diameter limits (e.g., 0.025 in. for $D = 1$ ft or 0.64 mm for $D = 0.305$ m). For this case a default value of about 1.0 might be appropriate for M. V_{50} will be influenced by the rheological properties of the homogeneous fraction, which will influence the fall velocity of the middling particles, thus enhancing suspension. An initial approach is to consider the homogeneous part of the slurry to act as a Newtonian fluid with effective viscosity significantly higher than that of water. The effect of enhanced viscosity on V_{50} is of the form shown on Figure 8. On this figure the value of V_{50} based on water viscosity is entered on the abscissa (corrected for relative density of solids if different from 2.65). The various curves are for different μ_r, the ratio of viscosity to that of water at 70°F (20°C). The multiplier read from the ordinate can be applied to the V_{50} value for water to estimate V_{50} in the higher-viscosity carrier fluid.

EXAMPLE 2 This example pertains to fully stratified flow of coarse magnetite particles ($d = 1$ in. or 25 mm). Because of their high relative density ($S_s = 4.4$) these particles are to be used as ballast for an offshore drilling rig. The magnetite will be transferred from ore carriers to the rig by a dredge pipeline with internal diameter 19.7 in. (0.500 m).

a. Find the deposition limit in this line, V_{sm}. Figure 6 is entered with the pipe diameter (on the left axis) and the particle diameter (on the demi McDonald).

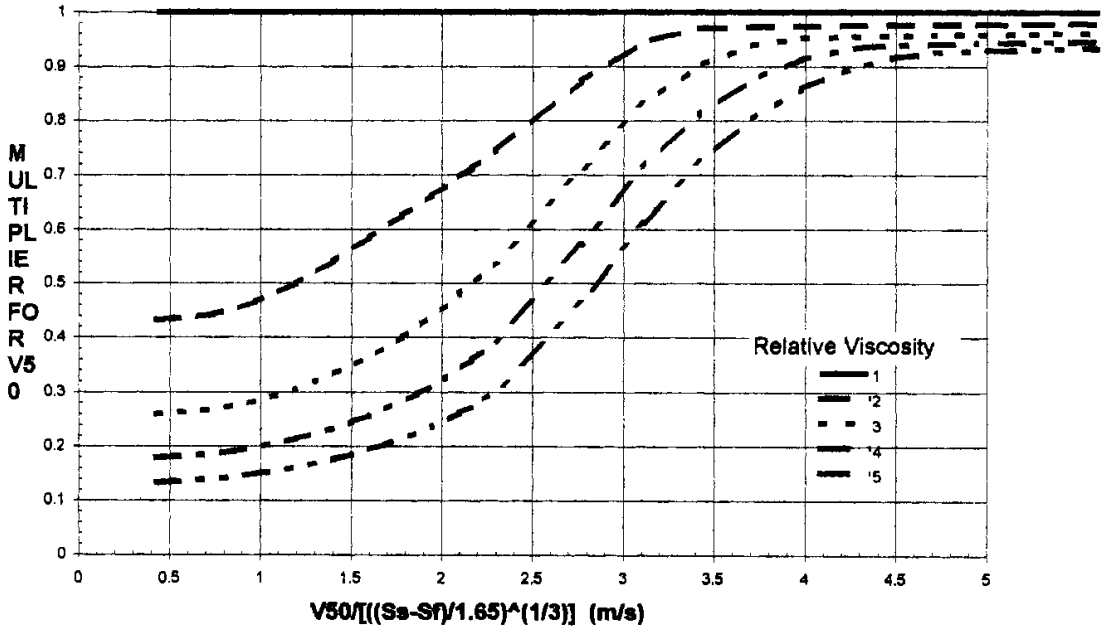


FIGURE 8 Variation of V_{50} with viscosity

Projection to the central axis gives V_{sm} = 8.7 ft/s (2.65 m/s) for sand-weight material. This number is joined to the relative density S (i.e., S_s) of 4.4 on the sloping axis and projected to the right-hand axis to give V_{sm} = 13 ft/s (4.0 m/s) for the magnetite.

b. Find the hydraulic gradient i_m for a volumetric solids concentration of 0.10 and a throughput velocity V_m = 16.4 ft/s (5.0 m/s). Assume rather angular material with B' = 0.9. The first step is to calculate the clear-water friction gradient of i_w. Using f_w = 0.013, i_w is given by $0.013V_m^2/2gD$ or 0.033 (height of water per length of pipe). To this must be added $(i_m - i_w)$, which Eq. 20 gives as $(S_m - 1)B'(V_m/0.55V_{sm})^{-0.25}$. The quantity $(S_m - 1)$ equals $C_v(S_s - 1)$ i.e., (0.10)(3.4) = 0.34. B' has been taken as 0.9 and the ratio $(V_m/0.55V_{sm})$ equals 2.27, giving $(i_m - i_w)$ = 0.250 and i_m = 0.28, i.e., 28 ft of water per 100 ft of line or 0.28 m of water per m of line. Very large gradients like this, which have been verified in prototype-scale testing, show why fully stratified flow is only of interest for short-haul applications.

EXAMPLE 3

a. Suppose that the magnetite of the previous example has been ground to give d_{50} = 0.008 in. (0.20 mm) and d_{85} = 0.012 in. (0.30 mm). As in the previous example, D = 1.64 ft (0.50 m) and V_m = 16.4 ft/s (5.0 m/s). In this case the volumetric concentration C_v = 0.20. Find the V_{sm} and the hydraulic gradient i_m.

V_{sm} is found as in the previous example. The pipe size is joined to the particle size of 0.008 in. (0.20 mm) and projected to the central axis to give the deposition velocity for sand-weight solids. Solids specific gravity is entered on the sloping line, and projected to the right-hand axis to give V_{sm} = 14.4 ft/s (4.4 m/s). As in the previous example this is less than the proposed operating velocity, which is satisfactory.

The flow of this slurry will be partially stratified, and from Eq. 24, $M = [\ell n(0.30/0.20)]^{-1}$ = 2.5. However, the maximum limit of M is 1.70, which will be used here. From Eq. 25,

$$V_{50} = 3.93(0.20)^{0.35}\left(\frac{3.40}{1.65}\right)^{0.45} = 3.1 \text{ m/s } (10 \text{ ft/s})$$

For this value of V_{50}, with M = 1.7 and V_m = 16.4 ft/s (5.0 m/s), the right hand side of Eq. 22 becomes 0.098. $(S_m - 1)$ equals $(S_s - 1)(C_v)$ = (3.40)(0.2) = 0.68. Hence
$i_m = i_w + (0.68)(0.098)$, and with i_w of 0.033 (found in Example 2) i_m is found to be 0.100 (ft water per ft or m water/m). Note that this gradient is only about one-third that for the coarse particles of the previous example, despite the fact that the solids concentration (and the tonnage transported) is twice as much.

b. As in part (a) but with d_{85} = 0.016 in. (0.40 mm). The power M is reevaluated as $M = [\ell n(0.40/0.20)]^{-1}$ = 1.44. For the unchanged values of V_{50} and V_m, the right-hand side of Eq. 22 becomes 0.110 and i_m equals 0.033 + 0.68 (0.110) or 0.108 (ft water/ft or m water/m).

c. As in part (a) but assume that there is a fine-particle fraction that increases the viscosity of the homogeneous component to 4 times that of water. From part (a), M = 1.70 and V_{50} = 10 ft/s (3.1 m/s) in water. Adjusting for the density difference from the sand-water case gives the abscissa of Figure 8 as $3.1/(3.40/1.65)^{0.33}$ i.e., 7.88 ft/s (2.4 m/s). With this abscissa and a relative viscosity of 4, Figure 8 gives a multiple of 0.42, for a revised estimate of V_{50} as (0.42)(3.1) = 4.3 ft/s (1.3 m/s). Thus the right-hand side of Eq. 22 beocmes 0.022, and i_m = .033 + 0.015 or 0.048 (ft water/ft or m water/m). This revised estimate is only about one-half the value of 0.100 obtained in part (a) for pure water as the carrier fluid.

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OTHER FACTORS

Vertical Flows For industrial application of vertical transportation of a solid-liquid mixture in a pipe, the operating velocity must be sufficient to maintain a continuous flow of solids at the discharge end. However, unnecessarily high velocity causes excessive pipe wear and energy losses. The appropriate operating velocity depends on the settling conditions of the solids, indicating that size, density, and concentration of particles are key parameters in the hydraulic design of a vertical particle-fluid transportation system.

Many experimental studies have been made of vertical slurry transport. For example, Sellgren (1979) used a pilot-scale facility with a centrifugal pump to investigate important design parameters for ores and industrial minerals taken from in-plant crushing and milling. The results of these experiments are summarized in the following paragraphs.

It is suggested that the allowable minimum mixture velocity be based on the settling velocity of the largest particles in still water multiplied by a factor of 4 or 5. Provided the velocity exceeds this value, then in most industrial applications, with volumetric concentrations of 15% to 30%, the corresponding pressure requirement in the vertical system can be determined by the equivalent-fluid model. As noted in connecetion with Eq. 5, this model is based on the density of the slurry and the friction factor for water. Applied to vertical flow, it gives

$$p = \rho_w g S_m z \left(1 + \frac{f_w V_m^2}{2gD} \right) \qquad (\text{for } V_m > V_{\text{all}}) \tag{30}$$

Here p is the pressure required, S_m is the relative density of the slurry, z is the length of vertical pipe, and V_{all} is the allowable minimum mixture velocity, approximately four times the settling velocity of the largest particle.

The settling velocity in still water of industrially crushed mineral particles is normally reduced significantly compared to smooth spheres of corresponding size. Tests have shown that, on average the settling velocity for particles in the range of 0.04 in. to 1.2 in. (1 mm to 30 mm) is reduced approximately 50%. Therefore, the criterion previously given for the minimum allowable velocity could alternatively be formulated as: V_{all} is twice the settling velocity of a smooth sphere of the same size as the largest particles.

Within the constraints previously discussed, systems operate under conditions where the effect of relative velocity between the components appears to be negligible. The maximum particle sizes considered are in the range of 0.04 in. to 1.2 in. (1 mm to 30 mm) in pipe diameters of 4 to 12 in. (0.1 m to 0.3 m). With larger particle sizes (up to 4 to 6 in./100 to 150 mm) and low concentrations, the relative velocity between the components becomes significant. Boundary-layer transitional effects may also introduce certain instabilities that must be carefully evaluated in long vertical risers. Particles larger than one-fifth the pipe diameter can promote slugging instability in vertical hoisting, and particles larger than one-third the pipe diameter may jam the pipe and should be avoided.

EXAMPLE 4 Centrifual slurry pumps are used to pump a sand slurry (d_{50} = 0.06 in./1.5 mm) out of a quarry. The pipe is vertical with a length of 328 ft. (100 m) and a diameter of 4 in. (0.10 m). Tests have shown that the settling velocity of the largest particles is approximately 1.5 ft/s (0.45 m/s). Select the operating velocity and calculate the head requirement in meters of slurry.

Solution Following the guidelines previously given, the velocity V_{all} is four times 0.45, i.e., 6 ft/s (1.8 m/s). At this velocity, $V_m^2/2g$ is 0.54 ft (0.165 m). With the friction factor f_w taken as 0.016 for smooth-pipe conditions (see Figure 2), the head is obtained from Eq. 30 as

$$328\left[1 + \frac{0.016(0.541)}{3.94/12} \right] = 336.5 \text{ ft of slurry}$$

in USCS units or 336.5 S_m in ft of water.

In SI units the calculation becomes

$$100\left[1 + \frac{0.016(0.165)}{0.1}\right] = 102.6 \text{ m of slurry}$$

or 102.6 S_m in meters of water.

Inclined Flows Lengths of pipe with an adverse slope often form part of pipelines transporting solids. Compared to the horizontal case, flow up an incline tends to require higher throughput velocities in order to avoid deposition. This is of greatest significance for coarse-particle flow.

In an experimental investigation carried out by Wilson & Tse (1984), four particle sizes between 0.04 and 0.24 in. (1 and 6 mm) were tested in a pipe at angles of inclination up to 40° from the horizontal. It was found that the velocity at the limit of deposition initially increases with the angle of upward inclination, θ, reaching a maximum when this angle is about 30°. For the materials tested this maximum velocity was approximately 50% larger than that required to move a deposit in a horizontal pipe. This large difference is clearly a matter of importance for both design and operation of pipelines with inclined sections.

As indicated schematically on Figure 1, V_{sm} marks the lower end of the range of desirable operating velocities for a pipeline. For purposes of comparison, it is appropriate to represent the deposition limit for inclined flow in terms of the dimensionless velocity, or Durand number, $V_{sm}/[2g(S_s - 1)D]^{1/2}$. The difference between the Durand number for inclined flow and that for horizontal flow, ΔD, is plotted against θ on Figure 9.

For the effect of pipe inclination on friction loss, the following widely used formula by Worster & Denny (1955) may be employed for heterogeneous flows and homogeneous flows in which the hold-up effects are small. Their approach, based on water as the carrier fluid, deals with the extra pressure gradient ($\Delta i(\theta)$, expressed in height of water per length of pipe) beyond that for pumping water alone. For horizontal flow, this extra gradient is simply the solids effect ($i_m - i_w$), which may be written $\Delta i(0)$. Worster and Denny's formula states that

$$\Delta i(\theta) = \Delta i(0)\cos\theta + (S_s - 1)C_{vd}\sin\theta \tag{31}$$

For highly stratified flows, Eq. 31 underestimates the losses. For further information see, for example, Wilson et al. (1997).

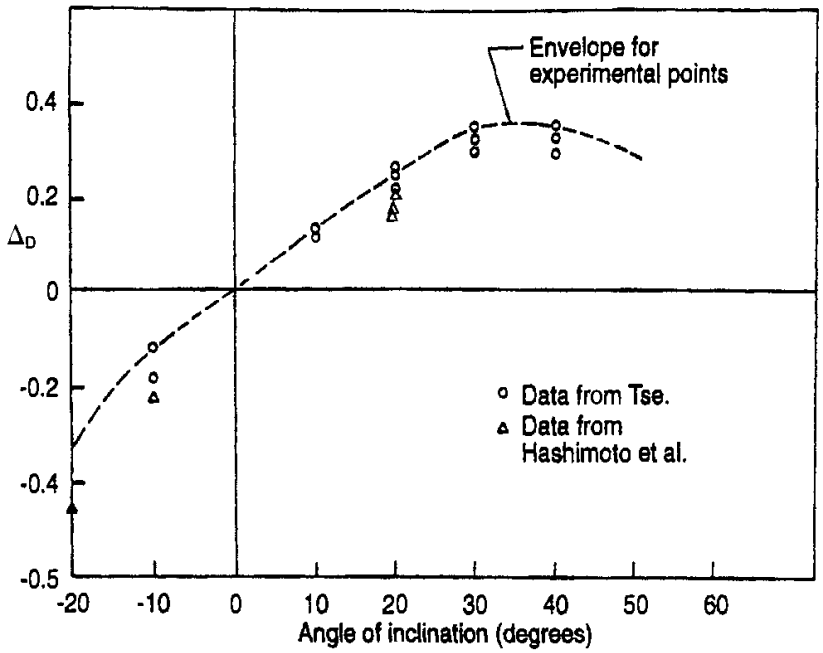


FIGURE 9 Effect of angle of inclination on Durand deposition parameter (after Wilson and Tse, 1984)

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EXAMPLE 5 Consider an inclined suction pipe located on a dredge ladder. Pipe length = 59 ft (18 m), D = 25.6 in. (0.65 m), angle to the horizontal = 30°. Determine the deposition velocity and the head losses at the proposed operating velocity of 21.3 ft/s (6.5 m/s). Assume S_s = 2.65 and C_v = 0.20. For a horizontal pipe, it is calculated that V_{sm} = 15.7 ft/s (4.8 m/s) and at the proposed operating velocity i_w = 0.0373, $(i_m - i_w)$ = 0.0239.

Solution From Figure 9, at 30°, ΔD = 0.33, equivalent to an increase in V_{sm} of

$$0.33[(64.4) \cdot 1.65(25.6/12)]^{1/2} = 5.0 \text{ ft/s}$$

in USCS units, or

$$0.33[(19.62)(1.65)(0.65)]^{1/2} = 1.5 \text{ m/s}$$

in SI units.

The resulting limit velocity in the inclined pipe is 15.7 + 5.0 = 20.7 ft/s, 4.8 + 1.5 = 6.3 m/s, which is slightly less than the proposed operating velocity. At the operating velocity Eq. 31 gives the excess gradient, $\Delta i(30°)$, as

$$\Delta i(30°) = 0.0239 \cos 30° + (1.65(0.20) \sin 30°$$

where 0.0239 represents the solids effect in the horizontal pipe. Thus $\Delta i(30°)$ equals 0.0207 + 0.1650 or 0.1857. The clear water gradient i_w was 0.0373, giving a total value of 0.223, and on multiplying this by the suction-pipe length of 59 ft (18 m), the drop is found to be 13.2 ft (4.0 m) of water.

SYSTEM DESIGN AND ANALYSIS

Operating and Economic Considerations In order to compare the merits of different transport systems, it is necessary to have a measure of the energy required to move a given quantity of product over a given distance. In slurry transport, the solids are normally the "payload" while the conveying liquid is merely the "vehicle." The specific energy consumption is therefore to be related to the solids transported rather than to the mixture. However, the power which the pump must supply is used to drive the slurry as a whole, and is given by $\rho_w g Q_m H$ where H is the head in height of water and Q_m is the volumetric flow rate of the mixture. The payload of solids is delivered over the line length, L, at a volumetric rate that is the product of the delivered solids concentration C_{vd} and the mixture flow rate. If the application is a hoisting system, with the line directed vertically upwards, the rate at which energy is being added to the solids is $\rho_s g C_{vd} Q_m L$ or $\rho_w g S_s C_{vd} Q_m L$, and the efficiency of the system can be obtained by dividing the power input, giving $S_s C_{vd} L/H$. In this case the specific energy consumption is the inverse of the efficiency i.e.,

$$\text{SEC} = \frac{H}{S_s C_{vd} L} \tag{33}$$

where, as before, H is the head supplied by the pump in height of water.

In the more usual case of an essentially horizontal pipeline, potential energy will not be added to the solids, and thus from the physicist's viewpoint no work will be done. However, in industrial practice transporting solids along the line represents useful work in the economic sense. We can no longer speak of efficiency in the physicist's terminology, but the appropriate economic measure of specific energy consumption is still given by Eq. 33. With H/L now equal to the friction gradient, i_m, the expression is written

$$\text{SEC} = \frac{i_m}{S_s C_{vd}} \tag{34}$$

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To obtain a value in horsepower-hour/ton-mile, this ratio is multiplied by 5.33, and for kWh/tonne-km the factor is 2.73. If the power supplied to the pump is required, the expression must be divided by pump efficiency, and the efficiency of the drive train or motor can be taken into account in the same way.

The lower the SEC, the more energy-effective the pipeline is as a means of transport. Since S_s is fixed by the nature of the solids, the quotient i_m/C_{vd} is the basic variable to be considered.

Homogeneous Slurries Figure 10 is from Chap. 15 (written by Clift) of Wilson et al. (1997). The figure shows in simple generalized terms, the effect of changes in solids concentration on the resistance of a piping system to flow of a homogeneous, non settling slurry. At velocities sufficiently high for the slurry to be in turbulent flow, the head lost in flowing through the system (measured as head of a fluid with the slurry density) can be estimated by treating the slurry as a fluid with an effectively constant friction factor, as shown in connection with Eq. 13. However, in laminar flow, to the left of the turbulent curve, the system head varies strongly with concentration, as shown schematically in Figure 10 where $C_1 < C_2 < C_3$. The transition between laminar and turbulent flow is indicated roughly by the intersection of the laminar system characteristic with the turbulent curve. Therefore, as illustrated, the transition velocity increases somewhat with solids concentration.

To examine the operability of such a system with centrifugal pumps, we now superimpose the pump head-discharge characteristic on to the system characteristic. For fine slurries the head delivered by the pump, evaluated in terms of the slurry density, is virtually unaffected by the solids. Therefore the characteristic of a fixed speed pump appears as a single curve in the coordinates of Figure 10, independent of slurry concentration.

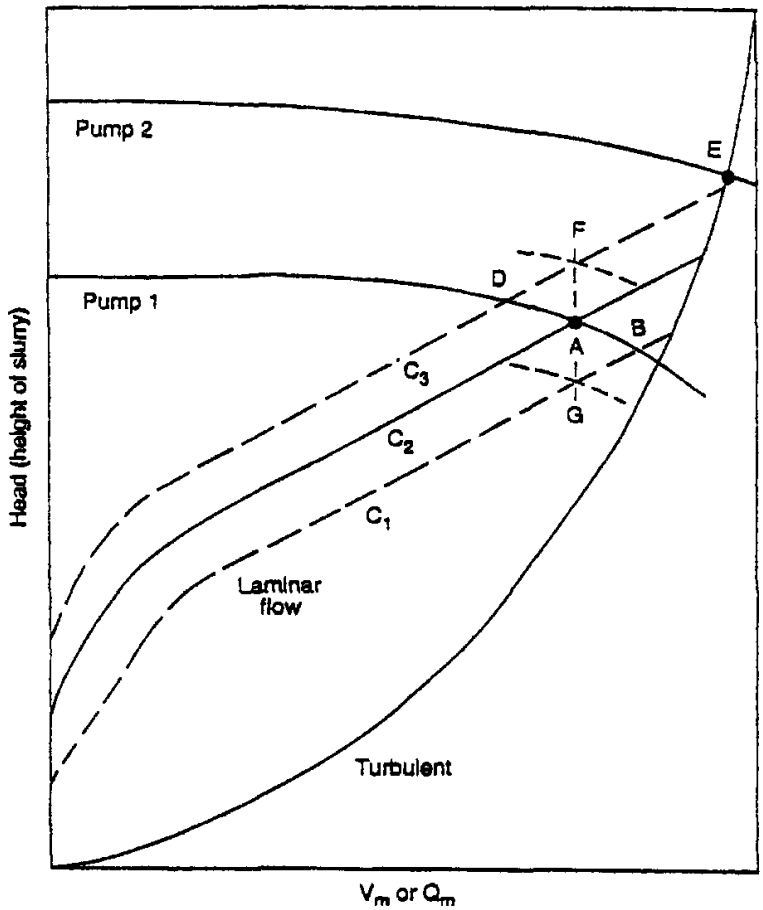


FIGURE 10 Schematic system and pump characteristics for flow of a homogeneous slurry at three concentrations (from Wilson *et al.*, 1997)

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Consider first a system which has been designed to operate in the laminar range, corresponding to point A in Figure 10 with slurry concentration C_2. The curve labeled "Pump 1" represents the characteristic of a pump selected for this operating point.

Figure 10 shows the effect of varying solids concentration. If it decreases to C_1 then the system operation shifts to point B, the new intersection of the system and pump characteristics. Similarly, a concentration increase to C_2 shifts operation to point D. It was shown on Figure 3 that relatively small variations in concentration can have strong effects on the laminar flow curves. Thus the shifts from point A may result from quite small changes in slurry consistency. Nevertheless, as shown by Figure 10, the associated changes in mean velocity (and hence solids throughput) are amplified by the way the system and pump characteristics intersect. Thus steady operation in laminar flow with a fixed-speed centrifugal pump and with no flow control valve is possible only if the slurry consistency is tightly controlled.

For variable solids concentration, two alternatives are available. One is to operate in turbulent flow, at a point like E in Figure 10. "Pump 2" shows the head-discharge characteristics of a centrifugal pump selected for this duty. By operating on the turbulent system characteristic, variations in slurry consistency can be accommodated without significant changes in mixture velocity. This is probably the main reason why many designers prefer to use turbulent flow even for homogeneous slurries. The design point is then selected to be in the turbulent range for the highest solids concentration expected. However, it should be noted that this will not correspond to the lowest specific energy consumption. This is an example of the general point that design for variable operation is usually incompatible with design for minimum energy consumption.

The other option is to use a variable-speed pump. This is illustrated by points F and G in Figure 10. If the solids concentration increases from C_2 to C_3, then the pump speed is increased to keep a constant flow rate. The corresponding increase in pump speed raises the pump head-discharge characteristic to the broken curve shown passing through F. Similarly, if the concentration falls from C_2 to C_1, the pump is slowed down to give the characteristic passing through point G. Again, it must be noted that there is an economic penalty for designing for variable concentration, this time represented by the cost of the variable-speed drive and the energy losses in the drive train.

Settling Slurries We now turn to problems of system operability for settling slurries. (For a more detailed treatment see Wilson et al., 1997.) Figure 11 shows typical "system characteristics" for a settling slurry at three delivered concentrations, in terms of head of slurry, j_m. Only the frictional contribution for horizontal transport is considered here. The form of the system characteristics is typical of heterogeneous flow of a settling slurry. The three slurry characteristics are scaled for three notional delivered relative densities, i.e., three values of S_{md}. The slurry curves all show the minimia typical of a heterogeneous slurry flow. As is typical for heterogeneous slurries, the minimum friction gradient occurs at velocities above the deposition point and the position of the minimum moves to somewhat higher velocities as the solids concentration increases.

Figure 11 also shows the point where j_m equals i_w. At this crossing point (but not elsewhere) the heterogeneous slurry behaves like an "equivalent fluid," with a value of 1.0 for the coefficient A' in the homogeneous-flow equation (Eq. 5).

It is now appropriate to consider the behavior of this system with one or more variable-speed centrifugal pumps, with characteristics shown in Figure 11 as "Pump (water)" and "Pump (mixture)." The latter decreases with increasing concentration, due to the effects of solids on pump performance discussed earlier in this section. Therefore, to maintain a constant head and flow rate at point A in Figure 11, the pump rotational speed must be adjusted for the actual solids concentration.

It sometimes happens in industrial practice that the pipeline characteristics for different concentrations cross over at a single point at the water curve and at a suitable operating velocity, as shown schematically in Figure 11. This behavior is expected for closely graded materials with volumetric concentrations less than 20%, but may not apply for broad gradings or higher concentrations. The results with the broadly graded sand in Table 3 show that j_m was approximately constant (equal to 0.055 over a concentration range 20% to 40%) for the selected velocity = 15 ft/s (4.5 m/s). However, the operating point

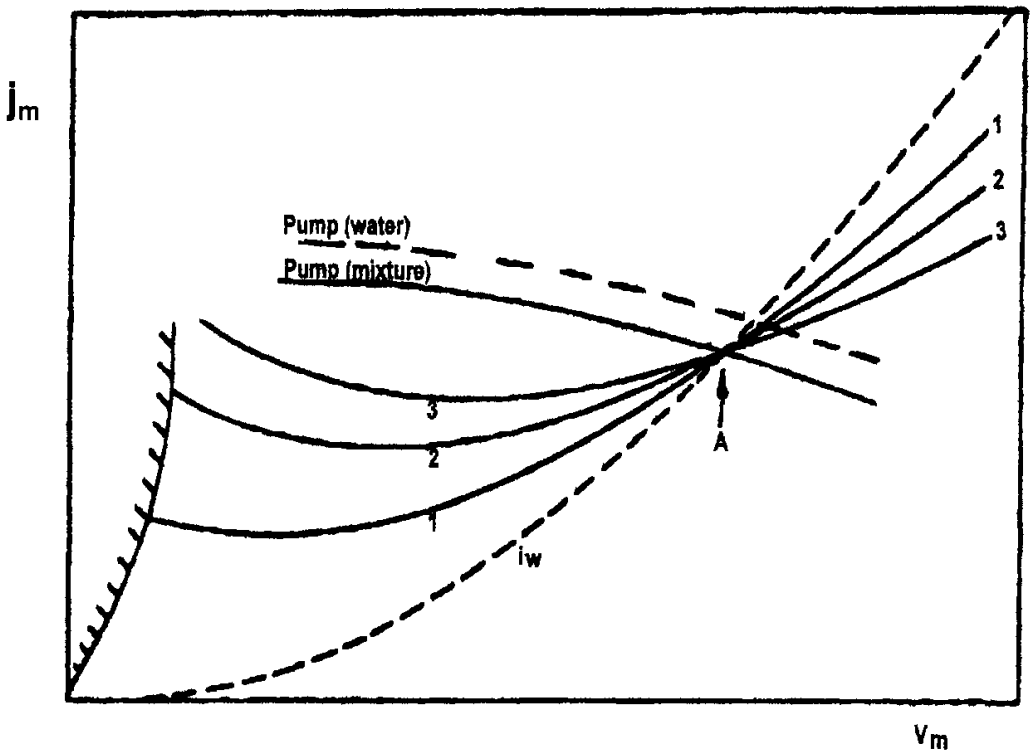


FIGURE 11 Schmematic system and pump characteristics for heterogeneous (settling) slurry flow at various delivered concentrations (after Wilson *et al.*, 1997)

was not located on the water curve (i_w = 0.041). It follows from Table 3 that the slurry friction losses, j_m, exceeded the corresponding water losses, i_w, by 35%.

The operating data for the sand in Table 3 are of great practical interest since they demonstrate the energy-efficiency of transporting this product at very high solids concentrations. Experiments by Sellgren & Addie (1993) with this sand indicated that the pump solids effect was not influenced greatly by increased solids concentrations. For the highest concentration studied here, the maximum values of R_H and R_η were estimated to be 13 and 15%, respectively. An example including pipeline friction and the effect of solids on the water efficiency of the pump is given in Table 4.

It follows that the energy required to overcome friction losses decreased from 0.039 to 0.024 HPh/ton (0.032 to 0.020 kWh/tonne) when C_{vd} was increased from 20 to 40%. The decrease corresponds to about 37%. The capacity of the pipeline is in this case doubled by the increase in concentration. Because the pump solids effect for this product and pump

TABLE 3 Comparison of slurry friction losses and water friction losses for a broadly graded sand with $d_{85}/d_{50} \approx 7.1$. Data from Sundqvist et al. (1996a)

Product	Sand		
Max. particle size, in. (mm)	0.4 (10)		
Average, d_{50}, in. (mm)	0.025 (0.65)		
Portion of fine particles, X_h%	0		
Pipeline—Diameter, in. (m)	12 (0.3)		
Velocity, ft/s (m/s)	15 (4.5)		
Water friction losses, i_w	0.041		
Conc. by volume, C_v	0.20	0.30	0.40
Slurry friction losses, j_m	0.055	0.055	0.055

TABLE 4 Data for the sand slurry of Table 3 in a 12 in. (0.3 m) diameter pipe using a 14 in. (0.35 m) by 12 in. (0.3 m) conventional heavy-duty pump with an impeller diameter of 36 in. (0.91 m). Pipeline length = 852 ft (260 m), flow rate = 5200 US gpm (328 l/s) and velocity = 15 ft/s (4.5 m/s). Volumetric concentrations 20, 30, and 40% (from Sellgren and Addie, 1996)

	Concentrations by volume (%)		
	20	30	40
Capacity, ton/h (tonne/h)	690 (627)	1035 (941)	1380 (1255)
Energy to overcome pipeline friction, HPh/ton (kWh/tonne)	0.039 (0.032)	0.029 (0.024)	0.024 (0.020)
Pump head reduction, R_H (%)	8	11	13
Required rotary speed (rpm)	333	338	342
Pump water efficiency (%)	79.5	79.4	79.3
Pump efficiency reduction, R_η (%)	6	11	15
Pump efficiency (%)	74.8	70.7	67.4
Total energy, HPh/ton (kWh/tonne)	0.052 (0.043)	0.041 (0.034)	0.037 (0.030)

remained fairly moderate also for the highest concentration, the total energy requirement per tonne decreased from 0.052 to 0.037 (0.043 to 0.030) with an increased C_{vd} value from 20% to 40%. The lowering then corresponds to about 30%.

The attrition of coarse and angular particles during circulation in a test-loop may have a strong influence on friction losses and pump performance. However, the results in Tables 3 and 4 with high solids concentrations and coarse particles embedded in a sand slurry are not expected to have been significantly influenced by circulation effects. In large-scale loop testing special loading and quick measurement procedures are now used to evaluate the interaction of particle attrition, pipeline friction, and pump performance. In general, designing a system to transport a slurry combining broad grading and high concentration requires pilot plant testing using the same material that will be pumped in the prototype line.

The solids concentration used throughout this section (denoted C_v or C_{vd}) refers to delivered concentration, and this is incorporated in the design-oriented equations. Large particles tend to have a lower velocity than small ones, and thus a larger resident (*in situ*) concentration in the pipeline, and this feature should also be considered in designing once-through pipelines on the basis of recirculating test-loop data.

For long-distance freight pipelines it is worthwhile to devote considerable effort to preparing and controlling slurry properties. For such lines solids concentrations range from 0.25 (25%) for ores with S_s of, say, 5.0 to 0.40 (40%) for coal with S_s = 1.4. Velocities are usually maintained below 6.6 ft/s (2.0 m/s). For a typical 10 in. (0.25 m) pipe, annual throughput of solids range from about 2.8 M tons (2.8 M tonnes) for ore to 1.4 M tons (1.3 M tonnes) for coal. The corresponding specific energy consumption, including a total pump and drive-train efficiency of 75%, are respectively, 0.13 and 0.17 HPh/ton-mile (0.065 and 0.085 kWh/ton-km). For such cases the energy cost for pumping amounts to roughly half the annual operating cost. This cost is, say, 20% to 30% of the total annual cost, which is seen to be dominated by repayment of capital expenditure.

REFERENCES

Blatch, N. S. (1906). Discussion of "Works for the purification of the water supply of Washington D.C." (Hazen, A. and Hardy, E. D.) *Trans Amer. Soc. Civil Engrs.*, Vol. 57, pp. 400–409.

Brown, N. & Heywood, N. (1992). *Slurry Handling Design of Solid-Liquid Systems.* Elsevier Science Publishers, United Kingdom.

Carstens, M. R. & Addie, G. R. (1981). A sand-water slurry experiment. *Jour. Hydr. Div.* ASCE, Vol. 107, No. HY4, pp. 501–507.

Clift, R. Wilson, K. C., Addie, G. R. and Carstens, M. R. (1982). A mechanisitically-based method for scaling pipeline tests for settling slurries.

Durand, R. (1951a). Transport hydraulique des matériaux solides en conduite, études expérimentales pour les cendres de la central Arrighi. *Houille Blanche,* Vol. 6, No. 3, pp. 384–393.

Durand, R. (1951b). Transport hydraulique des graviers et galets en conduite. *Houille Blanche,* Vol. 6, No. B, pp. 609–619.

Gillies, R. G., Shook, C. A. and Wilson, K. C. (1991). An improved two layer model for horizontal slurry pipeline flow. *Canad. J. Chem. Engng.,* Vol. 69, pp. 173–178.

Hedström, B. O. A. (1952). Flow of plastics materials in pipes. *Ind. and Eng. Chem.,* Vol. 44, No. 3, pp. 651–656.

Kostuik, S. P. (1966). Hydraulic hoisting and pilot-plant investigation of the pipeline transport of crushed magnetite. *The Canadian Mining and Metallurgical Bulletin,* January, 25–38.

Lumley, J. L. (1973). Drag reduction in turbulent flow by polymer additives, *J. Poly Sci., Macromol, Rev. 7,* A. Peterlin (Ed.), Interscience, New York, pp. 263–290 (1973).

Lumley, J. L. (1978). Two-phase flow and non-Newtonian flow. In *Turbulence,* P. Bradshaw (Ed.), Tiopics in Applied Physics, Vol. 12, Springer-Verlag, Berlin (1978), Chap. 7.

Maciejewski, W., Oxenford, J. and Shook, C. (1993). Transport of Coarse Rock with Sand and Clay Slurries, Hydrotransport, Vol. 12, Brügge, Belgium, pp. 705–724.

Mooney, M. (1931). Explicit formulas for slip and fluidity. *J. Rheol.* Vol. 2, p. 210 ff.

Newitt, D. M., Richardson, J. F., Abbot, M., & Turtle, R. B. (1955). Hydraulic conveying of solids in horizontal pipes. *Trans. Inst. of Chem. Engrs.,* Vol. 33, London, U.K.

Nnadi, F. N. & Wilson, K. C. (1992). Motion of contact load at high shear stress. *J. Hydr. Engrg.,* ASCE, Vol. 118 (12), pp. 1670–1684.

Pugh, F. J. (1995). *Bed-load Velocity and Concentration Profiles in High Shear Stress Flows.* Ph.D. Thesis, Queen's University, Canada.

Rabinowitsch, B. (1929). Über die Viscosität von Solen. *Zeitschrift physik. Chem.,* A 145, p. 1 ff.

Sellgren, A. & Addie, G. (1996). Pump and pipleine solids effects of transporting sands with different size distributions and concentrations. *Proc. 13th Int. Conference on the Hydraulic Transport of Solids in Pipes,* Johannesburg, South Africa, pp. 227–236.

Sellgren, A. & Addie, G. (1998). Effective integrated mine waste handling with slurry pumping. *Proceedings Fifth International Conference on Tailings and Mine Waste.* January 26–28, Ft. Collins, USA, pp. 103–107.

Shook, C. A. and Roco, M. C. (1991). *Slurry Flow Principles and Practice.* Butterworth-Heinemann.

Sundqvist, A., Sellgren, A. and Addie, G. R. (1996a). Pipeline friction losses of coarse and slurries: comparison with a design model, in *Powder Technology,* Vol. 89, pp. 9–18.

Sundqvist, A., Sellgren, A. and Addie, G. R. (1996b). Slurry pipeline friction losses for coarse and high-density industrial products, in *Powder Technology,* Vol. 89, pp. 19–28.

Thomas, A. D. (1979). The role of laminar/turbulent transition in determining the critical deposit velocity and the operating pressure gradient for long distance slurry pipelines, *Proc. Hydrotransport 6,* BHRA Fluid Engineering, Cranfield, UK, pp. 13–26.

Thomas, D. G. (1963). Non-Newtonian suspensions. Part 1, physical properties and laminar transport characteristics. *Ind. and Eng. Chem.* Vol. 55, No. 11, pp. 18–29.

Thomas, A. D. & Wilson, K. C. (1987). New analysis of non-Newtonian turbulent flow—yield-power-law fluids. *Canad. J. Chem. Engrg.,* Vol. 65, pp. 335–338.

LIBERTY EXHIBIT 1017, Page 564

Wilson, K. C. (1979). Deposition-limit nomograms for particles of various densities in pipeline flow, *Proc. Hydrotransport 6,* BHRA Fluid Engineering, Cranfield, UK, pp. 1–12.

Wilson, K. C. (1986). Modelling the effects of non-Newtonian and time-dependent slurry behavior. *Proc. Hydrotransport 10,* BHRA Fluid Engineering, Cranfield, UK, pp. 283–289.

Wilson, K. C. (1988). Evaluation of interfacial friction for pipeline transport models. *Proc. Hydrotransport 11,* BHRA Fluid Engineering, Cranfield, UK, pp. 107–116.

Wilson, K. C. (1989). Two mechanisms for drag reduction. *Drag Reduction in Fluid Flows, Techniques for Friction Control,* Ed. H. R. J. Sellina nd R. T. Moses, pp. 1–8. Ellis Horwood Ltd., Chichester, UK.

Wilson, K. C. & Addie, G. R. (1995). Coarse-particle pipeline transport: effect of particle degradation on friction. *Proc. 8th International Freight Pipeline Society Symposium,* pp. 151–156.

Wilson, K. C., & Judge, D. G. (1978). Analytically-based nomographic charts for sand-water flow, *Proc. Hydrotransport 5,* Solids in Pipes, BHRA Fluid Engineering, Cranfield, UK, pp. A1-1–11.

Wilson, K. C., & Nnadi, F. N. (1990). Behavior of mobile beds at high shear stress. *Proc. 22nd Int'l Conference on Coastal Engineering,* Delft, Netherlands, Vol. 3, pp. 2536–2541.

Wilson, K. C., & Thomas, A. D. (1985). A new analysis of the turbulent flow of non-Newtonian fluids. *Canad. J. Chem. Engrg.,* Vol. 63, pp. 539–546.

Wilson, K. C., & Tse, J. K. P. (1984). Deposition limit for coarse-particle transport in inclined pipes. *Proc. Hydrotransport 9,* BHRA Fluid Engineering, Cranfield, UK, pp. 149–169.

Wilson, K. C. & Watt, W. E. (1974). Influence of particle diameter on the turbulent support of solids in pipeline flow. *Prod. Hydrotransport 3,* BHRA Fluid Engineering, Cranfield, UK, pp. E1-1–E1-13.

Wilson, K. C., Addie, G. R., Sellgren, A. and Clift, R. (1997). *Slurry Transport Using Centrifugal Pumps, 2nd Ed.* Blackie (Chapman & Hall).

Wood, F. M. (1935). Standard nomographic forms for equations in three variables, *Canadian Journal of Research,* Vol. 12.

SECTION 4.2
APPLICATION AND CONSTRUCTION OF CENTRIFUGAL SOLIDS HANDLING PUMPS

GRAEME ADDIE
ANDERS SELLGREN

This section discusses the application of centrifugal pumps to the transport of slurries as introduced in Section 4.1.

Where the distance is large and the size of the solids less than about 100 microns, positive displacement pumps are usually applied. If the solids to be transported are smaller than about 60 mm in size, water is readily available or already part of the process, and the distance is less than 12 mi (20 km), a pipeline driven by centrifugal slurry pumps is usually more cost-effective than belt conveyors, trucks, or railway transportation.

Types of solids transported vary from sag mill feed in a copper mine, iron ore, phosphate matrix, coal, rock salt, tar sands, red mud, various types of waste tailings, and crushed rock and silt. An 18 in (0.45 m) diameter pipeline, for example, is capable of transporting as many as 2200 tons/h (2000 tonnes/h) of sand. This equates to about one hundred truckloads per hour over the distance involved.

The application that involves the largest quantities is in the dredging industry, continually maintaining navigation in harbors and rivers, altering coastlines, and mining material for landfill and construction purposes. Because a single dredge may be required to maintain a throughput of 7000 tons/h of slurry or more, very large centrifugal pumps are used. Figures 1 and 2 show, respectively, an exterior view of a dredge pump on test, and a view of a large slurry pump impeller (Addie and Helmly, 1989).

Because the aim is to transport solids and not water, the higher the concentration of solids, the better for energy consumption and capital cost. This does, however, mean that wear due to the abrasive solids will be significant, requiring a special type of centrifugal pump design and the use of special materials. Slurry pumps may be selected for low-concentration dirty-water service. Most of what follows, however, is about pumps designed and built to give cost-effective operation for heavy-duty slurry service, probably involving significant wear.

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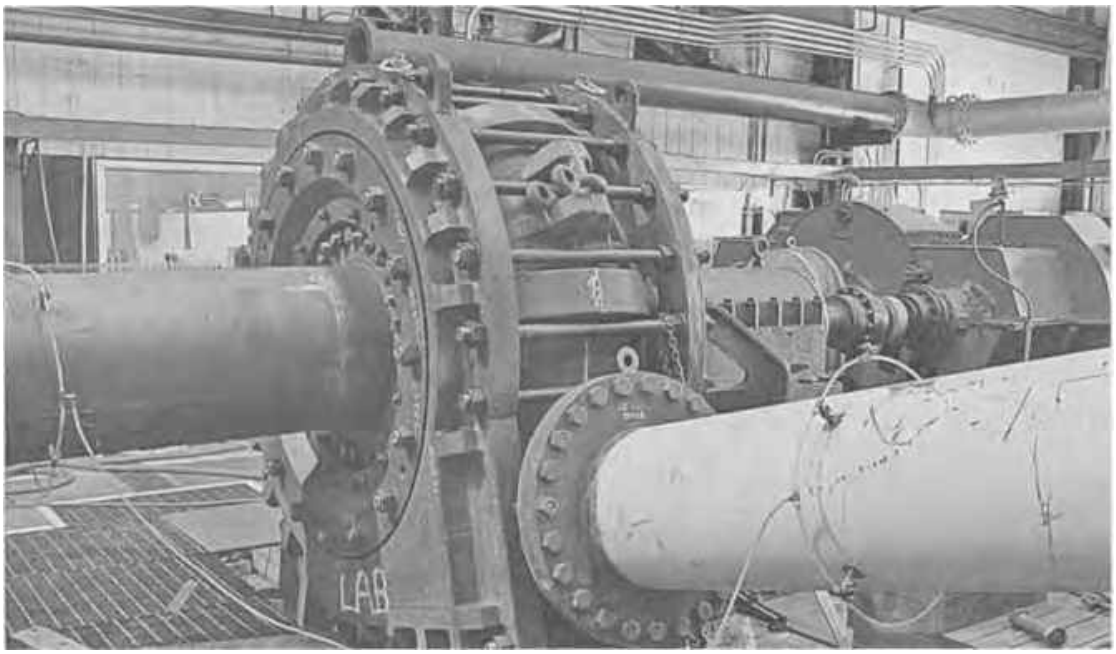


FIGURE 1 Testing a dredge pump at the GIW hydraulic laboratory

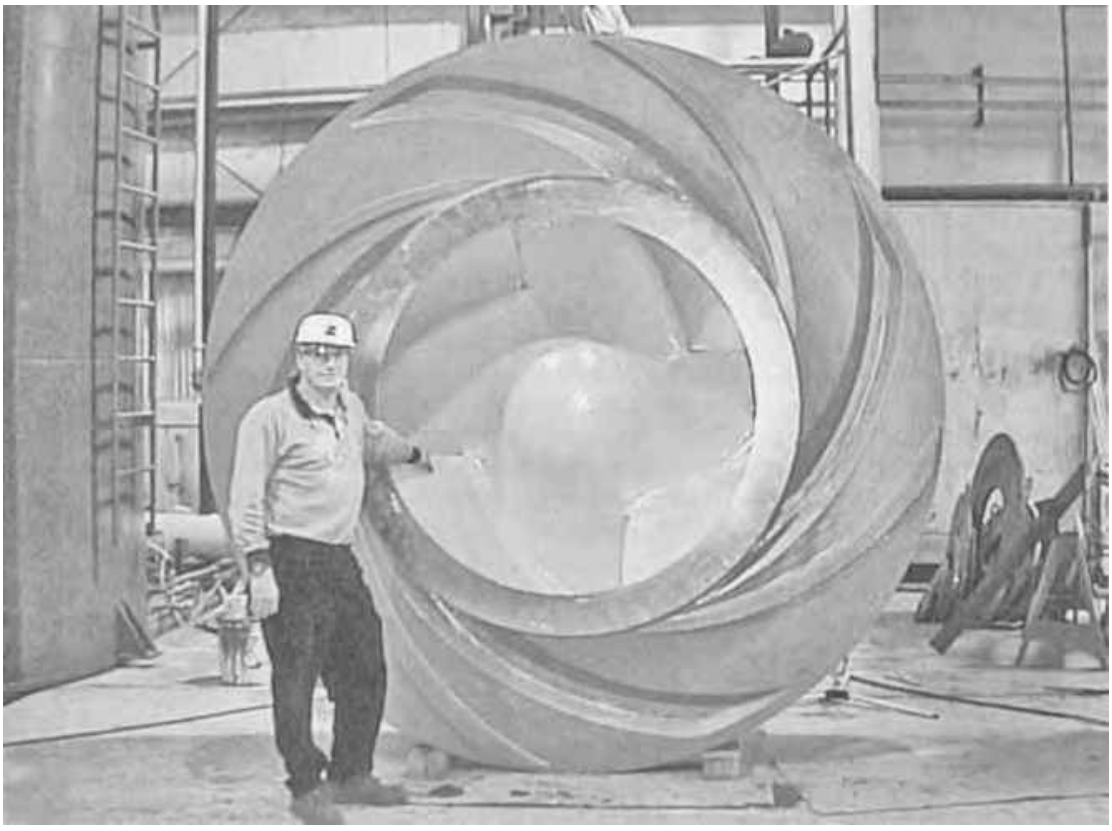


FIGURE 2 Dredge pump impeller, 113 in (2.87 m) diameter

SLURRY PUMP TYPES

The majority of centrifugal slurry pumps that fit the above definition are constructed as the single-stage horizontal radially split type. A sectional arrangement drawing of a single-wall metal type is shown in Figure 3. Where elastomers are used, they must be supported with an outer casing or sideplate. This type is commonly referred to as a double-wall type. An example of a double-wall rubber-lined pump is shown as Figure 4.

Vertical cantilever and vertical submersible centrifugal pumps are also built in significant quantities and serve mostly in general cleanup and other service. These pumps on occasion are called upon to operate as transporting devices and, depending on the design and service, include the special features described later.

CENTRIFUGAL SLURRY PUMP HYDRAULIC DESIGN

In order to reduce wear, the hydraulic design of a slurry pump should be such that the rotating wetted component and fluid velocities must be kept as low as practical while hard metal or elastomer wetted-section thicknesses are increased. Impeller meridional sections are also usually kept close to rectangular and with front sealing on a vertical face to minimize the axial rotating surfaces. Collectors (casings) are usually semi/volute (or even annular) to give better wear over a wider range of best-efficiency-point quantity (flow rate) (BEPQ).

Where larger solids are to be pumped, the inside shroud impeller widths are usually oversized, the number of vanes is reduced to two or three, and the vane overlap may be reduced. All of these distortions modify or reduce the hydraulic performance. Given the variety of types (and severity) of services, the design solution that provides the best compromise between wear life and efficiency and provides the end user with the overall lowest total cost of ownership is not simple to achieve. Although specific to a particular slurry type, Kasztejna et al. (1986) and Cooper et al. (1987) provide a good insight into

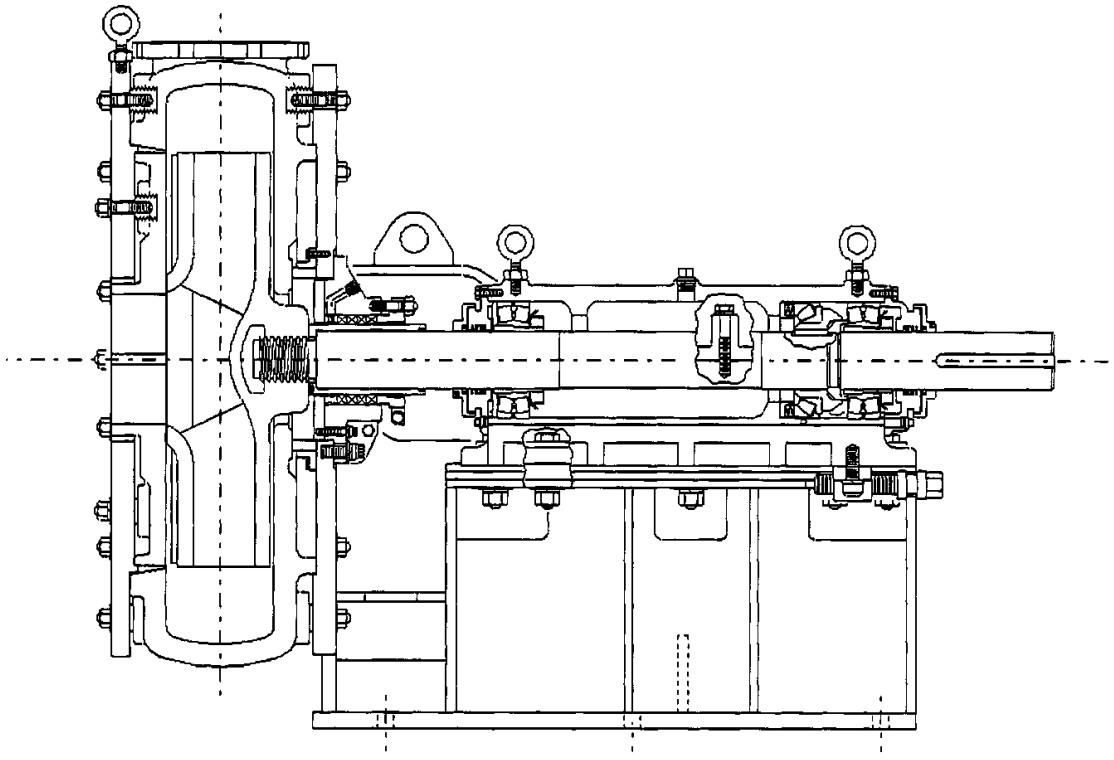


FIGURE 3 Typical single-wall hard iron slurry pump

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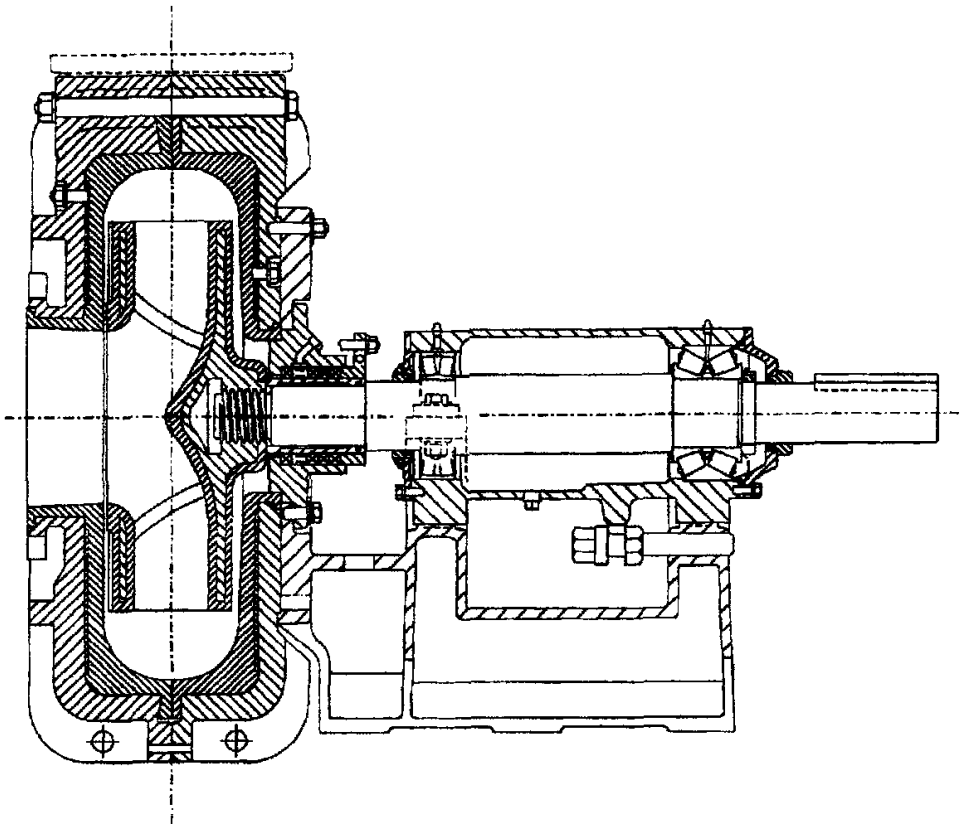


FIGURE 4 Typical double-wall elastomer-lined slurry pump

some of the design tradeoffs and special problems involved in the design of a centrifugal slurry pump.

In general, slurry pumps are of low specific speed (as defined in Section 2.1.1), in the range of 750 to 2000 for N_s (14.5 to 39 for n_q and 0.27 to 0.73 for Ω_s).They employ three to five vanes, have impeller inside shroud widths that are 75 to 200% wider than those a normal water pump, and use impeller vane and shroud thicknesses two to three times that of a water pump. The results of these distortions flatten the constant-speed head-flow curves and reduce efficiency by five or so percent. Typical head coefficients (as defined in Section 2.1.1) and efficiencies are shown in Figures 5 and 6.

In order to minimize wear over different ranges of percent of BEPQ, operation slurry pump casings vary from the true volute water pump T type (shown schematically on Figure 7) through the semivolute C type to the essentially annular A type. There are also a few examples of what is called the OB type with recessed tongue and special extended neck intended for limiting wear in the tongue area due to excessive recirculation (which itself is symptomatic of misapplication). The general applicability of each of these is shown in Figure 8, taken from Addie and Helmly (1989).

Impeller types may be roughly categorized also as being of the old-style square meridional section radial vane RV type (still common in rubber-lined pumps where it is difficult to mold a twisted vane), the most common rounded rectangular meridional section with twisted vanes ME type, and the HE type which is closer to a water pump.

Each combination of the types illustrated in Figure 7 has its own hydraulic performance and wear characteristics. The HE/T combination generally has the highest performance, but is not necessarily the most forgiving for wear, whereas the ME/C combination is capable of respectable efficiency while at the same time having more predictable wear performance. In extremely heavy-duty wear service, operation must be at low discharge flow rates—below BEPQ. The A type shell, or even the OB type, could be best for wear where a pump has been misapplied badly. For additional information on the selection of different hydraulic types, see Addie et al. (1996).

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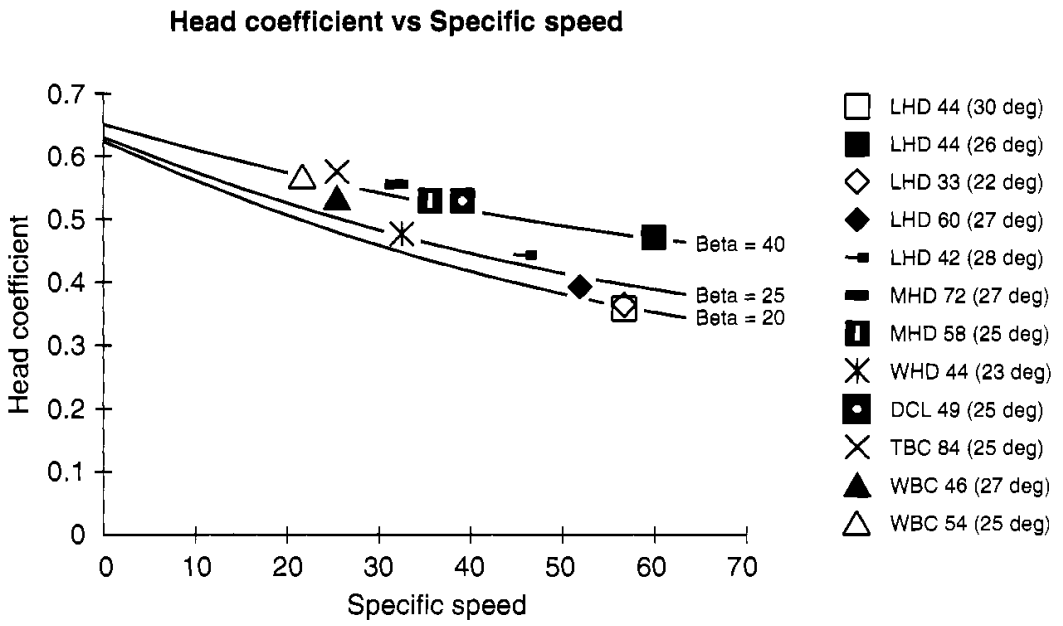


FIGURE 5 Head coefficient versus specific speed n_q, based on m³/s, m, and rpm (from Addie and Helmly, 1989) [Note: for specific speed N_s, based on gpm, ft, and rpm, multiply n_q by 51.65; for universal specific speed 1_s, divide n_q by 52.92.]

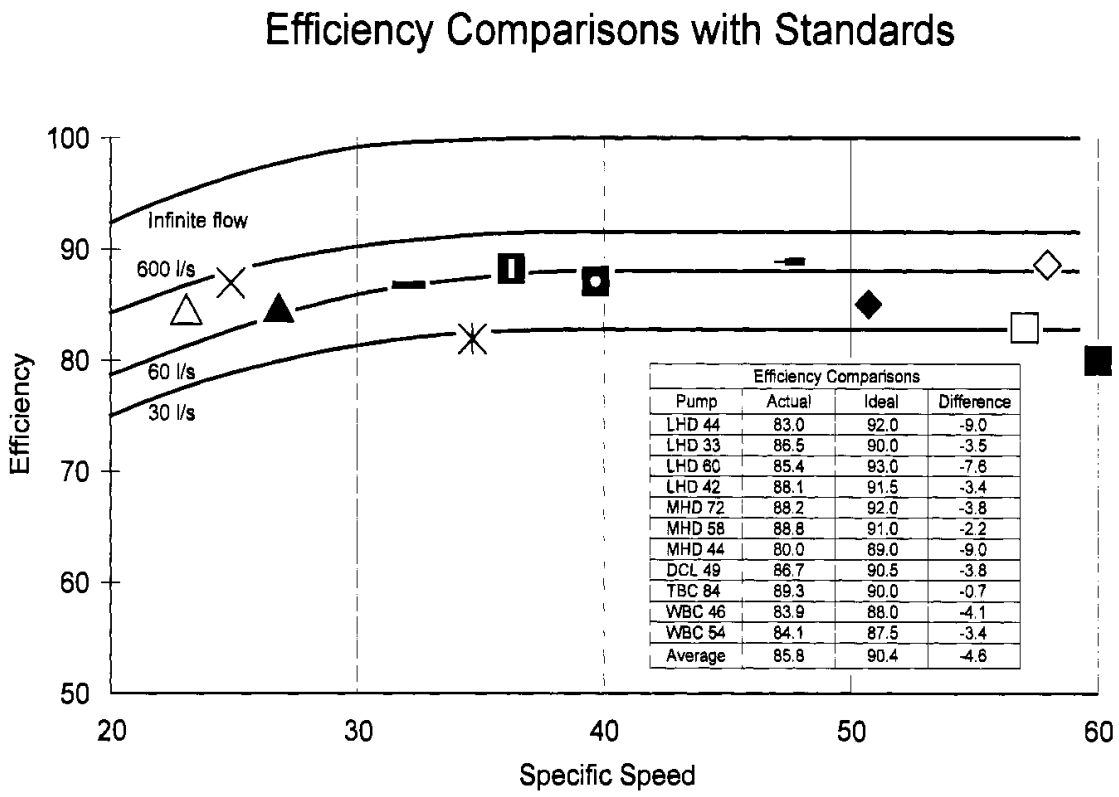


FIGURE 6 Pump efficiency as a function of specific speed n_q, based on m³/s, m, and rpm (from Addie and Helmly, 1989) [Note: for specific speed N_s, based on gpm, ft, and rpm, multiply n_q by 51.65; for universal specific speed 1_s, divide n_q by 52.92.]

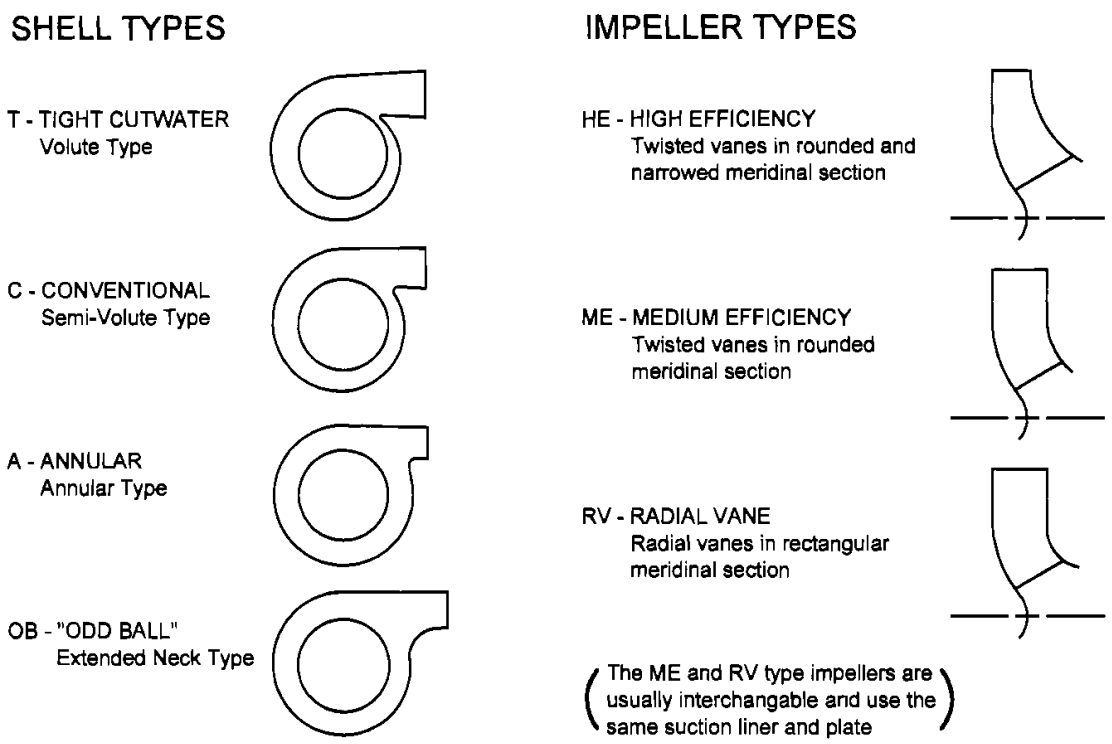


FIGURE 7 Types of shells (casings) and impellers

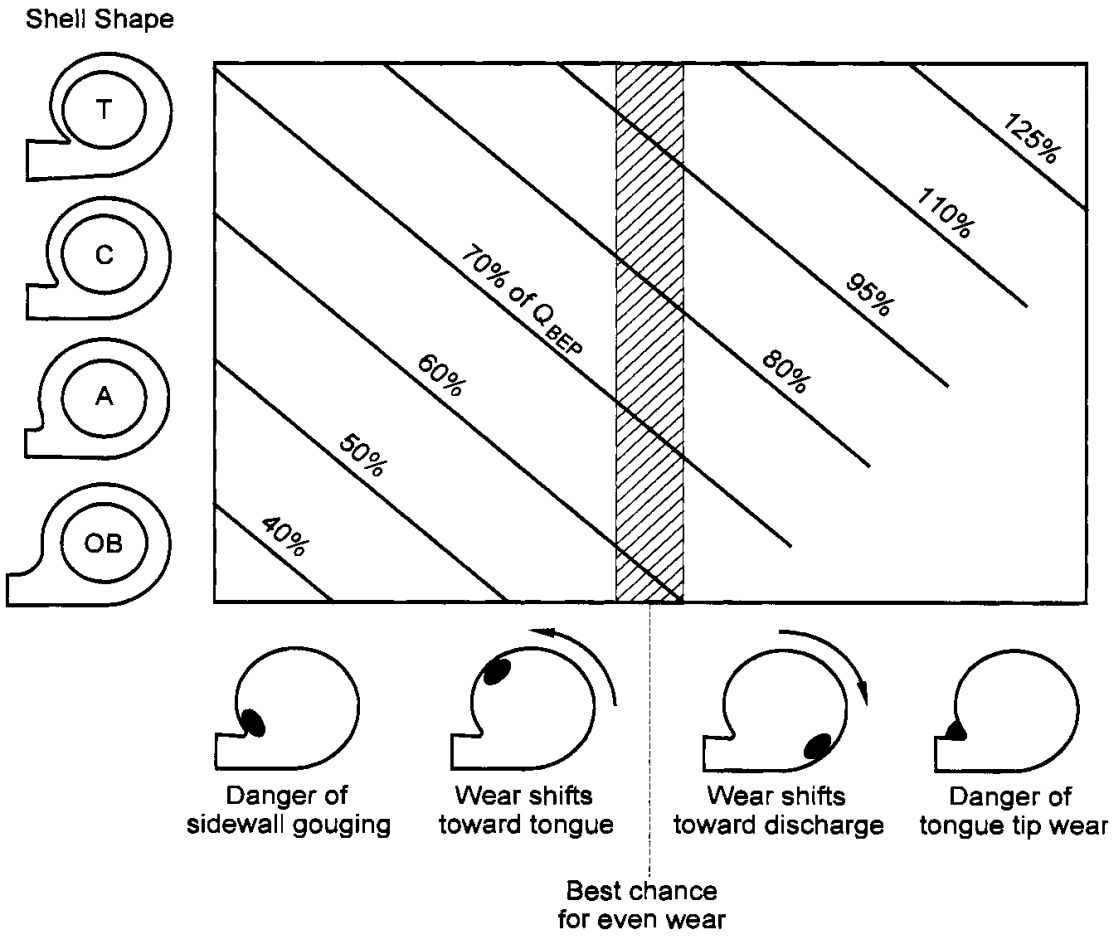


FIGURE 8 Trends in location of casing belly wear

Impeller and shells using elastomers such as rubber, neoprene, and urethane tend to be limited to impeller tip speeds less than about 75 ft/s (23 m/s), although this can rise with stiffer elastomers, at some expense to wear life. Impellers employing elastomers require higher available NPSH because of the thicker impeller vane sections needed, and this condition may limit their use, for example, in pumping flue-gas desulphurization slurries.

MECHANICAL DESIGN OF SLURRY PUMPS

Given the distortions noted earlier the mechanical design of a slurry pump is similar to that of a water pump. Slurry specific gravities (SG) of up to 1.6 and higher may have to be pumped, so the shaft and bearings of necessity must be more robust. Figures 3 and 4 show typical pump bearing housings.

Impeller attachment by an Acme-type screw has generally been found best capable of carrying the heavy loads required, and the connections can be manufactured economically in the hard materials most commonly employed in impellers. The large loads associated with heavy-duty service require roller bearings with separate roller thrust bearings. Designs with ball bearings are suited only for light-service pumps.

The absence of radial sealing in slurry pumps allows shaft deflections larger than those found in water pumps, and these may limit the life of stuffing boxes and mechanical seals. Newer and better designs, however, tend to have shorter shafts and smaller deflections, extending the seal life. A conventional packed stuffing box is still the simplest and most common rotating-assembly seal. Configurations are similar to those used for water pumps, with the lantern-ring supplying a clear-water flush to the center for minimum dilution, or to the product side for maximum life. An expeller-type seal is popular where dilution of the product is unacceptable. These seals are limited to one stage and involve additional efficiency losses, which usually range to 3% or more.

Mechanical seals are now available that take their coolant from the product and operate with no clear-water flush; they are mostly of the single partially balanced type. In some cases, where a pump may run dry, a double type of mechanical seal must be used. At present, mechanical seals are the preferred shaft sealing method for fine-particle light-duty service such as pumps for flue gas desulphurization. These seals are also fairly widely used in the aluminum industry for red-mud pumping service. A mechanical seal described in Maciejewski *et al.* (1993) is used for pumps handling tar sands tailings at pump discharge pressures up to 350 lb/in^2 (2400 kPa), with d_{50} about 120 microns and average slurry density near 0.0578 lb/in^3 (1600 kg/m^3). In this case, the process fluid is at an average temperature of about 130°F (55°C) and some 0.16 gpm (0.01 L/s) of water is used for external cooling. In this application, representative of the limits of mechanical seal technology at the present, the average seal life is about 3000 h.

As noted before, pump casings made entirely of elastomeric materials have insufficient strength to withstand the pressure loads, so it is necessary to have an outer casing of some sort, a configuration that is commonly called a double-wall design. For versatility, some manufacturers make these designs for interchangeable wetted internal shell components of hard metal, rubber, or urethane as the service warrants. For very large slurry pumps, the double-wall configuration is heavy and costly. As a cost-effective design for this case, it is worth considering tie-bolt construction using high-tensile outer side plates over a hard-metal plate and liners as shown in Figure 1.

Further, single-wall designs with the main shell casing in hard metal are simple to construct and maintain and thus are very popular. These can be used unlined or provided with liners of bonded rubber or other material. Ceramic materials are now available that are capable of several times the life of both elastomer and hard-metal components. At the moment, the cost-effectiveness of these materials is such that their use tends to be limited to areas of high wear and other selected areas of wetted surfaces.

In a shell of single-wall metal design, the wearing components must carry the water load. Wear tends to progress until leakage of the shell occurs, but the wear is almost always localized so leakage occurs before the structural integrity of the assembly is affected. Single-wall designs may easily be ribbed or thickened locally to increase strength

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and wear life in a particular area. In small slurry pumps, minimum casing thicknesses usually produce safe designs of adequate pressure rating.

SELECTION OF SLURRY PUMPS

As noted earlier, service conditions vary significantly, so when selecting a slurry pump, a recommended approach is to select a service class and limit nozzle (branch) and other velocities according to that. Figures 9 and 10 from Wilson et al. (1997) outline an approach that can be used.

For a given service class, Table 1 provides a guide to maximum impeller, nozzle (branch), and other velocities that should give reasonable wear life and keep the cost and size of the pump to a minimum.

Except in the case of dredges (which are usually diesel driven), most slurry pumps are driven by electric motors. Because of the difficulty associated with trimming impeller diameters and variations of system head, it is common for plant slurry pumps up to 250 hp (185 kW) to have V-belt drives. For larger pumps, four pole motors and gearboxes are most common, with one or more variable speed units when there are several (usually up to six) units operating in series.

SOLIDS EFFECT ON CENTRIFUGAL PUMPS

The presence of solid particles in the flow tends to produce adverse effects on pump performance, and detailed information on these effects is needed to achieve reliable and energy-efficient operation. Head-flow curves for centrifugal pumps are often rather flat. Also, the pipeline characteristic for slurry flow usually displays a slow rise with increasing discharge. As a result, the two characteristic curves often intercept at a rather shallow angle. Therefore, even a small diminution in pump head can produce a disproportionately large drop in flow rate. The resulting difficulties in operation can cause large and expensive systems to run inefficiently or not run at all.

The effects on pump characteristics are shown schematically in Figure 11, which is a definition sketch for illustrating the reduction in head and efficiency of a centrifugal pump

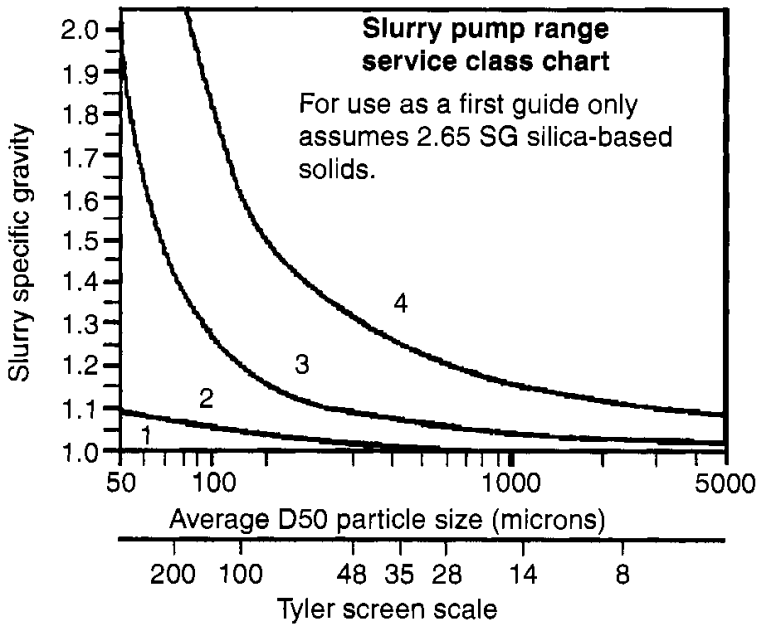


FIGURE 9 Slurry pump range service class chart

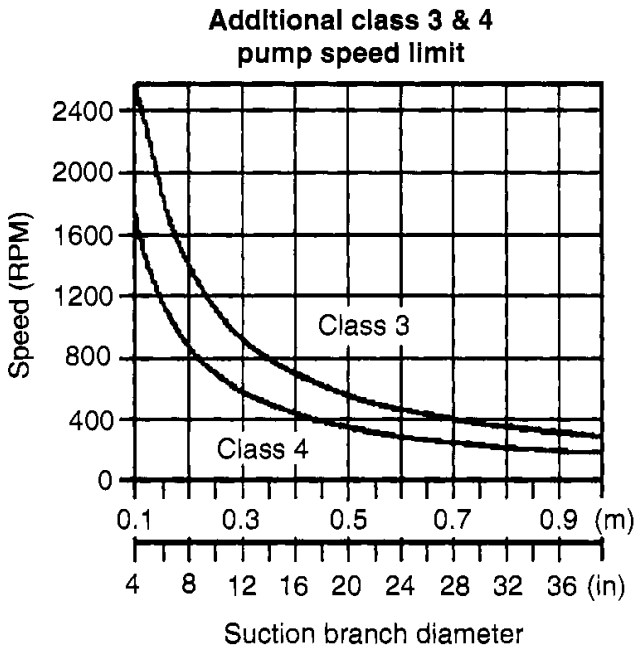


FIGURE 10 Pump speed limit chart for classes 3 and 4

operating at constant rotary speed and handling a solid-water mixture. In this sketch, and the discussion that follows, η_m represents the pump efficiency in slurry service, and η_w is the clear-water equivalent. Likewise, P_m and P_w are the power requirements for slurry service and water service, respectively. The head H_m is developed in slurry service, measured in height of slurry, whereas H_w represents the head developed in water service in height of

TABLE 1 Recommended operating limits for slurry pumps

Operating* Limits	Shell Type	Service Class 1	2	3	4
Maximum discharge (R) Velocity		40. (ft/s) 12.2 (m/s)	32. (ft/s) 9.8 (m/s)	27. (ft/s) 8.2 (m/s)	20. (ft/s) 6.1 (m/s)
Maximum throat (R) velocity		50. (ft/s) 15.2 (m/s)	40. (ft/s) 12.2 (m/s)	30. (ft/s) 9.1 (m/s)	20. (ft/s) 6.1 (m/s)
Recommended percent range (C) of BEP flowrate	Annular (A)	20–120%	30–110%	40–100%	50–90%
	Semivolute (C)	30–130%	40–120%	50–110%	60–100%
	Near Volute (T)	50–110%	60–130%	70–120%	80–110%
	Annular/Oblique Neck (OB)	10–110%	20–100%	30–90%	40–80%
All metal pump maximum impeller (R) peripheral speed		8500 (sfpm) 43.2 (m/s)	7500 (sfpm) 38.1 (m/s)	6500 (sfpm) 33.0 (m/s)	5500 (sfpm) 27.9 (m/s)
Rubber lined pump Maximum impeller (R) peripheral speed		5500 (sfpm) 27.9 (m/s)	5000 (sfpm) 25.4 (m/s)	4500 (sfpm) 22.9 (m/s)	4000 (sfpm) 20.3 (m/s)

[Note: sfpm = ft/min; for ft/s, Divide by 60]

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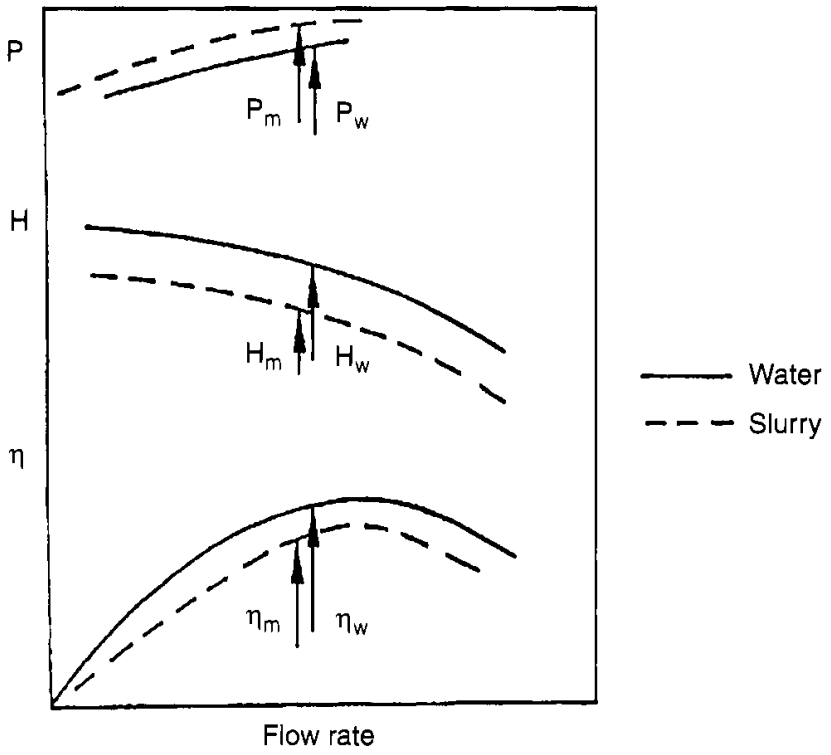


FIGURE 11 Effect of slurry on pump characteristics (schematic)

water. The head ratio H_r and the efficiency ratio η_r are defined as H_m/H_w and η_m/η_w respectively. The fractional reduction in head (the head reduction factor) is denoted by R_H and defined as $1 - H_r$; for efficiency, the fractional reduction (efficiency reduction factor) is R_η, given by $1 - \eta_r$.

Test loop results for a large variety of solids at the GIW Hydraulic Testing Laboratory have been put together in a generalized design diagram (Figure 12) for larger heavy-duty pumps and smaller pumps having impeller diameters of 8 in (0.2 m) to 16 in (0.4 m). The diagram gives R_H and H_r in terms of pump impeller diameter (D) and solid average size (d_{50}) at a solids concentration by volume (C_v) of 0.15 (15%) with a solids density ratio 2.65 and a negligible amount of fine particles ($X_h = 0$). For example, a pump with an impeller diameter of 32 in (0.81 m), R_H becomes 14% ($H_r = 0.86$) for a d_{50} of 0.39 in (10 mm).

Corrections for various values of C_0, S_s, and X_h will now be given together with estimations of the reduction in efficiency, R_η. Based on experimental results discussed in Wilson et al. (1997), the R_H values obtained from Figure 12 may be multiplied by the following factors when the solids density ratio and the fine particle content are different from 2.65 and zero, respectively.

solids density ratio, S_s: $$\left[\frac{S_s - 1}{1.65}\right]^{0.65}$$

fine particle content, X_h: $$(1 - X_h)^2$$

For example, with $S_s = 4.4$ and $X_h = 0.08$ (8%), the factors become 1.6 and 0.85, respectively. The pump previously mentioned with an impeller diameter of 36 in (0.91 m) had R_H of 14%. This value is now corrected to give

$$R_H = 14\ (1.6)\ 0.85 = 19\%$$

R_H can normally be related linearly to C_v, for cases with C_v less than 15 to 20%. Thus with $C_v = 6\%$ in the previous example, R_H will be 19 (6/15) = 8% because the reference value for C_v in Figure 12 is 15%.

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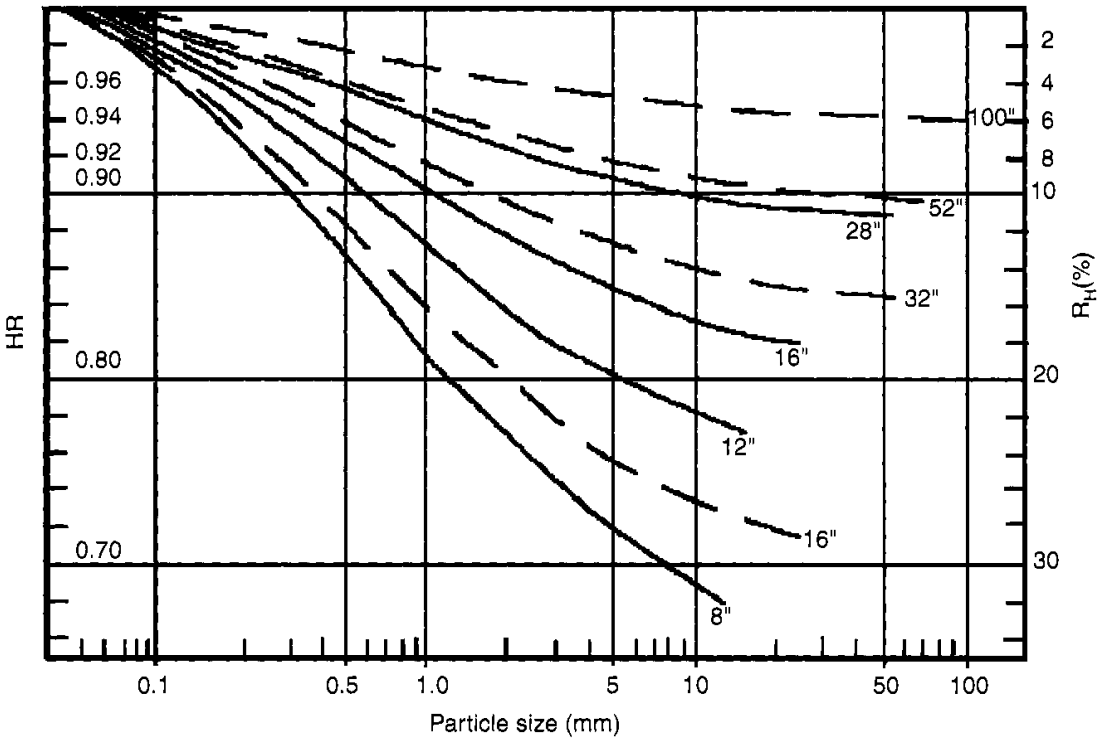


FIGURE 12 Effects of particle size (d_{50} mm) and impeller diameter (D in) on H_r and R_H. For solids concentration by volume, C = 15% with solids density ratio, S_s = 2.65 and a negligible amount of fine particles (X_b = 0). Solid lines represent smaller newer design pumps while dashed lines represent conventional heavy-duty pumps. Adapted from GIW Industries Inc., U.S.A. [Note: for particle size in inches, multiply by 0.039; for impeller diameters in mm, multiply inches by 25.4.]

For the larger pumps in Figure 12, the reduction in efficiency, R_η, is normally less than R_H. For the smaller pumps, with $D < 16$ in (0.4 m), R_η usually equals R_H; however, it may be sensitive to solids properties. Independent of the pump type in Figure 12, R_η may exceed R_H if C_v exceeds about 20%, indicated on Figure 13.

In opposition to these tendencies, there have also been investigations where the reduction in efficiency remains small at very high concentrations, specifically with broad particle-size distributions.

With equivalent reductions, $R_H = R_\eta$, the power consumption increases directly with the relative slurry density.

$$P_m = S_m \cdot P_w \tag{1}$$

with $R_\eta < R_H (H_r > \eta_r)$, Eq. 1 becomes

$$P_m > S_m \cdot P_w$$

that is, the power consumption becomes larger than that given by Eq. 1.

Finally, the calculated reductions may be underestimated if the solid particles are coarse and very angular. For example, the solids used in the preceding examples consisted of crushed angular particles and actually gave reductions in head and efficiency well over 10% (Sellgren et al., 1997), compared to R_H = 8% calculated here based on Figure 12.

On the other hand, with very fine particles, or large X_h-values and high solids concentrations, most slurries are practically nonsettling. In this case, the solid's effect is mainly related to the rheological behavior. When pumping nonsettling slurries in industrial applications, the slurry normally behaves in a non-Newtonian way, giving widely varied pump performance effects. In general, however, small pumps are affected more than large units when pumping highly viscous or non-Newtonian media. Furthermore, the influence on the efficiency is normally larger than on the head.

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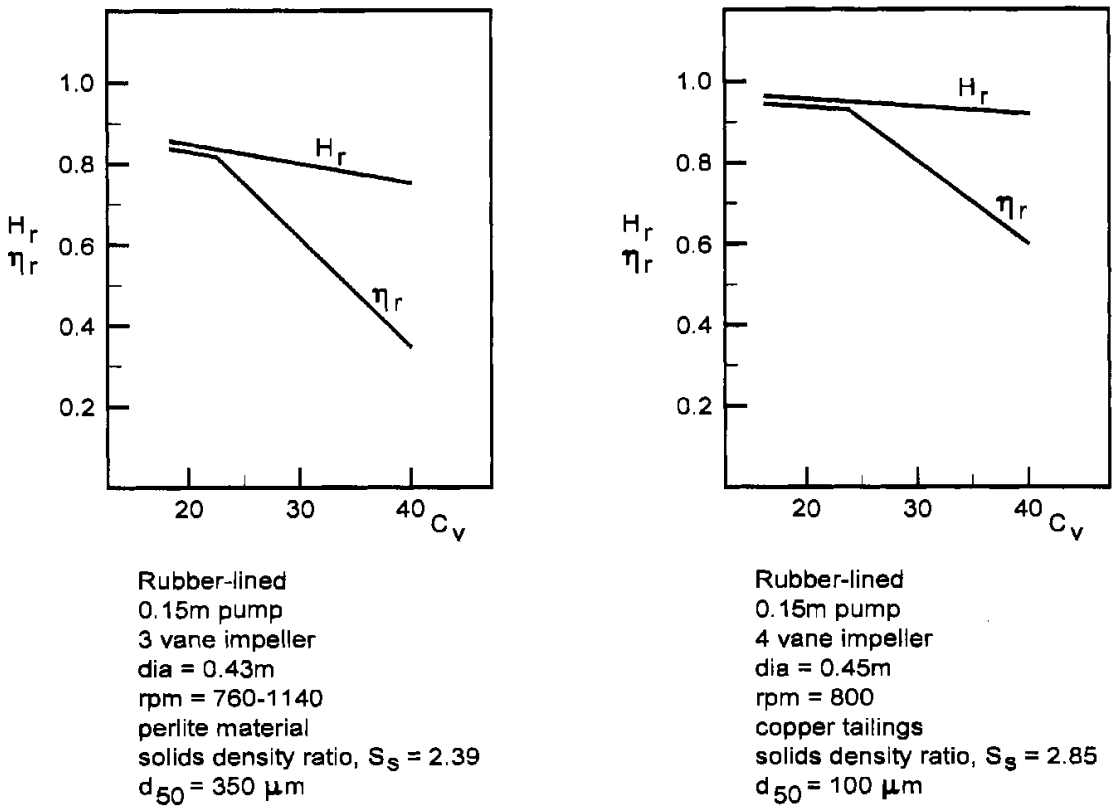


FIGURE 13 Head and efficiency reduction curves for two products (from Sellgren and Vappling, 1986) [Note: for dimensions in inches, divide mm by 25.4 and multiply m by 39.36.]

Occasionally, highly non-Newtonian slurries may be pumped by centrifugal pumps operating at flow rates much lower than the maximum-efficiency value Q_{BEP}. This may cause a dramatic drop in head, which creates an unstable head curve, seen, for example, in Figure 14.

WEAR MECHANISMS AND MATERIALS

As noted earlier, wear in a slurry pump can be severe. It is not possible, therefore, to use normal cast iron or other materials. Ideally, the life of components should be a year or more. In practice, even with the best design and materials, it can sometimes be as low as a month. Understanding wear and using the correct materials is key in the design of a slurry pump. The major erosive mechanisms are sliding abrasion and particle impact. The sliding-abrasion mode of wear typically involves a bed of contact-load particles bearing against a surface and moving tangentially to it, as illustrated on Figure 15. In pipelines, the stress normal to the surface is caused by gravity. For sliding abrasion, the erosion rate depends on the properties of particles and wear surfaces, the normal stress, and the relative velocity.

The normal stress is enhanced when the flow streamlines are curved, as in a pump casing. In this case there is a centrifugal acceleration, equal to u^2/r, where u is the local velocity and r is the radius of curvature of the streamlines. This acceleration can often be much greater than that of gravity, producing a commensurate increase in the normal stress between the moving contact-load solids and the wall material, and hence a greatly increased rate of sliding-abrasion wear. This type of behavior can cause a pump casing to wear through when adjacent straight-pipe sections of the same material have hardly been affected by erosion. Near the tongue of the pump casing, however, the other major wear type is more significant.

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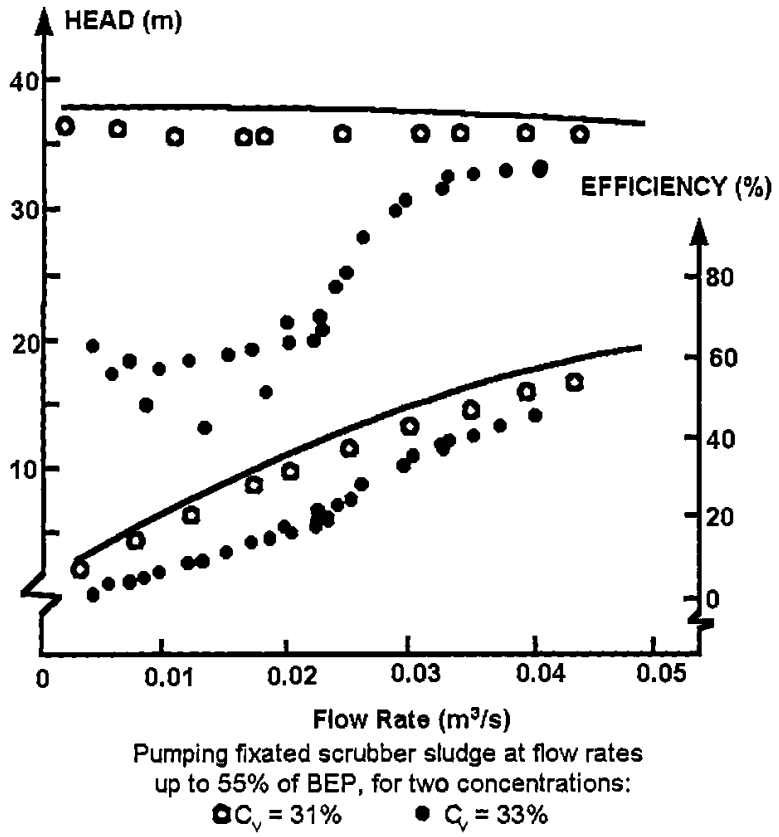


FIGURE 14 Effect of highly non-Newtonian slurries on pump head and efficiency (from Sellgren et al., 1997) [Note: for flow rate in gpm, multiply m^3/s by 15,850; for head in ft, multiply m by 3.28.]

The second type of wear is the particle-impact mode, which occurs where individual particles strike the wearing surface at an angle, despite the fact that the fluid component of the slurry is moving along the surface (Figure 16). Removal of material over time occurs through small-scale deformation, cutting, fatigue cracking, or a combination of these, and thus depends on the properties of both the wearing surface and the particles. Ductile materials tend to exhibit erosion primarily by deformation and cutting, with the specific type depending on the angularity of the eroding particles. Brittle or hardened materials tend to exhibit fatigue-cracking erosion under repeated particle impacts. For a given slurry, the erosion rate depends on properties of the wearing surface: hardness, ductility, toughness, and microstructure. The mean impact velocity and mean angle of impact of the solids are also important variables, as are particle characteristics such as size, shape, and hardness, and the concentration of solids near the surface.

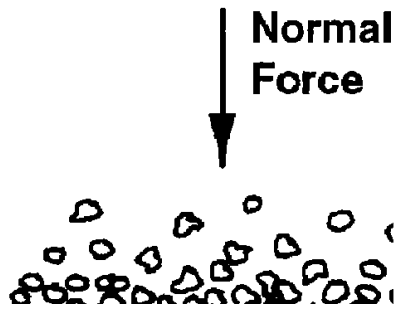


FIGURE 15 Erosion by sliding abrasion

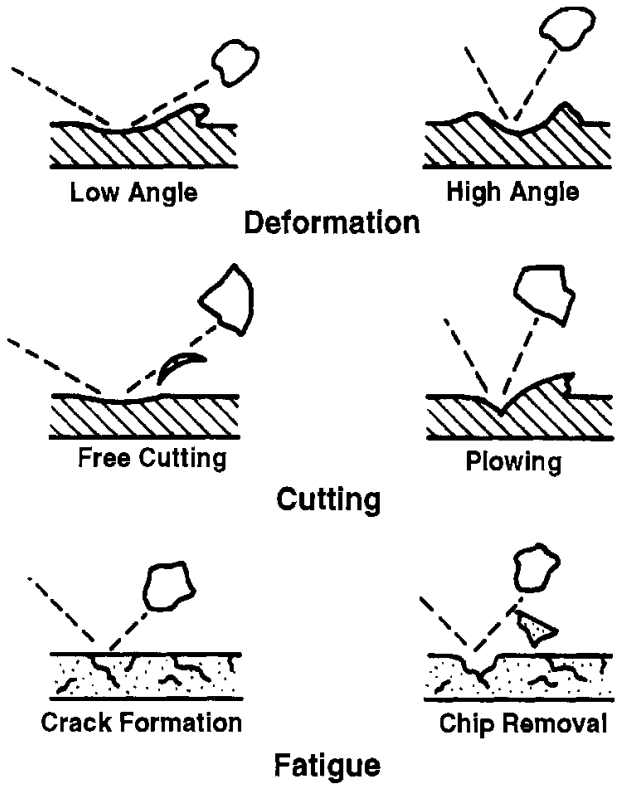


FIGURE 16 Mechanisms of particle-impact erosion

As wear mechanisms vary considerably, so also do the engineering materials that resist them. A survey of all the available wear-resistant materials and their applications would provide subject matter for a volume in itself. Nevertheless, there are a few groups of materials that have found widespread successful application against erosion in slurry systems, and thus merit particular attention. These materials fall into the broad categories of hardened metals, elastomers (rubbers and urethanes), and ceramics. Materials in each of these categories have their advantages and disadvantages, and the final choice of a material must account for wear resistance, strength, ease of maintenance, direct cost, and indirect costs such as plant downtime for maintenance, or failure, safety of operation, and effect on system efficiency.

Within the metals group, superior wear resistance is given by the high-alloy white cast irons, particularly nickel-chrome (e.g., NiHard), chromemoly (e.g., 15% chromium and 2% molybdenum), and high chrome (e.g., 27% chromium) alloys. These materials exhibit higher hardness (600 plus Brinell), but less toughness than steel alloys of similar tensile strength. The high carbon content and resultant massive carbides present in these metals gives them excellent resistance to all forms of particle erosion, especially in sliding or impact at low angles (cases often encountered in slurry applications). For the coarsest slurries, hard metal is common, principally the so-called NiHard, high-chrome, or chrome-molybdenum alloys of 600 to 700 Brinell hardness.

For slurries without coarse particles, elastomers such as rubber, neoprene, and urethane tend to wear better than hard metal, especially for liners and shells. This improvement applies only if there is no "tramp" (that is, extraneous material such as tools, bolts, or pieces of broken castings). Elastomer selection may also be determined by the corrosive

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characteristics of the slurry. Natural soft rubber remains an excellent and economical pump lining material to handle fine abrasive and corrosive slurries. Carbon black can be added for additional strength, hardness, and tear resistance, to better withstand the impact of large particles in the slurry. Like all other materials, rubber has limitations that should be considered in the selection process. Wear rates in the presence of coarse material or high tip speeds may make it uneconomical, and tramp material or adverse suction conditions can tear the rubber. High temperature or the presence of oils or chemicals may require the use of one of the higher-cost synthetic materials. For example, neoprene has been found to tolerate higher temperatures and higher impeller tip speeds than can be used with natural rubber.

CALCULATION OF SLURRY PUMP WEAR

The general selection rules noted earlier aim at keeping wear to generally acceptable levels, but are otherwise nonquantitative. Finite element and other methods such as those described by Addie et al. (1996), Bross (2001), or Pagalthivarthi (1991) are now available. They allow particle velocities inside the impeller and in the suction liner nose face area adjacent to the impeller of a pump to be calculated, which, using material wear tests coefficients as described by Pagalthivarthi (1990), can be used to determine wear rates and component lives.

For example, numerical simulations allow calculation of wear around the periphery of a given pump shell for a given set of conditions. Figure 17 is an example of how wear varies with different flows from Roco et al (1983). Addie et al. (1987) shows also how a particular collector can be altered and optimized for the best possible wear.

Although all of these methods cannot predict three-dimensional turbulent wear, they are now opening up the possibility of better designs, better selection, and the calculation and evaluation of the total cost of ownership, Addie et al (1998), and the benefits that could come from that.

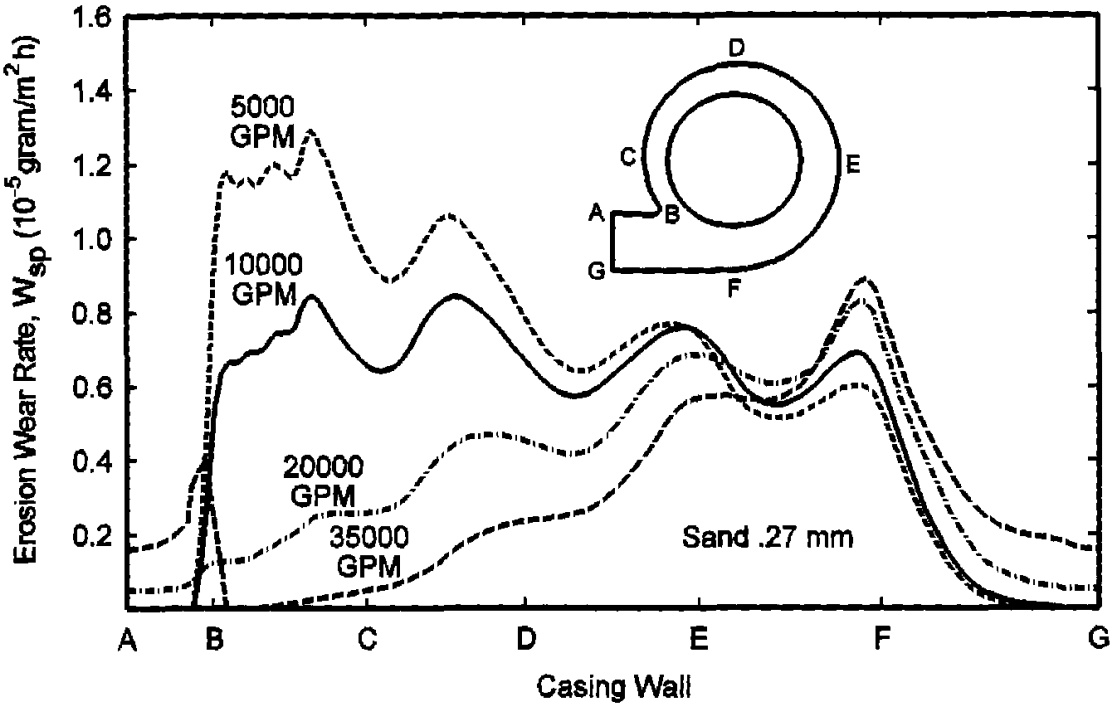


FIGURE 17 Erosion wear distributions calculated at different flow rates [Note: for flow in L/m, multiply gpm by 3.79; for particle size in inches, divide mm by 25.4.]

REFERENCES

Addie, G. R., and Helmly, F. W. (1989). Recent Improvements in Dredge Pump Efficiencies and Suction Performances. *Proc. CEDA Dredging Day,* Amsterdam.

Addie, Graeme R., Pagalthivarthi, K. V., and Visintainer, R. J. (1996). Centrifugal Slurry Pump Wear, Technology, and Field Experience. *Proc. ASME Fluids Meeting,* San Diego, CA, pp. 703–716.

Addie, Graeme R., and Carlson, R. R. (1998). Tools and Technology for Selecting Slurry Pumps. *CIM Montreal '98.*

Addie, G. R., Visintainer, R. J., and Roco, M. C. (1987). Experiences with a Numerical Method of Calculating Slurry Pump Casing Wear. *Proc. 4th International Pump Symposium and Short Courses.*

Cooper, Paul, et al. (1987). Centrifugal Slurry Pump Wear and Hydraulic Studies. Report, Ingersoll-Rand Company for U.S. Department of Energy Office of Fossil Energy and Pittsburgh Energy Technology Center. DE-AC22–82PC50035.

Kasztejna, P. J., et al. (1986). Hydraulic Development of Centrifugal Pumps for Coal Slurry Service. *Proc. 8th International Symposium on Coal Slurry Fuels Preparation and Utilization,* pp. 646–665.

Maciejewski, et al. (1993). Transport of coarse rock with sand and clay slurries, *Proc. Hydrotransport 12,* BHRA Fluid Engineering, Cranfield, UK, pp. 705–741.

Pagalthivarthi, K. V., and Helmly, H. W. (1990). Application of Materials Wear Testing to Solids Transport via Centrifugal Slurry Pumps. *Wear Testing of Advanced Materials, ASTM STP 1167,* Divakar and Blau, eds. American Society for Testing Materials, Philadelphia, PA.

Pagalthivarthi, K. V., Desai, P. V., and Addie, G. R. (1991). Quasi-3D Computation of Turbulent Flow in Centrifugal Pump Casings. *Proc. ASME Symposium on Multidisciplinary Application of CFD.*

Roco, M. C., and Addie, G. R. (1983). Analytical Model and Experimental Studies on Slurry Flow and Erosion in Pump Casings. *Proc. 8th International Conf. Technology.* Slurry Transportation Association, p. 263.

Sellgren, A., and Addie, G. (1993). Solids effects on the characteristics of centrifugal slurry pumps. *Proc. 12th International Conference on the Hydraulic Transport of Solids in Pipes,* Brugge, Belgium, pp. 3–18.

Sellgren A., and Addie, G. (1996). Pump and pipeline solids effects of transporting sands with different size distributions and concentrations. *Proc. 13th International Conference on the Hydraulic Transport of Solids in Pipes,* Johannesburg, South Africa, pp. 227–236.

Sellgren, A., and Vappling, L. (1986). Effects of Highly Concentrated Slurries on the Performance of Centrifugal Pumps. *Proc. International Symposium on Slurry Flows,* FED Vol. 38, ASME, USA, pp. 143–148, 1986.

Sellgren, A., Addie, G., and Wei-chung, Y. (1997). Effect of Coarse and Heavy Solid Particles on Centrifugal Pumps Head and Efficiency. *47th Canadian Chemical Engineering Conference,* Edmonton, AB.

Wilson, K. C., Addie, G. R., Sellgren, A., and Clift R. (1997). *Slurry Transport Using Centrifugal Pump,* 2nd edition. Blackie Academic and Professional, London.

SECTION 4.3
CONSTRUCTION OF SOLIDS-HANDLING DISPLACEMENT PUMPS

WILL SMITH

APPLICATIONS

Positive displacement pumps are especially suited for many solids-pumping applications because constant high pressures and good efficiency are achieved over the full range of pump capacities. Low relative velocities between abrasive liquid-solids mixtures and the pump parts minimize erosion. With centrifugal pumps, an increase in system resistance, such as a flow blockage caused by settled solids or the apparent viscosity increase characteristic of many slurries due to momentary capacity reduction, results in a self-defeating reduction in pump flow rate.

Positive displacement pumps are popular for a number of solids-handling applications. Each application presents peculiar demands for pump features.

Solids Transport Large-capacity positive displacement pumps at moderate pressures are used to pump coal and ores over relatively long distances. Solids for transport are usually slurried with water and are pumped at ambient temperatures. Double-acting horizontal piston pumps with large, "mud pump" valves and piston rings (Figure 1) typify transport service.

Most pipelines have multiple pumping stations at intervals along the route, dictated by topography and by pipeline and pumping station first and operating cost balances. Centrifugal booster pumps are employed at the first station to deliver the prepared slurry to the displacement pumps at a sufficient pressure to satisfy their NPSH, including acceleration head, requirement. Subsequent stations are located at points where there is sufficient residual pipeline pressure to fulfill the displacement pump's suction requirements. The capacity of each station is adjusted by pump speed control so the next station's inlet pressure is kept relatively constant.

Most stations muse multiple positive displacement pumps in parallel, with at least one standby pump. Capacity modulation requires but one of the pumps to be operating with

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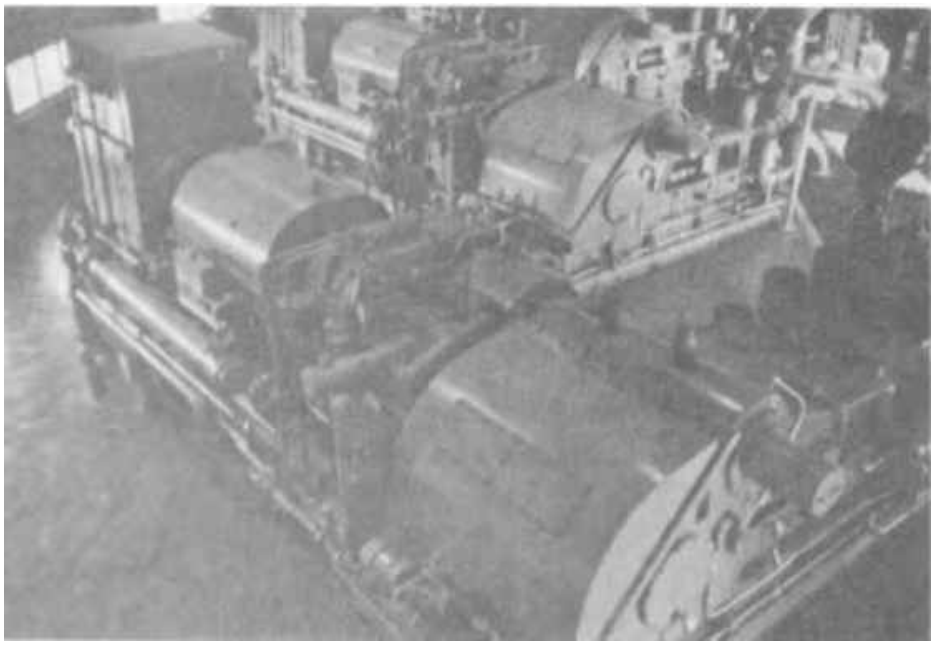


FIGURE 1 Horizontal double-acting duplex coal slurry pipeline pumps (Black Mesa Pipeline, Oil Well Div. of U.S. Steel)

speed control, although all are generally so fitted. Unlike centrifugal pumps, displacement pumps can provide full-rated pressures at all speed-controlled capacities.

Process Pumping Some chemical and petroleum processes require pumping of solids to high pressures for process reactions. Typical of these are bauxite ore in hot caustic for alumina plants, ground coal in water or coal liquids for synthetic fuel production, and grain in water for alcohol production (Figure 2).

Process streams involve a vast variety of liquids and solids. Concentrations may be as high as 70% solids by weight. Temperatures can reach 800°F (427°C). Pressures from several hundred to many thousands of pounds per square inch are accommodated.

Other Specialty Services A number of specialty services employ positive displacement pumps to handle solids. Some are relatively simple applications of standard catalog pumps, and others are single-purpose developments and thus have evolved with unique characteristics.

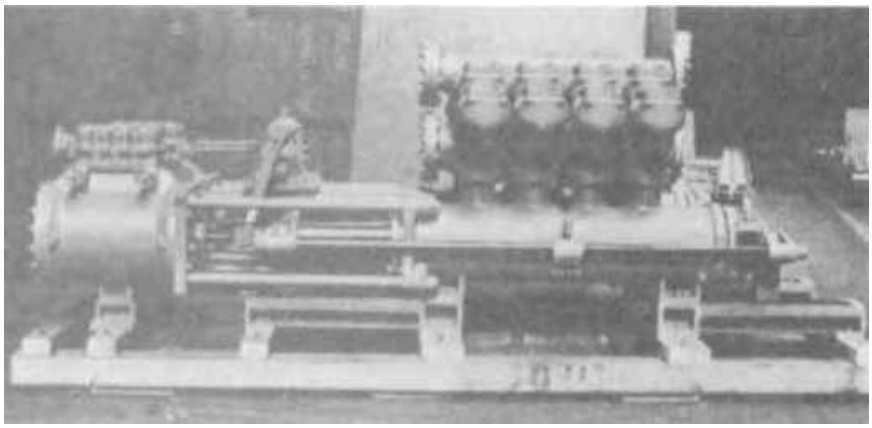


FIGURE 2 Horizontal double-acting plunger pump for slurry of grain mash and water for alcohol production (Flowserve Corporation)

Mud pumps are used to inject drilling mud, which transports cuttings out of wells and lubricates the drill bit during well drilling. Because of the limited duration of a drilling project, the life of expendable parts (packing, valves, rods) is compromised in favor of size and weight for portability and ease of overhaul. *Sludge disposal pumps* are common in sewerage treatment plants, and *tailings pumps* move solids out of mines.

PUMP CONSTRUCTION

Solids-handling displacement pumps differ from ordinary displacement pumps in the means used to alleviate the deleterious effects of the solids on the packing, the displacement element (whether plunger, piston, or diaphragm), and the suction and discharge check valves. To protect parts from the ravages of the solids, (1) particles are prevented from entering close clearances, (2) operation is at lower speeds to reduce the effects of erosion and abrasion, or (3) the susceptible parts are made of wear-resistant materials. Because sacrificial wear parts must be replaced more often than in clear liquid pumps, special design attention for ease of replacement of these parts is justified.

The designer's choice is between hard materials that resist wear and resilient materials that may accommodate the particles of solids. This choice is often limited by other factors, such as temperature and chemical compatibilities. Hard materials used include

Tungsten carbide

Ceramics—chrome oxide and aluminum oxide

Hardenable stainless steel

Cobalt-nickel alloys

Some common soft materials are

Synthetic elastomers

Softer metals (sacrificial)

Unlike centrifugal pumps, positive displacement pumps are more prone to abrasive than erosive wear. This is due to the much lower relative velocities between the liquid and the pump parts. Centrifugal pumps must generate high relative velocities, typically greater than 100 ft/s (30 m/s), for the dynamic specific energy that is converted to discharge pressure. Positive displacement pump velocities are kept low, limited usually by the settling velocity of the slurry, which is on the order of less than 10 ft/s (3 m/s).

There are four types of positive displacement solids-handling pumps:

1. Reciprocating piston pumps
2. Reciprocating plunger pumps
3. Diaphragm pumps
4. Hydraulic displacement pumps

Valves Common to all types are suction and discharge check valves, the purpose of which is to allow the solids-laden liquid to flow into the pump from the suction line and into the discharge line from the pumping chamber while preventing backflow. The features of slurry valves include

1. Large areas for low flow velocity
2. Smooth, unobstructed passages to avoid trapping and build-up of solids
3. Special designs to enhance sealing against pressure in the closed position
4. Special materials to minimize wear

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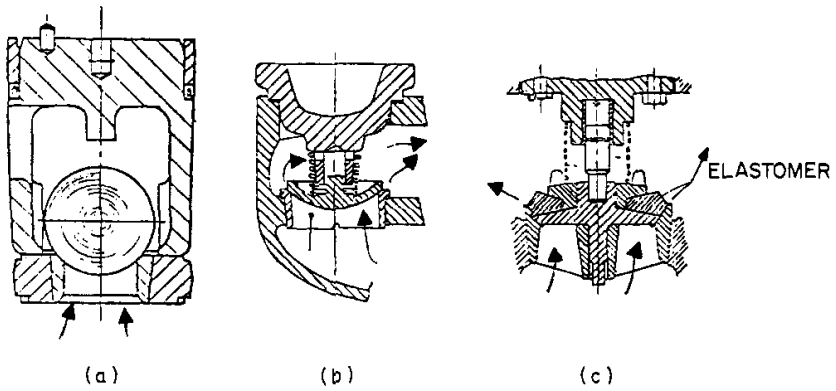


FIGURE 3A through C Three types of slurry valves: (a) free-floating ball valve, (b) spring-loaded spherical (Rollo) valve, (c) spring-loaded elastomer-seal (mud) valve

Figure 3 illustrates some valves developed particularly for solids-handling pumps. Which valve is selected for a particular application depends on the abrasiveness, viscosity, temperature, solids size, and uniformity of the slurry. It is important that the valve lift be sufficient to pass the largest particles of solids expected. Adequate sealing in the closed position is necessary to avoid excessive back leakage, which detracts from the volumetric efficiency of the pump.

Valve materials must meet temperature, corrosion, abrasion, and mechanical strength requirements. Materials for valves and seats range from cast iron and bronze through hard-enable stainless steels to solid tungsten carbide. The cost of replacement or refurbishment over the expected life, including labor and downtime, should be balanced against first cost.

Packing Protection A distinguishing feature of solids-handling displacement pumps is the means used to avoid or minimize packing wear. Five different approaches are

1. Use of conventional designs with special materials and operation at less demanding conditions, such as reduced speeds
2. Provision of a clean nonslurry environment for the packing rubbing surfaces
3. Separation of the slurry from the displacement element
4. Separation of the slurry from the pumping element
5. Total isolation of the slurry from the packing

CONVENTIONAL DESIGN WITH MODIFIED OPERATION Piston pumps usually fall into this category. By using larger pumps at speeds that are lower than those normally found with clear fluids, velocity-sensitive wear rates are reduced. *Special piston rings* (Figure 4) are used

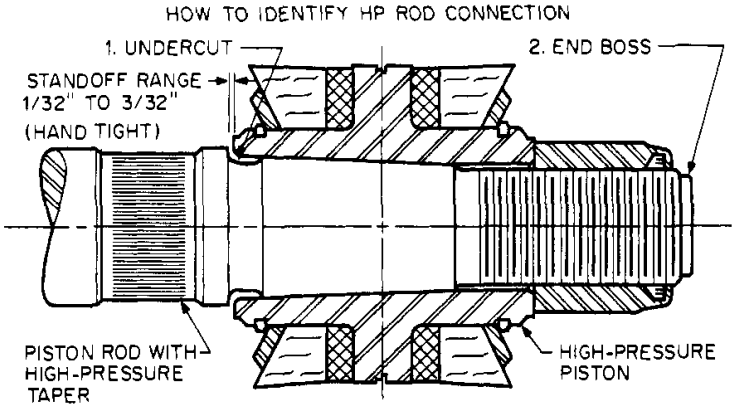


FIGURE 4 Double-acting mud pump piston. Standoff range varies between 0.030 to 0.090 in (0.75 to 2.25 mm).

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that tend to resist abrasion because they are made of zero clearance elastomeric materials and designed to scrape clean the cylinder wall. Such piston packing is a normal wear item and must be replaced at regular intervals.

CLEAN NONSLURRY ENVIRONMENT This is accomplished by the injection of clear liquid (Figure 5). Such injection may be either *continuous* or *synchronized* with the stroke of the plunger or piston. When effectively accomplished, flush injection presents essentially clear pumpage to the packing so it attains normal clear liquid packing life. Flush liquid that enters the pumping chamber dilutes the pumpage and decreases the suction volumetric efficiency.

Clear, compatible flush liquid may be scarce or expensive. Therefore special stuffing box plus flush injection systems have been developed. Required flush rates are a function of pump type and size and system employed. Satisfactory operation has been attained with flush rates of about 2 to 5% of pumped flow and even less for very large pumps. If other than a positive displacement flush pump is used, flush flow will increase as packing and throat bushings wear.

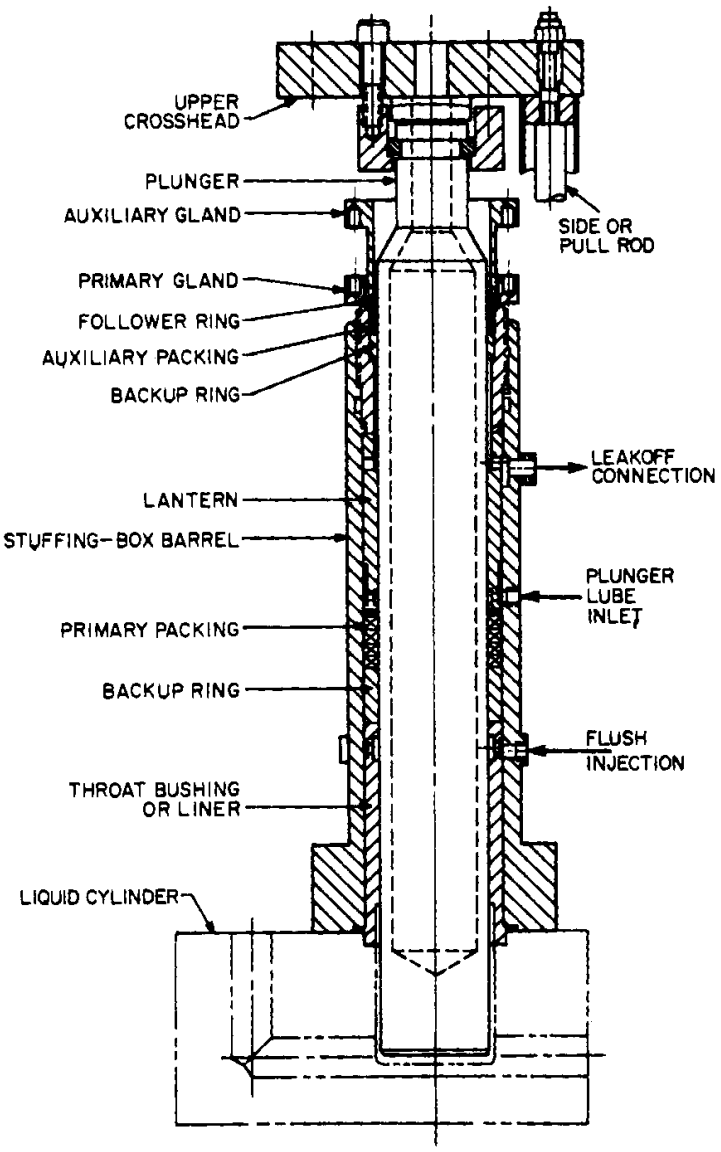


FIGURE 5 Typical flush injection

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Flushing prevents incursion of slurry particles by providing a liquid velocity counter to the unwanted slurry flow in clearances and by washing solids particles from the plunger or rod surfaces as they move toward the packing. Injection systems may be as simple as a constant-pressure liquid fed to a point between the packing and the throat bushing or as complex as a sophisticated synchronized injection system involving an additional positive displacement pump driven directly from the slurry pump drive train and independently furnishing a definite quantity of flush fluid to each plunger or piston during each pump stroke (Figure 6).

Combinations of positive, independent, synchronized pumped injection, and auxiliary buffer liquids have been proposed for very special situations.

SEPARATION OF SLURRY FROM DISPLACEMENT ELEMENT One means of reducing the concentration and temperature of slurry at the pump packing is the use of a liquid, or surge, leg (Figure 7). The displacement element—plunger or piston—is separated from the suction and discharge valves by a length of pipe that is filled with clear liquid. This liquid leg communicates with the valve chambers such that, as the liquid ebbs and flows in response to the motion of the displacement element, it acts as a liquid piston, inducing and then expelling slurry through the suction and discharge valves. Make-up liquid is injected into the surge leg, preferably through a throat bushing isolating the displacement chamber from the packing so there is theoretical flow away from the stuffing box. The surge leg thus offers the packing some degree of protection from the solids and from thermal effects.

Surge legs are, however, subject to the complete span of suction-to-discharge pressure change on each stroke. Therefore they must be designed for endurance fatigue loading. When high-temperature isolation is involved, thermal expansion and contraction must also be considered. Although surge leg pumps have been demonstrated to increase piston or plunger packing life, they do not totally prevent solids migration from the pumping

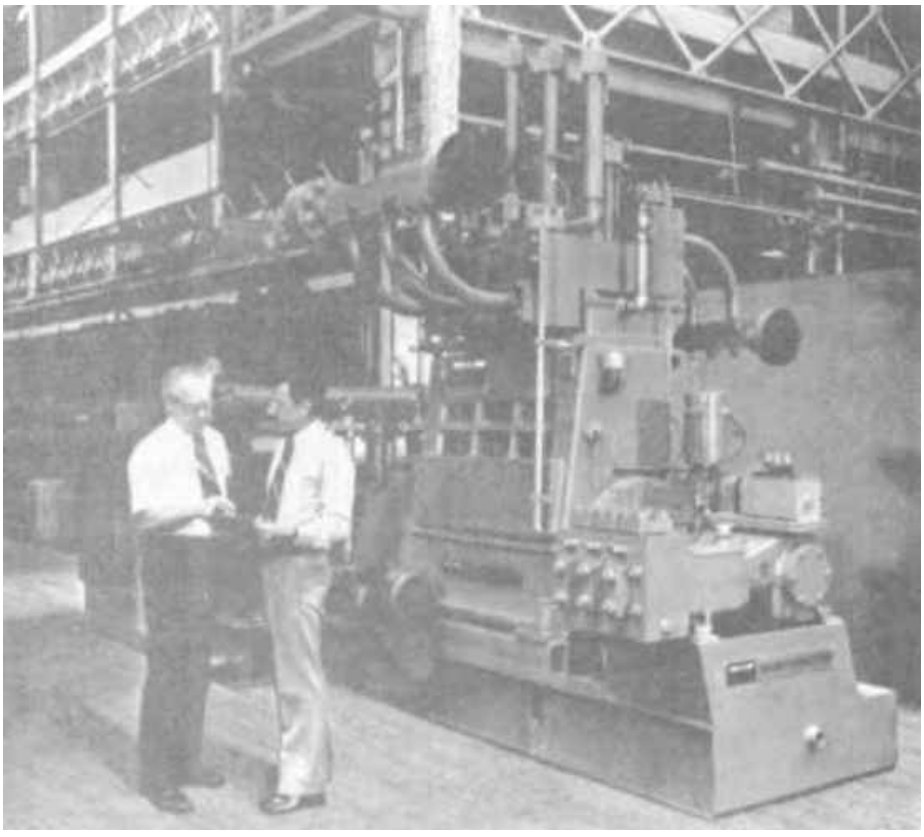


FIGURE 6 Vertical single-acting triplex plunger pump with synchronized flush liquid injection (Flowserve Corporation)

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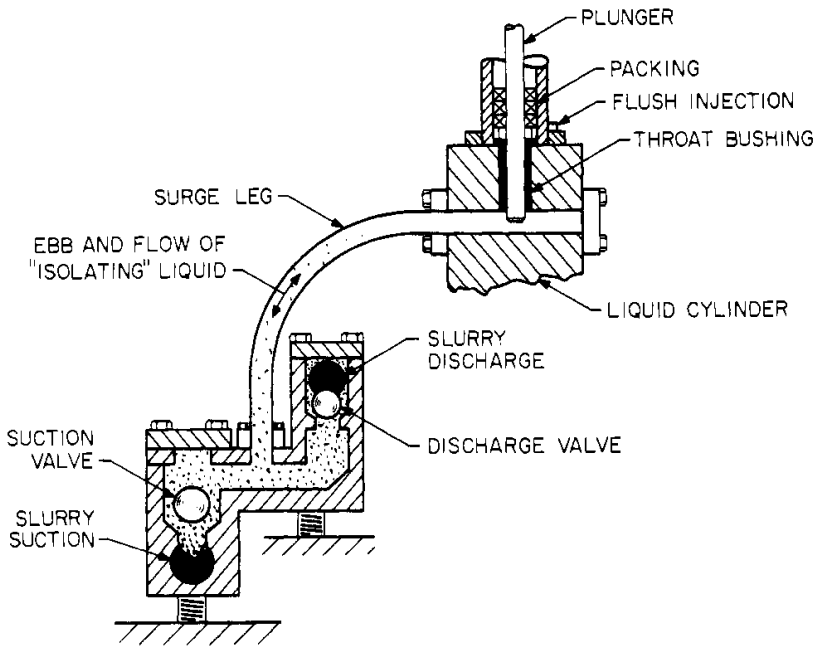


FIGURE 7 Surge leg plunger pump

chamber. Therefore special construction materials, flush injection, and reduced speeds are still appropriate.

SEPARATION OF SLURRY FROM PUMPING ELEMENT A special case of the liquid leg pump is the large, low-cycle chamber feeder (Figure 8). Although these units may or may not properly qualify as positive displacement pumps (because the motivating liquid may be delivered by a centrifugal pump), they do nevertheless have much in common with surge leg pumps.

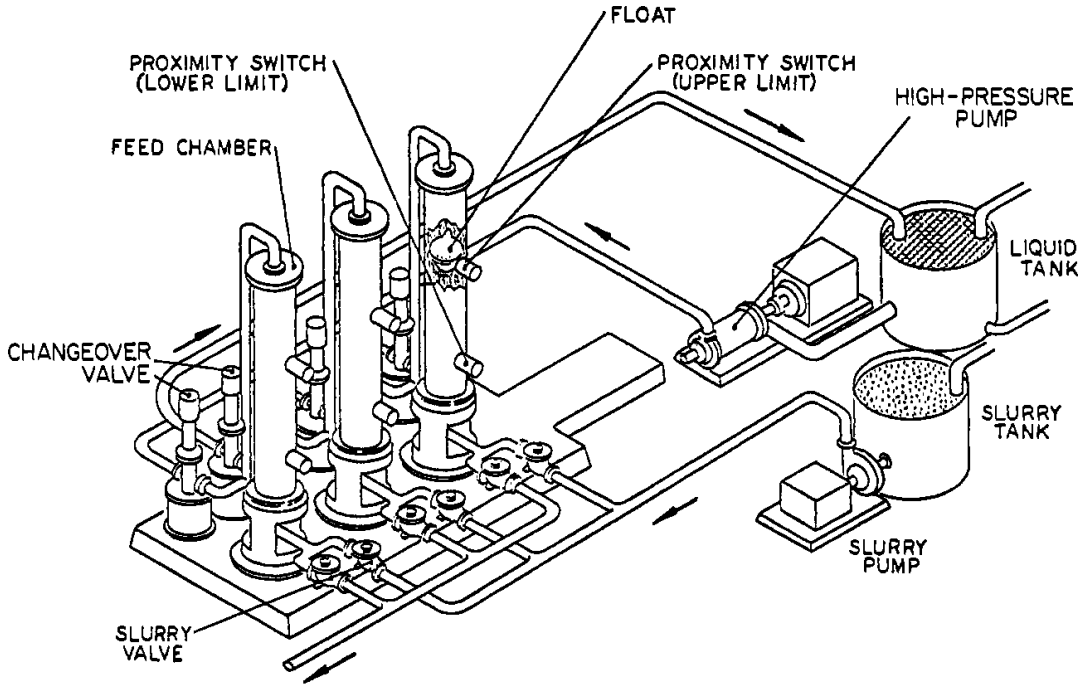


FIGURE 8 Large, low-cycle chamber feeders separate the high-pressure pumping element from the slurry (Hitachi)

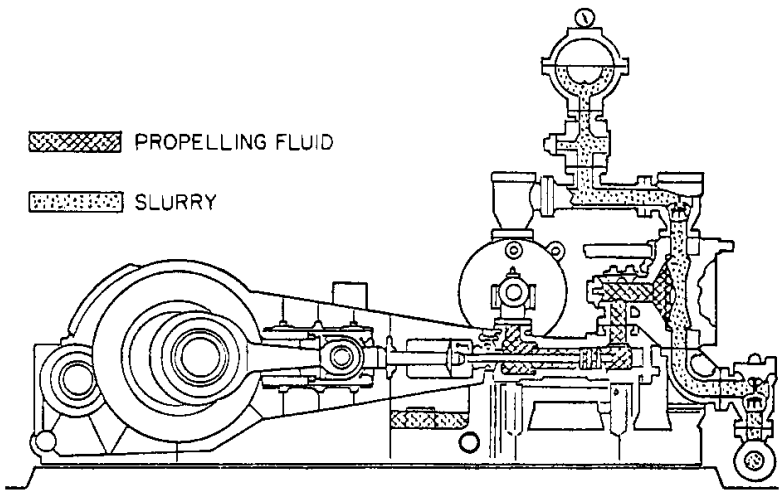


FIGURE 9 Diaphragm pump isolates slurry from packing (Geho)

The motivating liquid is pumped into a chamber and displaces a charge of slurry through suitable check valves. The chamber may have a moving separation barrier between the motivating and motivated liquids. When the slurry is expelled, controls cause valves to reverse the flow, filling the chamber again with slurry while voiding it of the clear liquid. By employing two or more chambers, a virtually continuous flow of slurry is obtained. Advantages of this scheme are the extra-low frequency of check valve operation, low slurry velocities, and high capacities. Centrifugal, positive displacement, or even compressed gas pumps can be used for motivation. Because the motivation pump is independent of the slurry system, it can operate at normal speed.

Two disadvantages of chamber feeders are their dependence on controls and switching valves in addition to the slurry valves and the fact that the systems are large.

TOTAL ISOLATION Many types of diaphragm pumps that completely separate the slurry from the running gear of the pump have been designed (Figure 9). Because the separating diaphragm is wetted on one side by the slurry, it is vulnerable to the mechanical, chemical, and thermal abuse that might otherwise apply to pistons, plungers, packing, and so on. Of course, the diaphragm is also subjected to full reversal loads and must be designed for fatigue life. Both elastomeric and metallic diaphragms have been used.

The severity of diaphragm failure problems has been alleviated somewhat in some designs that employ double diaphragms with intervening barrier liquids. The thesis is that there is a much reduced probability of both diaphragms failing before detection and repair.

DIAGNOSTIC SYSTEMS

Diagnostic monitoring of vibration, pressure pulsations, speed, flow rate, temperatures, and pressures may be applied to these pumps as well as other large machines. Such monitoring is relevant to slurry pumps because of the necessity to replace worn parts more frequently than is the case with clear-liquid pumps. Designers and users of displacement pumps for handling solids should consider using available monitoring equipment as an aid to maintenance and as a means to ensure the safety of personnel by warning of imminent failure. Vibration signatures are readily obtained from sensors placed in critical locations on the pump, and this information can be valuable in assessing the operating integrity of the pump. To this should be added valve wear detection, as would be indicated by a difference between the expected flow rate at the speed involved and the actual flow rate. In addition, proximity probes and vibration sensors could be used to monitor wear of plungers, piston rods, bearings, and crossheads.

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PUMP SEALING

SECTION 5.1
CENTRIFUGAL PUMP PACKING

JAMES P. NETZEL

Packing is used in the stuffing box of a centrifugal pump to control the leakage of the pumped out liquid, or the leakage of air in, where the shaft passes through the casing. This basic form of a seal can be applied in light- to medium-duty services and to those liquids that prove difficult for mechanical seals.

THE DESIGN OF PACKING RINGS

Packing may be referred to as compression, automatic, or floating. Each term describes the type of operation in which the packing will be used.

Automatic and floating packings require no gland adjustments in controlling leakage. Automatic packings are confined to a given space and are activated by the operating pressure. Automatic packing rings are designed in the form of V rings, U cups, and O rings. Floating packing includes piston rings and segmental rings that may be energized by a spring. These types of packing are commonly used in reciprocating applications.

Compression packing is most commonly used on rotating equipment. The seal is formed by the packing being squeezed between the inboard end of the stuffing box and the gland (see Figure 1a). A static seal is formed at the ends of the packing ring and at the inside diameter of the stuffing box. The dynamic seal is formed between the packing and shaft or shaft sleeve. Under a load, the packing deforms down against the shaft, controlling leakage. Some leakage along the shaft is necessary to cool and lubricate the packing. The amount of leakage will depend on the materials of construction for the packing, the operating conditions of the application, and the condition of the equipment.

Packing must be able to withstand equipment variables (see Figure 1b). The design of the packing ring and the materials of construction must be resilient to follow shaft runout and misalignments, as well as to compensate for thermal growth of the equipment without an appreciable increase in leakage.

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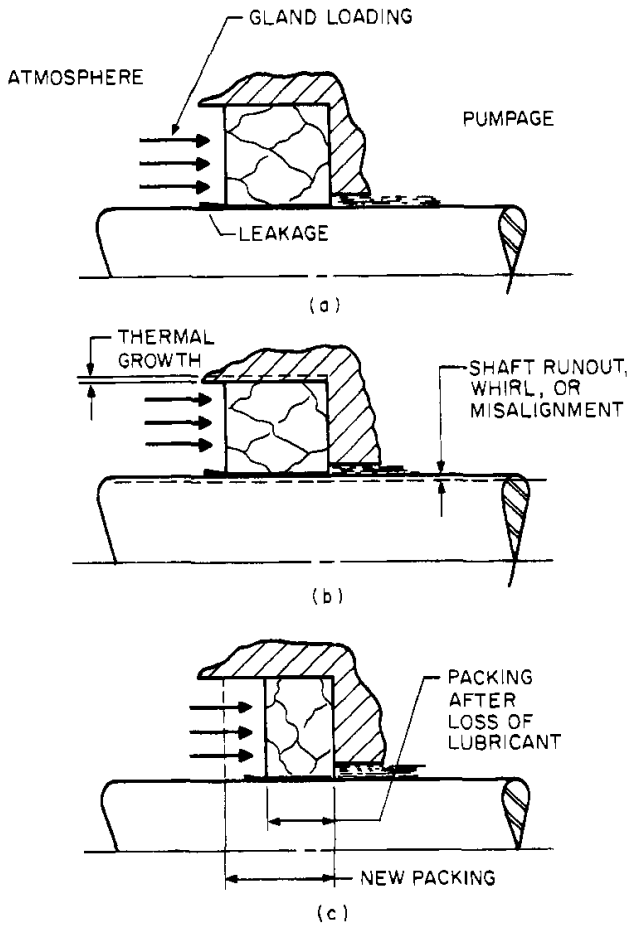


FIGURE 1 Compression packing; (a) new packing installation, (b) installation variables, (c) installation after many adjustments

As rotating equipment is operated, the load on the packing must be adjusted to control leakage (see Figure 1c). Care should be taken not to overtighten the packing. Most compression packings have a lubricant designed into them to prevent overheating of the packing or scoring of the shaft. Repeated adjustments will drive some of the lubricant from the packing, which will result in reduced operating time.

Compression packings are made of twisted, braided, woven, or wrapped elements formed into square or round cross-sections or other configurations. Square cross-sections are more common for rotating equipment.

A selection of the proper materials for the packing must include the chemical resistance to the product being sealed as well as the temperature, pressure, and shaft speed. Complete lists of the construction materials, packing lubricants and binders are given in Tables 1 and 2.

OPERATING FUNDAMENTALS

The Size and Number of Packing Rings The number of packing rings may vary, depending on the objective of the sealing system or the requirements of the rotating equipment. The most common packing arrangement for rotating equipment is illustrated in Figure 2. Three rings of packing are used to seal the process liquid from the packing lubricant. Two rings between the lantern and gland are used to restrict the leakage of the lubricant to the atmosphere. The size of the packing depends on the size of the equipment. Typically, for rotating shafts, the standard square size packings shown in Table 3 may be considered.

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TABLE 1 Common materials of construction for packing

Fibers				Metals
Mineral	Animal	Vegetable	Synthetic	Lead
Metal	Wool	Flax	Nylon	Copper
Graphite	Hair	Ramie	Rayon	Brass
	Leather	Jute	PTFE	P-bronze
		Cotton	Carbon	Aluminum
		Paper	Aramid	Iron
			Polyamide	Stainless Steel
				Nickel
				Monel
				Inconel
				Zinc

TABLE 2 Common lubricants and binders for packing

Lubricants		Dry Lubricants	Binders
Mineral	Animal	Graphite	Grease
Lube Oil	Tallow	Moly	Waxes
Paraffin	Glycerol	Mica	Elastomers
Petrolatum	Beeswax	Talc	TFE
Waxes	Lard Oil	Teflon	Other Resins
Greases	Fish Oil	Carbon	
	Soap	Tungsten Disulfide	
Vegetable	Synthetic		
Caster Oil	Oils		
Palm Oil	Waxes		
Cottonseed Oil	Fluorolubes		
Linseed Oil	Silicones		
Carnauba Wax			

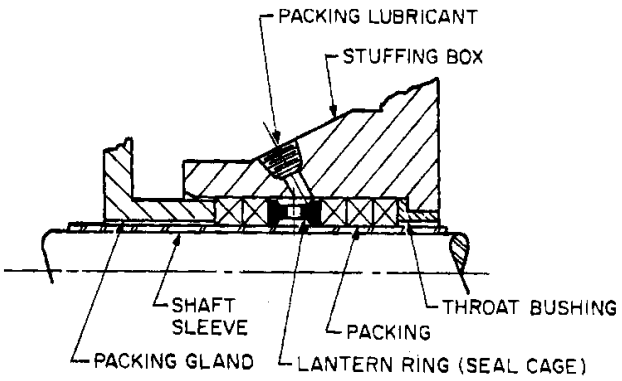


FIGURE 2 Common packing arrangement

TABLE 3 Packing sizes for rotating shafts

Shaft (or sleeve) Diameter, in (mm)	Packing size, in (mm)
$\frac{5}{8}$ to $1\frac{1}{8}$ (15 to 30)	$\frac{1}{4}$ (6.3)
$1\frac{1}{8}$ to $1\frac{7}{8}$ (30 to 50)	$\frac{5}{16}$ (8)
$1\frac{7}{8}$ to 3 (50 to 75)	$\frac{3}{8}$ (10)
3 to $4\frac{3}{4}$ (75 to 120)	$\frac{1}{2}$ (12.5)
$4\frac{3}{4}$ to 12 (120 to 305)	$\frac{5}{8}$ (16)

The packing size for an existing piece of equipment can be found by using the formula:

$$\text{Packing size} = \frac{\text{box ID} - \text{shaft or sleeve OD}}{2}$$

For packing to operate properly, the finish on the shaft sleeve must be at least 16 μin (0.4μm) centerline average (CLA) and the finish in the bore should be 63 μin (1.65 μm) CLA. The sleeve *must* be harder than the packing and chemically resistant to the liquid being sealed. If the sleeve has a coated material for a hard-wear surface, the sleeve must also have good thermal shock resistance.

Lantern Rings (Seal Cages) When an application requires that a lubricant be introduced to the packing, a lantern ring is used to distribute the flow (refer to Figure 2). This ring is used at or near the center of the packing installation. For ease of assembly, most lantern rings are axially split. The construction materials range from metal to TFE (tetrafluoroethlyene). TFE lantern rings are usually filled with glass or with glass and molybdenum disulfide. They are inherently self-lubricating and will not score the shaft. A throat bushing at the bottom of the stuffing box can be used to provide a closer clearance with the shaft to prevent packing extrusion.

Stuffing Box Gland Plates All mechanical packings are mechanically loaded in the axial direction by the stuffing box gland (refer to Figure 2). In cases where leakage of the process liquid is dangerous or can vaporize and create a hazard to operating personnel, a smothering gland is used to introduce a neutral liquid at lower temperatures (see Figure 3). A sufficient quantity of quenching liquid should be used to eliminate the danger from the

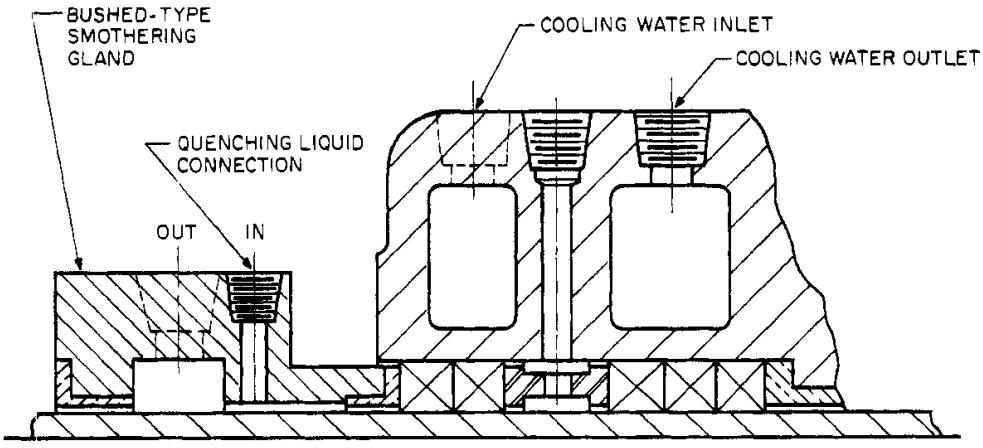


FIGURE 3 A smothering gland and water-cooled stuffing box

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TABLE 4 Leakage to prevent packing burning and sleeve scoring

Pressure lb/in² (bar)	Leakage, drops/min	cc/min
0–60 (0–4.0)	60	4
61–100 (4.1–6.8)		190
101–250 (6.9–17)		470

liquid being pumped. The neutral liquid circulated in the gland mixes with the leakage and carries it to a safe place for disposal. Close clearances in the gland control the leakage of the combined liquids to the atmosphere. This quench can also be used to protect the packing from any wear through abrasion, because the leakage cannot vaporize and leave behind abrasive crystals.

Glands are usually made of bronze, but cast iron or steel can be used for all-iron pumps. When iron or steel glands are used, they are normally bushed with a nonsparking material like bronze or nonmetals like TFE.

Leakage and Power Consumption The basic operating parameters for compression packing are the PV (pressure × velocity) factor and the projected bearing area of the assembly. Together they determine the rate of heat generation for the system. Some leakage of the product being sealed or of the packing lubricant is necessary to keep the packing from burning up or scoring the shaft sleeve. The minimum values for leakage at different packing pressures are given in Table 4. Flexible graphite and carbon filament compression packings can be used with reduced leakage rates, or in dry, gas-tight pump stuffing boxes, as the developed heat is dissipated through the packing and pump housing.

The exact pressure P between the shaft sleeve and the packing is a function of the pressure distribution over the length of the packing and the axial loading from the gland. For ease of calculation when determining the PV value, the gage pressure of the liquid at the packing is multiplied by $\pi \times$ the packing ID $\times$ rpm. The heat generation at the packing can then be estimated as

$$Q = \frac{f\pi^2 PND^2 L}{CJ}$$

where Q = heat generated, Btu/min (W)
f = coefficient of friction
P = liquid pressure at packing, lb/in² gage (bar)
N = shaft speed, rpm
D = sleeve OD or packing ID, in (m)
L = sleeve length covered by packing, in (m)
C = 12 (60 for SI units)
J = mechanical equivalent of heat = 778 ft · lb/Btu (1 N · m/s · W)

The coefficient of friction for various packings at a pressure of 100 lb/in² (6.8 bar) is given in Table 5. The heat generated by the packing must be removed by the leakage through the packing.

TABLE 5 Typically coefficients of friction

Material	f
Plain cotton	0.22
TFE impregnated fiber	0.17
Grease-lube fiber	0.10
Flexible graphite	0.05

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FIGURE 4 Graphite acrylic packing in continuous form (John Crane Inc.)

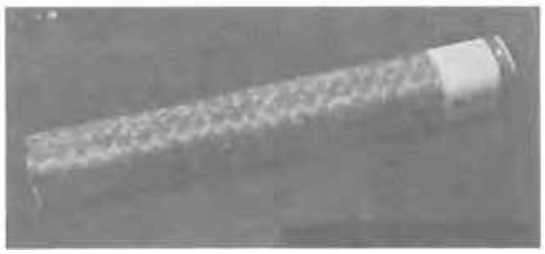


FIGURE 5 Interlaced construction packing (John Crane Inc.)

APPLICATION INFORMATION AND SEALING ARRANGEMENTS

Materials of Construction Basically, stuffing box packing is a pressure breakdown device. In order for a packed stuffing box to operate properly, the correct packing must be applied and the appropriate design features must be included in the system. Numerous types of packing materials are available, each is suited to a particular service. These may be grouped into three categories:

1. **Interlaced (braided) packing** Materials used for interlaced (braided) packing include cotton, TFE filament, Aramid fiber, aromatic polyamides, graphite/carbon yarn, and flexible graphite. All of these materials, with the exception of flexible graphite, can be impregnated with various lubricants. An example of graphited acrylic packing is shown in Figure 4. With the exception of flexible graphite, all the materials are of an interlaced construction for greater flexibility (see Figure 5). Packing made with this type of construction remains intact even if individual yarns wear away at the inside diameter of the packing.
2. **Flexible graphite** These types of packing rings were originally available cut from laminated sheet stock that required the rings to be made with an interference fit. Flexible graphite rings are made today from ribbon tape that is easily compressed to the shaft and bore to affect a better seal. Flexible graphite rings are available in split or endless rings (see Figure 6) or as the ribbon tape itself for ease of maintenance and installation. Operating limits for common packings are found in Table 6.
3. **Metallic packing** The basic materials of construction are babbit, aluminum, and copper in either wire or foil form. Metallic packing rings have flexible cores of materials such as twisted glass fiber. The packing is impregnated with graphite grease and/or oil lubricants (see Figures 7 and 8). Babbit is used in water and oil services at



FIGURE 6 Flexible graphite packing rings (John Crane, Inc.)

TABLE 6 Service limitations of common packing materials[a]

Packing Material	Pressure (max.)[b] lb/in² gage (bar)	PV rating (max.)[c] lb/in² gage · fpm (bar · m/s)	Temp. (max)[d] °F (°C)	pH Range	Comments
Cotton	100 (6.8)		150 (65.6)	5–7	Nonabrasive material; for cold water and dilute salt solutions
Flax/ramie	100 (6.8)	188,000 (65.8)	150 (65.6)	5–7	High, wet strength and excellent resistance to fungi and rotting; for cold water and dilute salt solutions
Plastic	100 (6.8) 250 (17)	188,000 (65.8) 471,000 (165)	600 (315.5) 150 (65.6)	4–8	Excellent sealing qualities, reacts well to gland adjustments, can extrude at higher pressure if not backed up by braided or metallic packing
Graphited acrylic	250 (17)	471,000 (165)	350 (175)	4–8	For mild chemicals and solvents
Acrylic TFE-impregnated	250 (17)	471,000 (165)	350 (175)	2–10	For mild chemicals and solvents
Babbitt (lead)	250 (17)	471,000 (165)	450 (232.2)	2–10	Shaft sleeve must have a Brinell hardness of 500 or more; for hot oils and boiler feed water
Aluminum or copper	250 (17)	471,000 (165)	750 (398.8)	3–10	Shaft sleeve must have a Brinell hardness of 500 or more; for hot oils and boiler feed water
TFE filament	250 (17)	471,000 (165)	500 (260)	0–14	For corrosive liquids and food service; usually requires slightly higher break-in leakage
Aramid fiber	250 (17)	471,000 (165)	500 (260)	3–11	Strong resilient packing; maximum speed 1,900 fpm (9.6 m/s); good in abrasives and chemicals
Graphite/carbon filament[e]	250 (17)	471,000 (165)	750 (398.8)	0–14	For corrosive liquids and high- temperature applications
Flexible Graphite[f]	250 (17)	471,000 (165)	1000 (540)	0–14	Excellent conductor of heat from the sealing surfaces; operates with minimum leakage; excellent radiation resistance

(a) Continuous lubrication introduced at the lantern ring. This table is only a guide. Consult the packing manufacturer with complete operating conditions for exact recommendations.

(b) Pressure relates to the operating pressure at the stuffing box.

(c) PV data based on a 2 in (5.08 cm) shaft at 1,750 and 3,600 rpm.

(d) Temperature is the product temperature.

(e) Functional temperature. Graphite can be used up to 3000°F (1650°C) in nonoxidizing atmospheres.

(f) Functional temperature. Flexible graphite can be used to 5300°F (2970°C) in nonoxidizing atmospheres.

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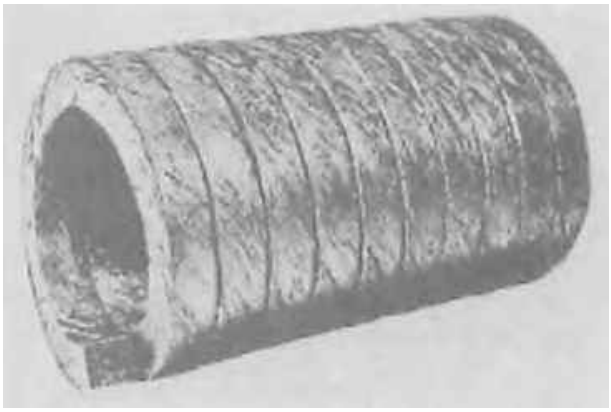


FIGURE 7 Metallic packing in spiral form (John Crane Inc.)

temperatures up to 450°F (229°C) and pressures up to 250 lb/in^2 (17 bar). Copper foil is used with water and low sulfur oils. Aluminum is used in oil service and heat transfer fluids. Both copper and aluminum have a temperature range up to 1000°F (537°C) and pressures of 250 lb/in^2 (17 bar). Babbit is not suitable for running against brass or bronze shaft sleeves, and where copper or aluminum is used, the sleeves should be 550 Brinell (55–60 Rockwell C) or harder.

Additional service limitations for common packing materials can be found in Table 6.

ENVIRONMENTAL LIMITATIONS

Pressure Every pumping application results in either positive or negative pressure at the throat of the pump stuffing box. A positive pressure will force the liquid pumped through the packing to the atmosphere side of the pump. Higher pressure will result in greater leakage from the pump. This results in excessive tightening of the gland, which



FIGURE 8 Metallic packing in ring form (John Crane Inc.)

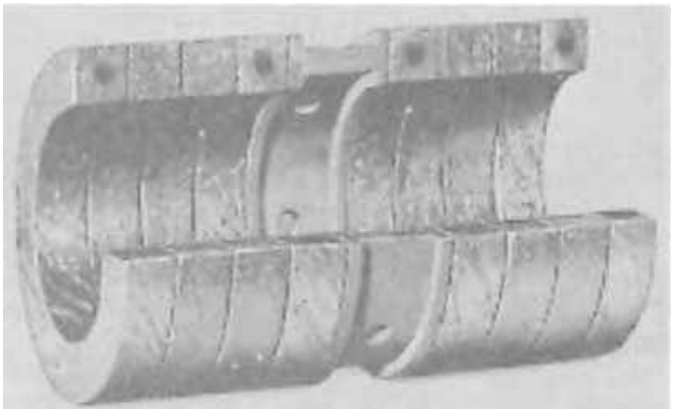


FIGURE 9 Combination of hard and soft packing (John Crane Inc.)

causes accelerated wear of the shaft or shaft sleeve and packing. For pressures at the stuffing box greater than 75 lb/in^2 (5.1 bar), some means of throttling the pressure should be considered. A combination of hard and soft rings die-formed to the exact stuffing box bore and sleeve dimensions can be used (see Figure 9). Harder rings at the inboard end of the box and at the lantern ring and gland break down the pressure and prevent the extrusion of the packing.

If the packing itself cannot be used to break down the pressure, then a throttle bushing must be used (see Figure 10). This is a typical arrangement for a vertical turbine pump where the stuffing box is subject to the discharge pressure of the pump. Here the throttle bushing is used to bring the liquid to almost suction pressure, and most of the leakage through the bushing is bled back to the suction. When the suction pressure is less than atmospheric, as in condensate pump service, an orifice is used in the piping back to the suction in order to maintain pressure above atmospheric pressure in the stuffing box. This prevents air leakage into the pump and excessive flow and wear at the bushing.

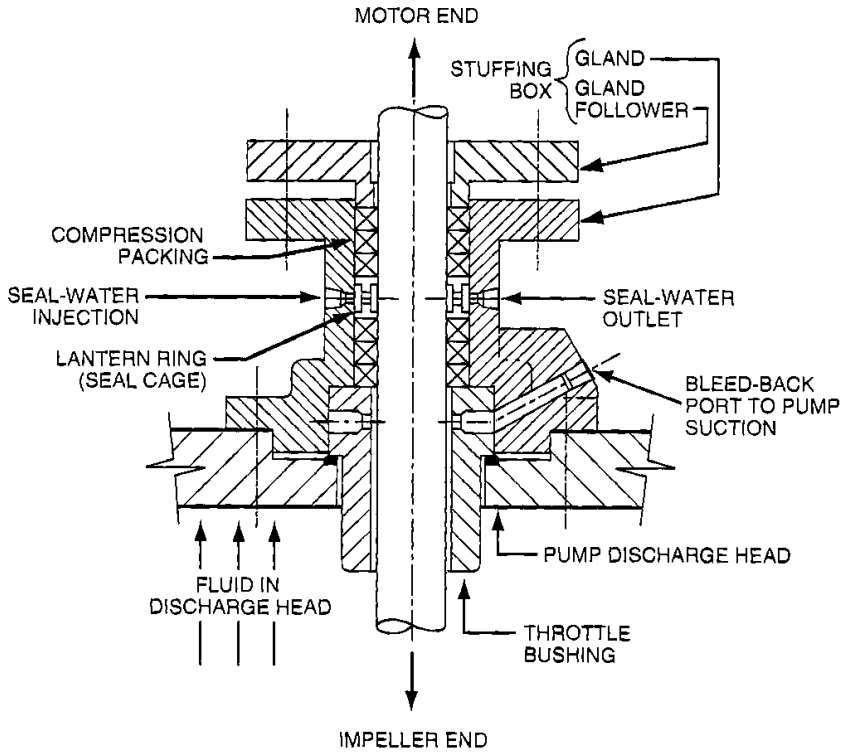


FIGURE 10 Compression packing with throttle bushing for pressure breakdown

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When a pump is fitted with a bypass line from the discharge, a valve can also be used to reduce the pressure at the stuffing box. This is another method for reducing pressure for the benefit of the installation.

Air Leakage When the pressure at the stuffing box is atmospheric or just below during normal operation, a bypass from the pump casing discharge through an orifice can be used to inject liquid into the lantern ring. The sealing liquid will flow partly into the pump and partly out to the atmosphere, thereby preventing air from entering at the stuffing box. This arrangement is commonly used to handle clean, cool water. In some pumps, these connections are arranged so that liquid can be introduced into the lantern ring through internally drilled passages.

When the negative pressure is very low, such as when the suction lift is in excess of 15 ft (4.6 m), an independent injection of 10 to 25 lb/in^2 (4.8 to 11.7 bar) higher than the atmospheric pressure must be used on the pump. Otherwise, priming may be difficult. Hot well pumps or condensate pumps operate with as much as 28 in (0.7 m) of vacuum, and air leakage into the pump would occur even on standby service. Here a continuous injection of clean, cool water is required, or alternatively, a cross connection of sealing water to another operating pump will provide sealing pressure as long as one pump is operating. A lantern ring is provided for this, as shown in Figure 10.

Temperature The control of temperature at the stuffing box is an important factor in promoting the life of the packing. Even though packings are rated for high product temperatures, cooling in most cases is desirable. The heat developed at the packing must also be removed. The rules of thumb are as follows:

- For light service conditions with pressures at 15 lb/in^2 (1 bar) and temperatures at 200°F (90°C), cooling is desirable.
- For medium service conditions with pressures at 50 lb/in^2 (3.4 bar) and temperatures at 250°F (118°C), lantern ring cooling is desirable, such as one gpm (3.78 l/min) at 5 lb/in^2 (0.34 bar) above process pressure.
- For high service conditions with pressures at 100 lb/in^2 (6.8 bar) and temperatures at 300°F (131°C), lantern ring cooling is desirable, such as one gpm (3.78 l/min) at 5 lb/in^2 (0.34 bar) above process pressure plus a water-cooled stuffing box (refer to Figure 3).

When cooling water is required, it is circulated through the stuffing box at the lantern ring. For horizontal-shaft pumps, the inlet should be at the bottom, with the outlet at the top of the stuffing box. Some of the coolant may flow to the process liquid and some to the atmosphere. The outboard packing rings seal only the cool liquid.

If the product to be sealed will solidify, then the packing box will have to be heated before the pump is started. This can be accomplished by steam or electric tracing of the pump stuffing box.

Abrasives Liquids that contain abrasives in the form of *suspended solids* such as sand and dirt will shorten the life of the packing. Particles will imbed themselves in the packing and will begin to wear the shaft or shaft sleeve. Abrasives can be eliminated at the sealing surfaces by injecting a clean liquid into the lantern ring. The injection may take the form of a bypass line from the pump discharge through a filter or centrifugal separator. Where necessary, the clean injection may also be from an external source.

To keep the abrasives from the packing in horizontal-shaft pumps, the injection can be made directly to the inboard end of the stuffing box through the lantern ring (see Figure 11). A soft rubber gasket between the lantern ring and the box shoulder can be used to limit the flow of clean liquid and the ingress of abrasives.

When separators and filters cannot be used, an injection from an external source must be considered. Two rings of packing are located between the lantern ring and the inboard end of the stuffing box to keep the product dilution to a minimum. External flushing should be injected into the stuffing box at a pressure 10 to 25 lb/in^2 (1.7 bar) greater than the pressure at the inboard end of the box from the liquid being pumped. A regulating

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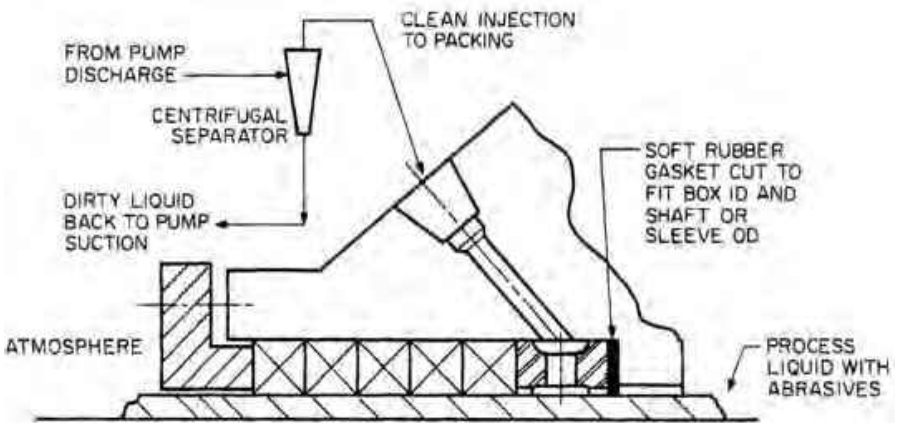


FIGURE 11 A clean injection through a centrifugal separator to keep abrasives out of the stuffing box

valve, illustrated in Figure 12, can be used to control the pressure and flow to the packing installation. Flow to the packing can be regulated to ensure the best operating environment for this seal, while conserving water used for injection.

Dissolved solids in the liquid being pumped can also create a wear problem. Here an increase or decrease in stuffing box temperatures may be necessary to keep solids in solution.

INSTALLING CONTINUOUS COIL PACKING

To install continuous coil packing, perform the following steps:

1. Loosen and remove the gland from the stuffing box.
2. Using a packing puller, begin to remove the old packing rings.
3. Remove the split lantern ring (if present) and then continue removing the packing with the puller.

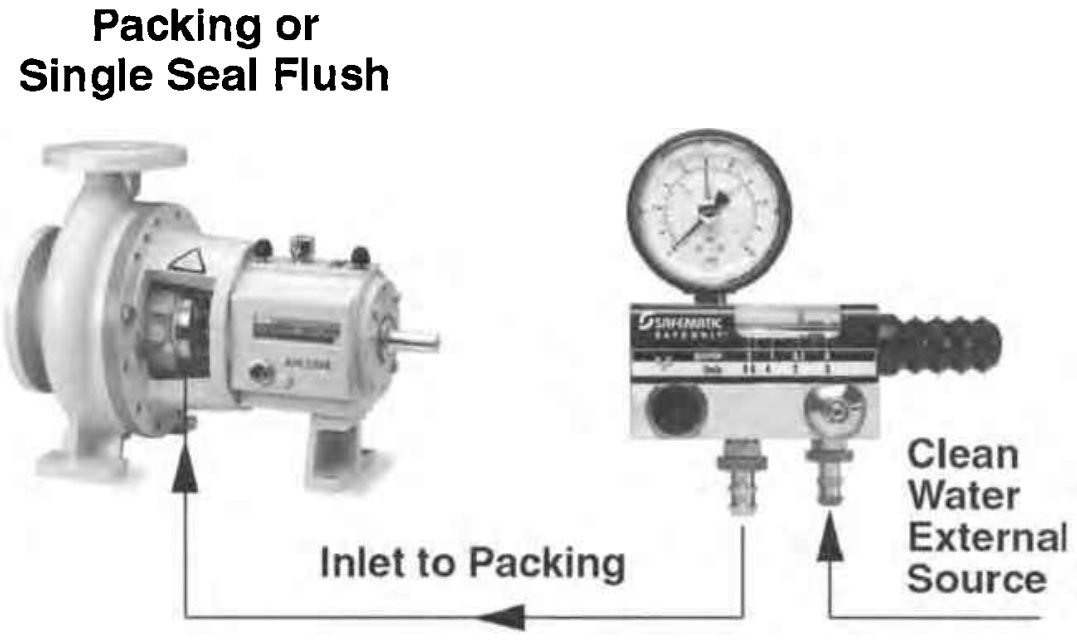


FIGURE 12 A regulation valve that controls and monitors the flow of water to the packing (Safematic)

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TABLE 7 Packing troubles, causes, and cures

Trouble	Cause	Cure
No liquid delivered by pump	Lack of prime (packing loose or defective, to allowing air leak into suction)	Tighten or replace packing and prime pump.
Not enough liquid delivered by pump	Air leaking into stuffing box	Check for leakage through stuffing box while operating. If no leakage occurs after after reasonable gland adjustment, new packing may be needed. or Lantern ring may be clogged or displaced and may need centering in line with sealing liquid connection. or Sealing liquid line may be clogged. or Shaft or shaft sleeve beneath packing may be badly scored, allowing air to be sucked into pump.
	Defective packing	Replace packing and check the smoothness of the shaft or shaft sleeve.
Not enough pump pressure	Defective packing	As per preceding
Pump works for a while and then quits	Air leaks into stuffing box	As per preceding
Pump takes too much power	Packing too tight	Release gland pressure and retighten reasonably. Keep leakage flowing. If none, check packing, sleeve, or shaft.
Pump leaks excessively at stuffing	Defective packing	Replace worn packing or replace packing damaged by lack of lubrication.
	Wrong type of packing	Replace packing not properly installed or run in. Replace improper packing with correct grade for liquid being handled.
	Shaft or shaft sleeve scored	Put in lathe and machine-true and smooth or replace. Recheck dimensions for correct packing size.
Stuffing box overheats	Packing too tight	Release gland pressure and retighten.
	Packing not lubricated	Release gland pressure and replace all packing if any burnt or damaged.
	Wrong grade of packing	Check with pump or packing manufacturer for correct grade.
	Insufficient cooling water to jacket	Check for open supply line valve or clogged line.
	Stuffing box improperly packed	Repack.
Packing wears too fast	Shaft or shaft sleeve worn or scored	Remachine or replace.
	Insufficient or no lubrication	Repack, making sure packing is loose enough to allow some leakage.
	Packing packed improperly	Repack properly, making sure all old packing is removed and the box is clean.
	Wrong grade of packing	Check with pump or packing manufacturer.
	Pulsating pressure on external seal liquid line	Remove cause of pulsation.

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4. After the packing has been removed, check the sleeve for scoring and nicks. If the shaft sleeve or shaft cannot be cleaned up, it must be replaced. Check the size of the stuffing box bore and the shaft sleeve or shaft diameter to determine which size packing should be used.
5. After the size of the packing has been determined, wrap the packing tightly around a mandrel, which should be the same size as the pump shaft or sleeve. The number of coils should be sufficient to fill the stuffing box. Cut the packing along one side to form the individual rings.
6. Before beginning the assembly of any packing material, be sure to read all the instructions from the manufacturer. Assemble the split packing rings on the pump. Each ring should be sealed individually with the split ends staggered 90° and the gland tightened to seal and fully compress the ring. Be sure the lantern ring is reinstalled correctly at the flush connection. Then back off the gland and retighten it, but only finger-tight. The exception to this procedure is that TFE packing should be installed one ring at a time, but not seated because TFE packings have high thermal expansion.
7. Allow excess leakage during break-in to avoid the possibility of rapid expansion of the packing, which could score the shaft sleeve or shaft so that leakage could not be controlled.
8. Leakage should be generous upon startup. If the packing begins to overheat at startup, stop the pump and loosen the packing until leakage is obtained. Restart only if the packing is leaking.

CAUSES FOR A SHORT PACKING LIFE

In order for packing to operate properly, the equipment must be in good condition. Shafts should be checked for runout and eccentricity to be sure they are within the manufacturer's recommended tolerances. Surfaces in contact with the packing should be finished to the correct smoothness and tolerance. Table 7 lists common troubles that affect the packing life. Causes and possible cures are also given.

FURTHER READING

Berzinsa, A. "Finer Points of Compression Packing Help with Centrifugal Pumps," *Power*, June 1974, pp. 58–59.

Ciffone, J. G. "Packing," John Crane Mechanical Maintenance Training Center, Morton Grove, Illinois, November 1994.

Crane Packing Company. *Engineering Fluids Sealing, Materials, Design and Applications: An Information Compendium of Technical Papers*. Morton Grove, IL 1979.

Elonka, S. "Packing: Power Practical Manual," *Power*, March 1955, pp. 107–130.

Krisle, O.M. "How to Get Longer Life out of Pump Packing and Shaft Sleeve," *Water and Sewage Works*, June 1967, pp. 199–201.

Neale, M.J., "Packing Glands," *Tribology Handbook* (Sec. A36), Wiley, New York; 1973.

SECTION 5.2
CENTRIFUGAL PUMP MECHANICAL SEALS

MICHAEL B. HUEBNER

INTRODUCTION

Mechanical seals can be simply described as devices to minimize or prevent process fluids or gases from leaking out of the pressure casing of a pump along a rotating shaft. In practice, though, a mechanical seal is a highly engineered device capable of operating under a wide variety of fluids, pressures, temperatures, and speeds. Over the decades of their development, mechanical seals have evolved from simple flat rings to very sophisticated sealing surfaces. Through this evolution, they still continue to serve the same purpose: to allow pumps and rotating equipment to operate reliably and efficiently while minimizing process leakage to atmosphere.

Historically, there have been a number of concepts applied to sealing a rotating shaft in a stationary housing. These include mechanical packing, labyrinth seals, close clearance bushings, and split carbon rings. All of these techniques have been used successfully and continue to be applied in specific applications. They all share a common limitation though they all attempt to control leakage through an annular clearance. The most common method for controlling leakage is to reduce the annular clearances between the stationary and rotating components to a minimum. This has a practical limit based on shaft motion, thermal expansion, and wear of components. The result is a reduction in leakage but not to the extent required for most modern sealing applications.

To overcome these limitations, mechanical seals move the sealing surfaces to a plane perpendicular to the shaft rotation. In this location, the sealing surfaces allow for radial motion and thermal expansion without affecting the clearance between the faces. The seal faces are mounted so that they float on a thin fluid film and allow for small axial motions of the shaft caused by pump manufacturing tolerance and alignments. In this orientation, the fluid film between the seal faces can be orders of magnitude smaller than in annular seals while minimizing face wear and frictional drag. Current seal technology can result in reliable seal performance for many years without operator adjustment or repair.

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DESIGN FUNDAMENTALS

While the configurations and complexity of mechanical seals vary significantly, they all operate with the same fundamental principles. All seals have one face that is affixed to the stationary pump casing and one face that is affixed to and rotating with the pump shaft. The gap between these faces controls the performance of the seal. The stationary face is generally mounted into a seal gland that interfaces with the pump casing. The rotating face is generally mounted to a sleeve that interfaces with the pump shaft. Appropriate adaptive components and gaskets complete the design. The design and material selection of these components help establish the operating limits for a specific design, which can range from a simple low-duty seal (Figure 1) to a more complex, higher duty seal (Figure 2).

Even though all mechanical seals share many of the same components, the individual component design may vary significantly between different seals. This can be a function of the operating window for the seal, the materials of construction, the size of the installation window in the pump, or other application requirements (such as cleaning or sterilization). It can also depend upon factors as diverse as the presence of abrasive particles or allowances for large shaft motions. For these reasons, there will never be a universal seal design suitable for every application. There will always be a wide range of offerings from seal OEMs.

Mechanical seals can generally be divided into two main design types—pusher seals and bellows seals. Pusher seals have one of the seal faces mounted on a flexible gasket that allows the face to move axially (Figure 3). Bellows seals have one of the seal faces mounted in a bellows assembly that provides a pressure boundary while allowing for axial face motion. Bellows designs vary from elastomeric construction to more complex welded metal designs (Figure 4). Each of these designs has advantages for specific applications.

Mechanical seals can also be provided as a component seal or a fully assembled cartridge seal (Figure 5). Component seals consist of a limited number of critical seal components including the seal faces, springs, and adaptive gaskets. This design relies on internal mating dimensions in the pumps or manual installation to establish the seal setting. Installation of a component seal requires careful practices and a clean environment to prevent contamination or damage of the seal faces. Cartridge seals are fully assembled prior to installation. They are installed by bolting the complete cartridge into the pump seal chamber. This ensures the correct seal setting and minimizes the opportunity for damage to occur. The complete cartridge can also be tested for integrity by the seal OEM before shipping. The cartridge configuration has become more popular due to improved reliability and is required in some seal standards.

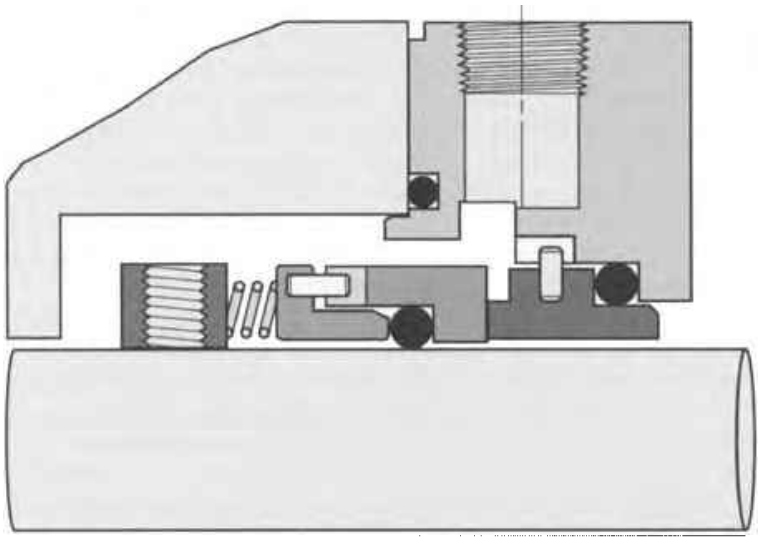


FIGURE 1 Low duty mechanical seal

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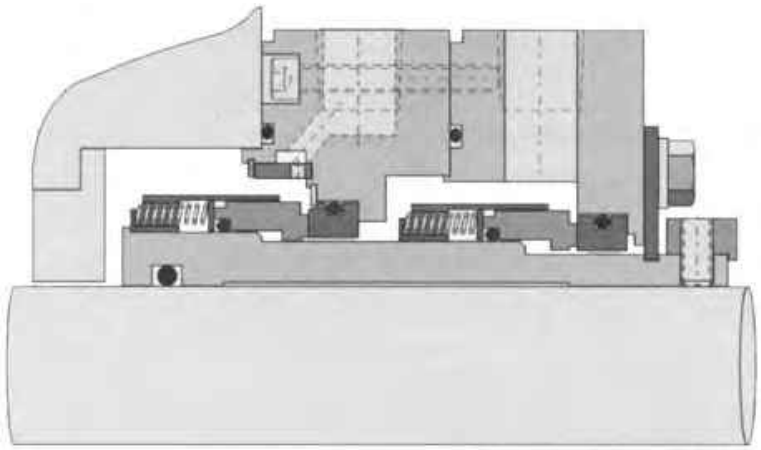


FIGURE 2 General duty mechanical seal

SEAL PERFORMANCE

The faces of a mechanical seal are the most critical design element and largely define the operating window of the seal. The seal designer has great flexibility in the face design to control face deformations and hydraulic loading. These factors control the formation of the fluid film, seal leakage, power requirements, and life of the seal. If the face contact pressure is too high, it will result in too thin of a fluid film, which will create high heat and excessive face wear. If the contact pressure is too low, it will result in a thicker fluid film, which will allow excessive seal leakage.

The primary method for controlling the contact pressure between the faces is balancing the hydraulic closing forces against the hydraulic opening forces. The fluid in the seal chamber exerts an equal pressure on all hydraulically wetted surfaces. Hydraulic closing area is defined as the surfaces that act to increase the face contact pressure. The term "balance ratio" or seal balance is used to describe the ratio of the hydraulic closing area to the face contact area. The seal balance b is defined as:

$$b = a_c/a_o$$

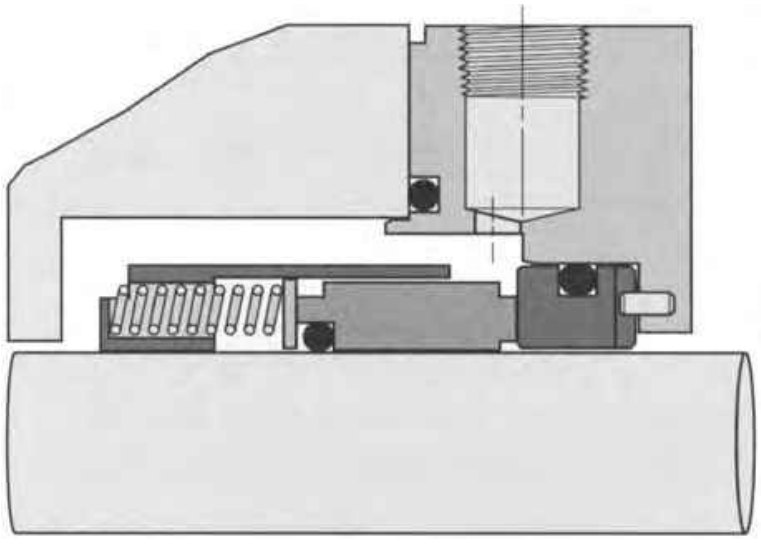


FIGURE 3 Pusher seal

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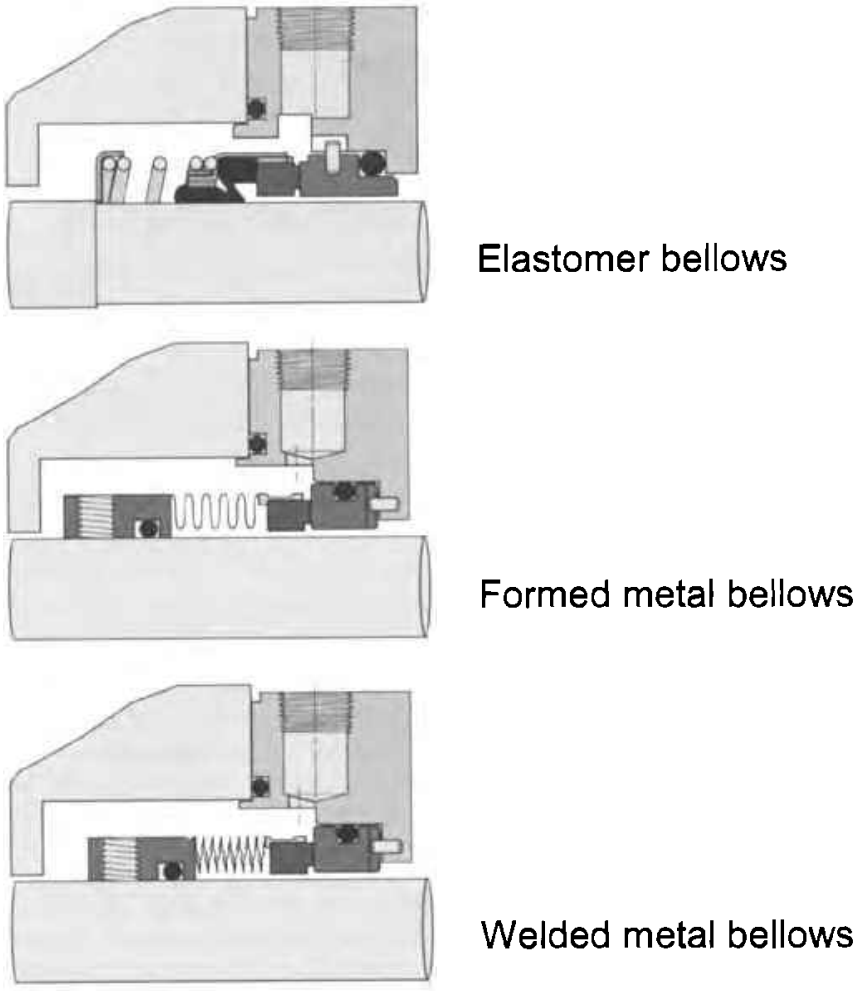


FIGURE 4 Bellows seal—elastomeric, formed, welded metal

where b = balance
a_c = hydraulic closing area, $in^2 (m^2)$
a_o = hydraulic opening area (or face contact area), $in^2 (m^2)$

For actual seal designs, the seal balance is defined in terms of the seal face diameters and the seal balance diameter. These are described by the following equations:

OD pressurized seals (Figure 6)

$$b = (OD^2 - BD^2) / (OD^2 / ID^2)$$

ID pressurized seals (Figure 7)

$$b = (BD^2 - ID^2) / (OD^2 / ID^2)$$

Commonly, contacting liquid seals have balances in the range of 0.65 to 0.85 in operation. Seals with balances less than 1.0 are defined as balanced seals indicating that the seal is designed to hydraulically reduce (or balance) the effects of the seal chamber pressure on the seal faces. Seals with balances of 1.0 or greater are defined as unbalanced seals. Unbalanced seals have a higher face loading and are generally limited to operating pressures less than 10 bar (145 psi). Seal balance can also be expressed as a percentage such as 75% balance.

In addition to the hydraulically generated closing forces, mechanical seals employ a spring device to mechanically load the seal faces. This mechanical loading provides closing

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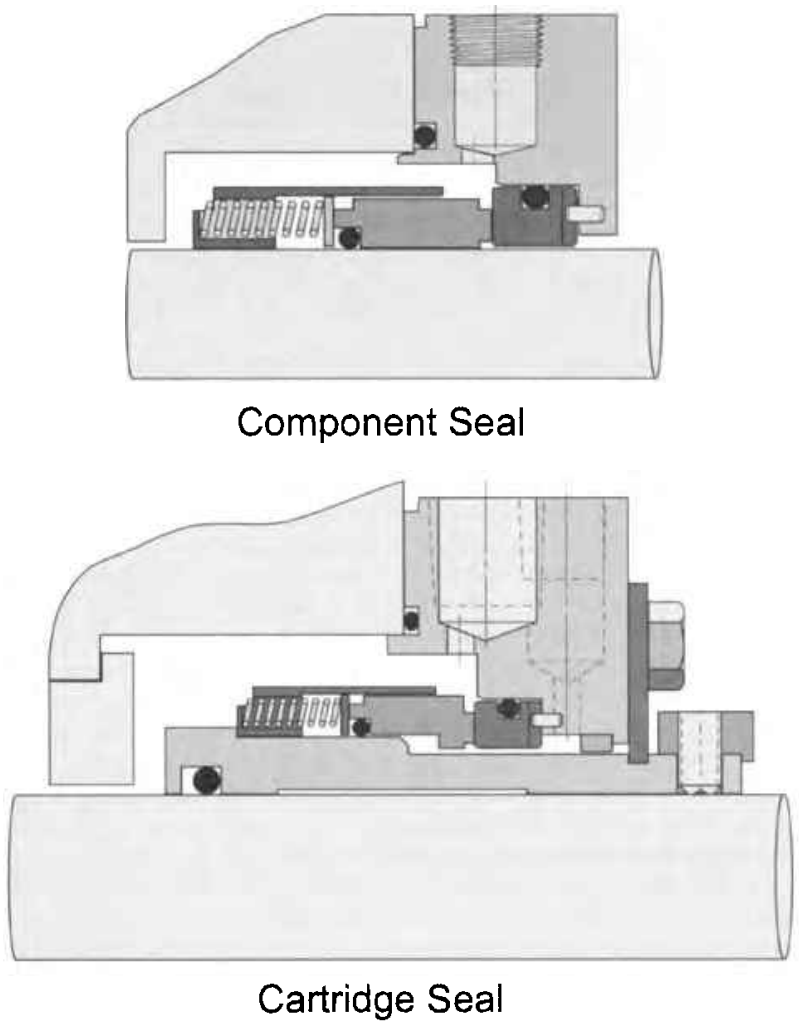


FIGURE 5 Component seal and cartridge seal

forces during low-pressure operation. It also provides the forces necessary to overcome axial friction forces on the seal face due to O-ring drag and drive friction. Spring pressure exerted on the faces is defined as:

$$P_{sp} = F_{sp} / a_o$$

Where P_{sp} = face contact pressure generated by the springs, psi (N/m^2)
F_{sp} = force generated by the springs at working length
a_o = hydraulic opening area (or face contact area), in^2 (m^2)

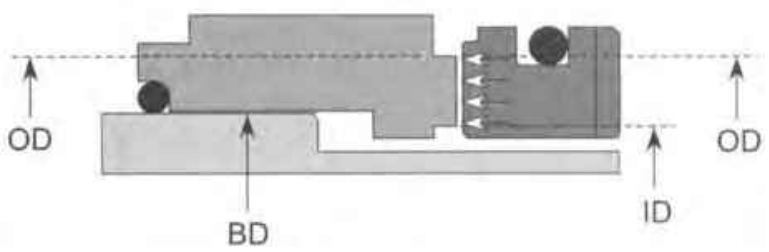


FIGURE 6 OD pressurized seal

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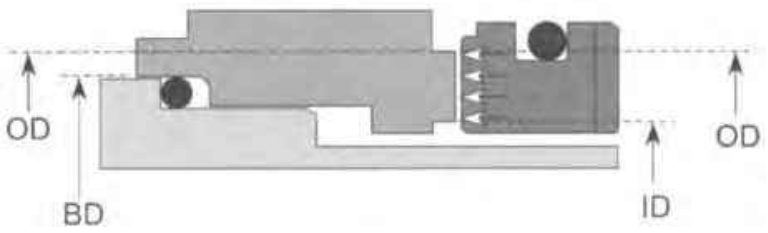


FIGURE 7 ID pressurized seal

In contacting liquid seals, P_{sp} will normally vary between 100 to 240 kPa (15 to 35 psi). For other designs such as dry running seals and gas seals, the value of P_{sp} will generally be approximately 14 to 40 kPa (2 to 6 psi).

The pressure in the fluid film between the seal faces creates an opening force that acts in a direction opposite the hydraulic closing force and the spring force. On a typical single seal, the pressure profile is not constant. It will be equal to the seal chamber pressure at the outer diameter of the seal faces, decreasing to atmospheric pressure at the ID of the seal face. The integration of the pressure across the face area will result in a force, which decreases the contact pressure between the faces. The exact profile of the pressure drop depends upon the flatness (or parallelism) of the seal faces and the state of the fluid film (liquid, gas, or flashing). The integration of the area across the face is described by the pressure gradient factor k and has a numerical value between 0 and 1. For a parallel liquid film, k has a value approximately 0.5. For seal faces which have a converging fluid film (film thicker on the high pressure side), k will be greater than 0.5. For faces with a diverging fluid film (film thinner on the high pressure side), k will be less than 0.5 (Figure 8).

The other factor which affects k is the phase or change of phase of the fluid between the seal faces (Figure 9). A fluid which remains in a liquid state between the seal faces will have a linear pressure drop as it migrates from the high pressure side to the low pressure side. If the fluid film is supported entirely by a gas film, the pressure profile is different due to the compressibility of the gas that results in a larger opening force. A fluid which changes from a liquid phase to vapor phase across the seal face will result in an even larger opening force. This is a normal condition with fluids that will flash under atmospheric conditions such as light hydrocarbons.

The goal of a successful seal design is to create the optimal fluid film thickness by using the seal balance b and the spring pressure P_{sp} to counteract the effects of the pressure drop coefficient k opening the seal faces. The ideal resultant face contact pressure allows for long seal life with minimal leakage and minimum heat generation. The total face contact pressure P_f can be calculated by the following equation:

$$P_f = \Delta P(b - k) + P_{sp}, \text{ kPa (psi)}$$

where ΔP = differential pressure across seal faces, kPa (psi)

With the calculation of P_f, other factors can be derived such as PV, starting torque, and power consumption.

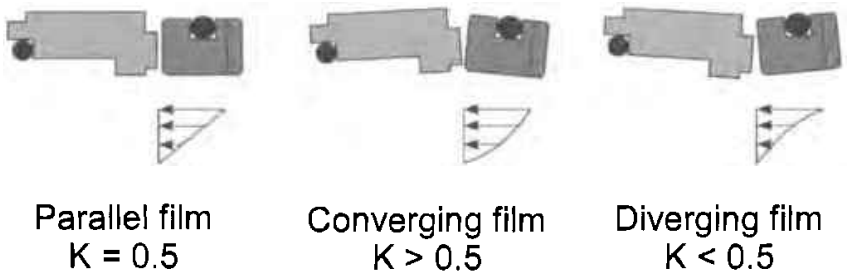


FIGURE 8 Variations in pressure drop coefficient k

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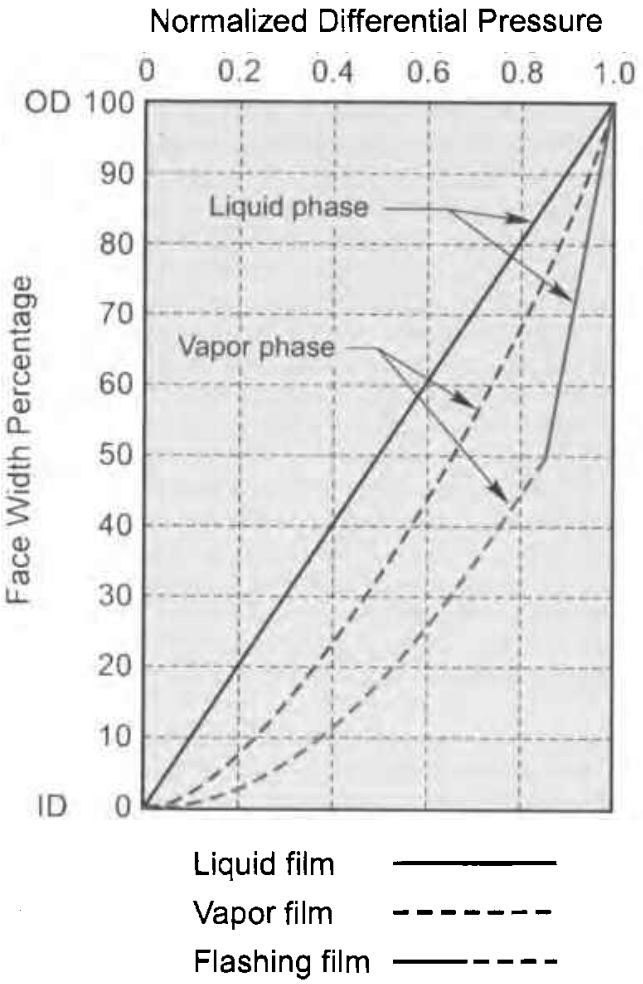


FIGURE 9 Variation in pressure profile with different fluid phases (Flowserve Corporation)

Pressure-Velocity The concept of pressure-velocity has a long history in tribology and material sciences. The product of the pressure times the velocity (or more commonly *PV*) is used to describe a sliding interface between two surfaces. While this is extremely useful in material science, it is less applicable to mechanical seals. Some published data for mechanical seals state *PV* limits considering only the material selection of the seal faces and does not consider the lubricating properties of the fluid film. Other references put *PV* values in terms of seal chamber pressure and not the contact pressure at the seal faces. The conflicting definitions and considerations in *PV* data require the reader to carefully understand the definition used and the considerations made in applying the *PV* limits to a specific seal.

The most accurate method for calculating *PV* is to multiply the total face contact pressure P_f with the velocity at the mean diameter of the seal face V_m.

$$PV = [\Delta P(b - k) + P_{sp}]\, V_m, \text{ kPa} \times \text{m/s (psi} \times \text{ft/min)}$$

where V_m = velocity at the mean diameter of the seal face, m/s (ft/min)

TABLE 1 Typical values for pressure gradient factors k

Non-flashing fluids	0.50
Flashing fluid	0.70 to 0.75
All vapor of gas	0.66

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Seal OEMs may produce *PV* tables for seals that can be of some value in evaluating a seal selection. The user should not extrapolate this data or compare it to *PV* values from other sources.

Overall Coefficient of Friction Two sliding surfaces exhibit resistance to motion, which is classically referred to as friction. This is characterized by the coefficient of friction, which is defined as the ratio of the tangential forces required to create or sustain motion divided by the normal forces between the surfaces. Friction in a nonlubricated interface is a predictable value based on the combination of the mating materials. The coefficient of friction in a mechanical seal face is more complex and less predictable. The resistance to motion is caused not only by classical friction but also by the shearing of the fluid in the interface, which is proportional to the fluid viscosity and inversely proportional to the film thickness. The combination of frictional effects from mechanical contact and fluid shear can be calculated only if all of the variables are known. Even then, it is only possible in sophisticated computer modeling of the seals.

To help provide a more accessible value for resistance to motion, mechanical seals calculations have used the concept of an effective coefficient of friction *f*. The values used for *f* represent the combined effects of all resistances to motion and can be used in standard equations for torque, power consumption, and heat generation. The numerical values for *f* have been derived from seal testing on a variety of fluids under a wide range of operating conditions. Published values of *f* typically range from 0.01 to over 0.15. Typical values for *f* (Table 2) have proven useful for predicting the performance of mechanical seals.

Torque The torque required to spin a mechanical seal is directly proportional to the total face pressure P_f, the face area, the coefficient of friction *f*, and the mean radius of the seal face. Torque is most often used to determine the breakout torque required to start a pump. Under these conditions, the value of *f* is generally three to five (3 to 5X) times higher than values found in a running seals:

$$T = P_f A f R_m, \text{ in-lb (N-m)}$$

where A = face area, in^2 (mm^2)

R_m = mean radius, in (mm)

Power Consumption Calculating the power requirements for a mechanical seal technically consists of the sum of the power requirements from three areas: seal faces friction, fluid turbulence, and pumping ring circulation. The friction between the seal faces creates a resistance to rotation, which must be overcome by the pump driver. The power required to rotate the seal faces P_f can be estimated with the following equation:

$$P_f = (T N) / C, \text{ kW (hp)}$$

where N = rotational speed rpm

C = constant for English units = 63,000 (constant for SI units = 9550)

Power is also consumed by windage or turbulence in the seal chamber. The actual power requirement is a function of the diameter of the seal, the seal rotational speed, the fluid viscosity, and the configuration of the seal chamber. Likewise, power is required for a pump-

TABLE 2 Effective coefficient of friction values for various fluids

Low viscosity fluids (e.g., light hydrocarbons)	0.05
Medium viscosity fluids (e.g., water based solutions)	0.07
High viscosity fluids (e.g., lubricating oils)	0.10

Note: The above values are typical for a seal face combination of carbon versus silicon carbide or tungsten carbide.

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ing ring (if applicable) and this is a function of the pumping ring and seal gland design, the rotational speed and the fluid viscosity. The seal OEM can estimate these power requirements if all of the factors are known. From lab and field experience though, the seal face generated is the most significant contributor to seal power consumption and actual power requirements can be estimated by considering only P_f.

Temperature Control For stable seal performance, the heat generated by the mechanical seal must be dissipated. This is done to control the temperature of the fluid film between the seal faces, minimize thermal distortion of the faces, and control the fluid properties of the process fluid. In many applications, this temperature rise in the seal chamber is controlled by providing a flow of fluid (generally process fluid) through the seal chamber. The temperature rise in the seal chamber ΔT can be estimated by the following equation:

$$\Delta T = P_f / (C_1 \cdot SG \cdot Cp \cdot Q)$$

where ΔT = temperature rise, °C (°F)
P_f = power, kW (Btu/h)
C_1 = constant for SI units = 0.06969 (constant for English units = 500)
SG = specific gravity
C_p = specific heat, kcal/kg °C (Btu/h °F)
Q = injection flush flow rate, L/m (gpm)

In most cases, an attempt should be made to keep the temperature rise in the seal chamber less than 10°F (5°C).

Modeling of Seal Faces Modern mechanical seals successfully operate on an effective fluid film of approximately 20 millionths of an inch (0.5 micron). To maintain such a thin film, the seal faces must be manufactured flat and they must remain flat in operation. This is a challenge considering the wide range of speeds, pressures, and fluids found in seal applications. There are many mechanisms that attempt to distort the seal faces. These include drive loads, gasket loads, spring forces, and shrink fits. Seal designers can effectively minimize these distortions through careful seal design. There are two other mechanisms though which are always present and offer the greatest challenge in seal design—thermal distortions and pressure distortions.

When a seal is in operation, heat is generated between the sliding seal faces that results in an increase in temperature in the interface. Heat is conducted through the seal faces and largely dissipated by convection into the process fluid. This results in a temperature gradient along the axial length of the seal face. Since the contacting end of the face is at a higher temperature, it will thermally grow more than the cooler end of the seal face. The sealing surface will no longer be flat but will become convex. This phenomenon is referred to as thermal distortion (Figure 10). Thermal distortion is a function of the heat generated, the

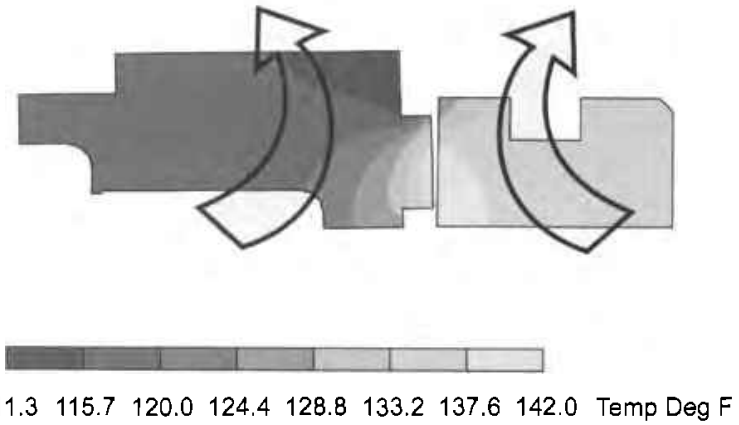


FIGURE 10 Thermal distortion

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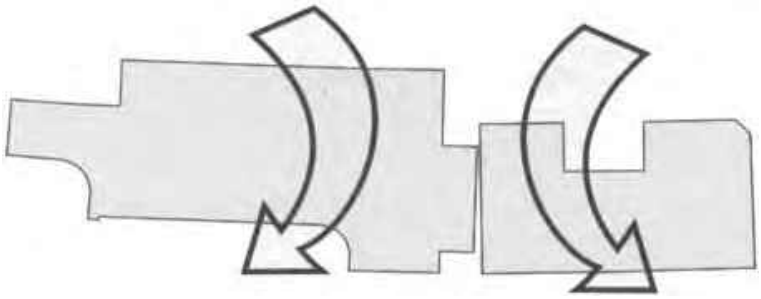


FIGURE 11 Pressure distortion

convective heat losses, the thermal conductivity of the seal face, and the coefficient of thermal expansion of the seal face. This distortion will result in a higher opening force between the faces and greater leakage from the seals.

To counteract thermal distortions, seal designers will design faces that tend to distort in the opposite direction so the sealing surfaces become concave. This phenomenon is referred to as pressure distortion (Figure 11). Pressure distortion is a function of the differential pressure, the shape of the face, and the modulus of the face material. The amount of pressure distortion should ideally be equal to the thermal distortion so the net result is that the seal faces remain flat. This is a formidable task since the fluid film is on the order 0.5 micron (20 millionths of an inch).

Current modeling techniques for modeling mechanical seals have evolved significantly in the past decade. While early attempts at modeling seals involved the simple use of finite element analysis (FEA), current techniques involve a coupling of FEA, computational fluid dynamics, and specialized code for mechanical seal. The specialized code defines the unique effects of contacting and noncontacting seal faces, variable fluid film properties, and multiphase films. It also describes the effects of modified faces design that provide addition fluid film support and hydrodynamic lift. These advances have allowed not only more accurate predictions of seal performance but have also allowed mechanical seals to be applied in services not possible in the past. A typical output from such an analysis is shown below (Figure 12).

Face Technologies The first mechanical seals were designed with flat faces to help establish a thin fluid film. The thin fluid film, at the microscopic level, provides an effectively tortuous path to minimize leakage. This aspect of seal design is still the basis for modern mechanical seals. The concept of flat face sealing was also driven by the available manufacturing methods. Since seal faces must be very flat, lapping has historically been the primary method for preparing the sealing surfaces of a seal face. Relatively large-scale features such as hydropads were introduced to improve film formation with some success. It was not until manufacturing methods for micro features such as grooves and waves were

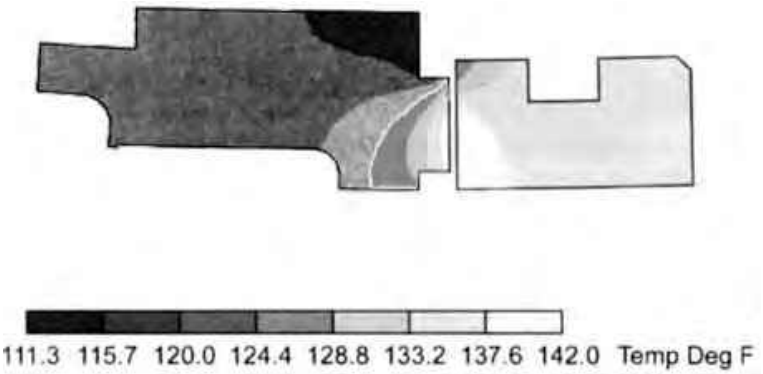


FIGURE 12 Net face distortion

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FIGURE 13 Flat seal face

developed that more effective face designs were available. Current laser-based manufacturing technology allows seal manufacturers to manipulate sealing surfaces within millionths of an inch. This allows engineers to create sealing surfaces that are capable of acting as more than simple flow restrictions. Seal faces can now act as engineered surfaces that will lead to more advanced seal designs, expanded operating windows, and improved reliability.

A review of available face technologies can be broken four categories. Each of these technologies has different capabilities and application requirements:

FLAT FACES Flat seal faces (Figure 13) have smooth featureless surfaces and are manufactured to be flat within about 0.5 micron (20×10^{-6} in). This is achieved by lapping with an appropriate lapping compound such as diamond grit. The surface finish (or roughness) of the face is determined by the grit size of the lapping medium that is normally in the range of 3 to 8 microns. In special cases, lapping with courser compounds, commonly referred to as matte lapping, may be required to improve film formation. Flat faces are measured on an optical flat to insure they are within specification.

FILM ENHANCING FACES There are applications though where the fluid properties and application conditions do not allow for an adequate film formation with flat seal faces. Fortunately, many techniques are available to help promote film formation and improve seal performance (Figure 14). Hydropads are large scallops on the sealing faces that provide a path for fluid to enter into the sealing interface. This can provide better cooling and the formation of a thicker fluid film. Very low amplitude waves are now being used successfully to create a more stable film, reduce heat generation, and reduce friction. This has been especially successful in light hydrocarbons and hot water applications. Face texturing involves creating small features such as dimples into the sealing surface that can improve the film formation of flat faces, but this approach has seen limited use in field applications.

HYDRODYNAMIC FACES Hydrodynamic faces are designed to create high pressures within the contact area of the seal faces. This high pressure increases the opening forces so that the faces achieve total lift-off and operate without any mechanical contact between the seal faces. Since the faces operate on a thick fluid film, leakage will be significantly higher than comparable seals operating with contacting faces. Operating on a thick fluid film allows

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FIGURE 14 Film enhancing seal face

for higher speed operation and lower heat generation. Most hydrodynamic seals are used in gas seal designs where the seals are expected to operate continuously on vapor or gas. Special designs have been built which can operate on liquid, gas, or mixed phase processes (e.g., light hydrocarbon services). Typical designs include spiral grooves, T-grooves, or wavy faces (Figure 15).



FIGURE 15 Hydrodynamic seal face



FIGURE 16 Special design seal face

SPECIAL FACE DESIGNS The advent of micromachining capabilities has lead to a fundamental shift in how seal faces are viewed. Rather than a large flat surface, seal faces have the potential to be engineered devices with expanded functions other simple sealing. An example of such a device uses the surface tension properties of the sealed fluid to create a hermetic, noncontacting seal (Figure 16). Fluid in the seal faces collects on a series of small surface tension bands micro machined into the seal surface. This prevents further movement of the fluid. Face separation is maintained by hydrodynamic waves, which operate on air from outside of the process fluid. The combination of hydrodynamic lift and surface tension has proven successful in applications where other technologies have failed. Other examples of new face technology are being developed and will be offered commercially in the near future.

MATERIALS OF CONSTRUCTION

The many components of a mechanical seal have vastly different functions and material requirements and therefore require a wide range of materials. Over the years, seal manufacturers have standardized on materials that cover a broad range of seal designs and applications.

Face Materials The selection of seal face materials is one of the most critical decisions in a seal design. In most applications, a face combination is selected where one face is a soft face (e.g., carbon/graphite) and the mating face is a hard face (e.g., ceramic based material such as silicon carbide, tungsten carbide, or alumina oxide). This combination of a soft and hard face allows for predictable wearing of the surfaces, a low coefficient of friction, and better tolerance of face contact. This combination is the default selection for mechanical seals. In some application, two hard faces are selected which makes the seal more tolerant of abrasive process fluids.

Seal face materials should ideally meet the following criteria:

- Good tribological properties
- High modulus
- High thermal conductivity

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- Excellent chemical corrosion resistance
- Resistant to impact damage and breakage
- Easy to manufacture
- Relatively low cost

No single material meets all of these criteria but many come very close. The most common seal face materials are listed below. These descriptions are generic terms and may represent several (or dozens of) grades offered by a specific material supplier. The seal OEMs have qualified specific grades that meet the requirements for their seals.

Resin Impregnated Carbon Graphite This is the most common grade of carbon material used in seal faces. It is a mixture of amorphous carbon and carbon graphite, which has been impregnated with a resin or phenolic impregnant. This grade provides the best combination of low friction, wear resistance, and chemical compatibility of the carbon grades. It is limited in aggressive chemical services by the resistance of the impregnant.

Metal Impregnated Carbon Graphite Metal impregnated carbon grades are also a mixture of amorphous carbon and carbon graphite but these have been impregnated with a low melting point metal. The metal impregnant provides high strength, a higher modulus, and greater resistance to blistering. Currently, antimony is the most common metal used in mechanical seals. This grade is used extensively in light hydrocarbon applications.

Reaction Bonded Silicon Carbide Reaction bonded silicon carbide (RBSiC) is formed by reacting molten silicon metal with a mixture of carbon and SiC powder. This results in the SiC being "reaction bonded" together. The material also contains approximately 10% unreacted silicon metal. This grade has excellent hardness, high thermal conductivity, a high modulus, and good tribological properties. RBSiC is the most common grade of SiC used in the refinery industry. Its use in aggressive chemicals is limited by the chemical compatibility of the silicon metal. It should not be used in caustics and aggressive acids.

Self-Sintered Silicon Carbide Self-sintered silicon carbide (SSSiC) is formed by directly sintering silicon carbide together with the use of minute amounts of a sintering aid. The result is a material composed entirely of SiC, which has excellent hardness, high thermal conductivity, a high modulus, and good tribological properties. In addition, it is chemically resistant to almost any fluid likely to be sealed by a mechanical seal. For this reason, it is the preferred material for chemical applications or highly corrosive services. This grade is also known as direct sintered SiC, alpha sintered SiC, or simply sintered SiC.

Nickel Bound Tungsten Carbide Nickel bound tungsten carbide (Ni-WC) is a cemented ceramic with tungsten carbide particles in a metallic nickel matrix. Ni-WC is a very dense, very tough material, which is used in applications where toughness, clamping, or impact resistance is required. While the material is excellent in most services, it can easily heat check in marginally lubricated applications. Ni-WC is chemically limited by the compatibility its nickel binder.

Alumina Oxide Alumina oxide is a white ceramic material manufactured in various purities from around 94% to over 99%. It shares many of the same properties as other ceramics such as high hardness, high modulus, and excellent chemical resistance. It has lower thermal conductivity though which contributes to its poor thermal shock resistance. Its use in mechanical seals is generally limited to lower duty services in chemical or pharmaceutical applications.

Other Specialized Materials

DRY RUNNING CARBONS In some designs, the seal faces are required to run without a supporting fluid film. Dry running grades of carbon are high graphitic materials designed to

operate in a contacting mode with low friction, low heat generation, and low wear. These grades are specifically designed to run with very low contact pressures.

ACID GRADE CARBONS Acid grade carbons are very pure grades of carbon with a very low concentration of impurities and ash found in standard carbon grades. This allows the materials to be applied in extremely aggressive chemical services. There are trade-offs in performance so this grade is used only where the chemical resistance is required.

METALLIC MATERIALS Metallic materials are used for a variety of components in seals including springs, drive mechanisms, sleeves, glands, and fasteners. The materials in these applications are selected primarily for their corrosion resistance and physical properties. The selection of materials for welded metal bellows introduces additional considerations since it must not only be corrosion resistant but must also be easy to weld. High temperature applications also require that the materials retain their physical properties at extreme temperatures. The most common metals are listed below:

316 AUSTENITIC STAINLESS STEELS (UNS S31600) Austenitic stainless steels (especially AISI 316) are the default materials for most mechanical seals. This grade is also available in other variations such as 316L (UNS S31603), which can be easily welded into bellows, and 316Ti (UNS S31635), which is more available in some locations. AISI 316 is resistant to a wide range of chemicals such as waters, hydrocarbons, and hydrogen sulfide. It is resistant to many acids and bases although this varies with the specific chemical, concentration, and temperature. It should be used with caution in high chloride environments due to pitting corrosion.

ALLOY C-276 (UNS N10276) Alloy C-276 is a nickel-molybdenum-chromium alloy with excellent chemical compatibility in a wide range of aggressive applications including many acids and brines. It is used extensively for all seal parts including springs and welded metal bellows.

ALLOY 400 (UNS N04400) Alloy 400 is a copper-nickel alloy with excellent chemical resistance to aggressive acids and bases. Due its high cost, its use is generally limited to hydrofluoric acid applications. This grade can be used in the construction of welded metal bellows. Related grades such as Alloy K-500 (UNS N05500) are used if high strengths are required for components such as fasteners.

ALLOY 20 (UNS N08020) Alloy 20 is a nickel-chromium-molybdenum alloy which has excellent chemical resistance to hot sulfuric acids and many other aggressive chemicals. It can be easily welded and it is used in the construction of welded metal bellows. It is generally only used when more common alloys (such as 316 or alloy C-276) are not suitable for a specific application.

ALLOW 718 (UNS N07718) Alloy 718 is used almost exclusively in the construction of high temperature bellows. Its combination of high temperature capabilities, resistance to sour environments, and welding properties have made it the standard bellows materials for high temperature refinery services.

Low Expansion Alloys Low expansion alloys are a family of iron-nickels alloys that have a coefficient of thermal expansion similar to most seal face materials. This makes it a suitable material for shrink fit constructions in high temperature bellows applications. These alloys generally have poor corrosion resistance so their use is limited to high temperature services.

Elastomeric and Gasketing Materials Mechanical seals have a number of gaskets (commonly referred to as secondary seals) that prevent leakage of process fluids between the components of the seal. These include gaskets at the interface of the sleeve and shaft, gland and seal chamber, and between internal seal components. The gaskets must allow for thermal expansion of the components, conform to irregular surfaces, and be chemically resistant

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to the process fluids. The selection of the gasket materials is generally guided by the chemical and temperature requirements of the application. The selection of gasket materials may also drive the design of seal components due to the gasket installation requirements. Commonly used gasket materials are listed below:

ETHYLENE PROPYLENE (EPR) AND ETHYLENE PROPYLENE DIENE (EPDM) EPR AND EPDM, while widely used in general industry, have limited applications in mechanical seals. They possess excellent resistance to hot water, ketones, and alcohols but are incompatibility with petroleum-based fluids and lubricants.

NITRILE (NBR) Nitrile is an excellent general duty elastomer used in seal applications. It is suitable for most petroleum-based fluids although it has limited compatibility to many aggressive chemicals. Nitrile has the unique ability to be compounded for low temperatures services (down to −54°C or −65°F) so it is often used in cryogenic applications.

FLUOROELASTOMERS (FKM) Fluoroelastomers provide excellent chemical resistance and good high temperature capabilities. As such, they have become the standard elastomer grade for general duty mechanical seals. They have excellent chemical resistance to hydrocarbons, acids, and water. They are not recommended for ketones, amines, and hot water services.

PERFLUOROELASTOMERS (FFKM) Perfluoroelastomers are highly fluorinated elastomers with an excellent chemical resistance and high temperature properties. This includes applications in very aggressive acids, bases, and amines. The actual chemical resistance and temperature limits for FFKMs are very dependant on the specific compound. Some compounds are intended for broad general duty use: others are intended for specific applications. The seal OEM or elastomer manufacturer can provide guidance in selecting specific grades.

FLEXIBLE GRAPHITE Flexible graphite has near universal chemical resistance as a gasket material. It is also stable at extremely high temperatures. This has made it the standard selection for high temperature seal applications. Flexible graphite has almost no resiliency so it must be mechanical clamped or hydraulically loaded against its mating surface.

POLYTETRAFLUOROETHYLENE (PTFE) Polytetrafluoroethylene is a fully fluorinated material commonly used in a variety of gaskets. It has near universal chemical compatibility, which makes it useful for aggressive services or batch processes. It is a hard, inflexible material that requires special gasket groove designs and installation practices.

CLASSIFICATION OF MECHNICAL SEALS

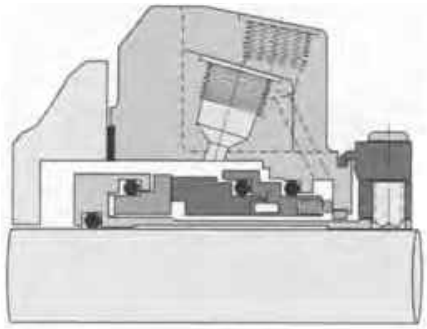
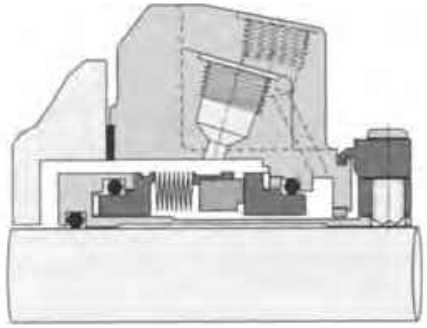
With the wide range of seal designs and applications available, it is natural to categorize the options to assist in referring to specific features or designs. There are many different ways to categorize seals and these are fairly consistent among seal OEMs. Common categories include basic design, orientation, and arrangement. Each of the options in these categories has advantages and disadvantages that may help the user in selecting the correct seal.

Classification by API 682 Seal Type API 682, "Pumps—Shaft Sealing Systems for Centrifugal and Rotary Pumps" is a mechanical seal standard published by the American Petroleum Institute. This standard, first published in 1994, has become the most popular standard for mechanical seals. API 682 was written primarily by end users with input from seal and pump OEMs. While the standard was originally intended solely for refinery applications, later revisions expanded the scope to include chemical duty applications. API 682 is also published as ISO 21049.

API 682 has introduced several definitions to help categorize seal by design features. Seal Type refers to a seal definition that includes the operating window for the seal, basic design features, and materials of construction. Each Seal Type can be applied to different arrangements and orientations.

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TABLE 3 Classification by design and orientation

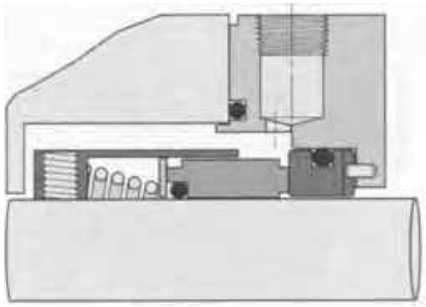
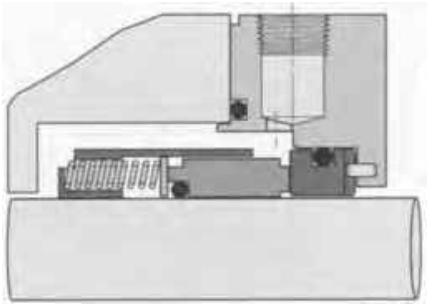
Pusher seal  **FIGURE 17**	Definition: Seal with a flexibly mounted seal face supported by a dynamic gasket, which is allowed to move axially to compensate for shaft motion or face wear	Advantages: Simple design, flexible material selection, can be designed for high pressures Disadvantages: Requires dynamic gasket, subject to hang-up or fretting under gasket
Bellows seal  **FIGURE 18**	Definition: A seal with convolutions of metal or elastomeric material providing sealing of process fluids and mechanical loading of the seal faces.	Advantages: Eliminates dynamic gasket, can be constructed without elastomeric gaskets, can be designed for high temperatures Disadvantages: Limited material selection for bellows, lower pressure rating than pusher seals, thin metal diaphragms subject to erosive wear

(*Continued*)

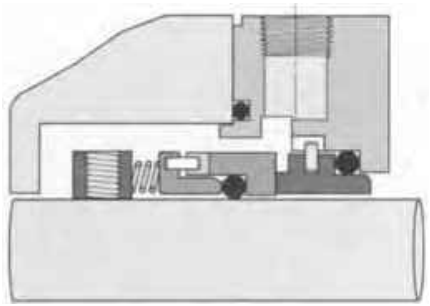
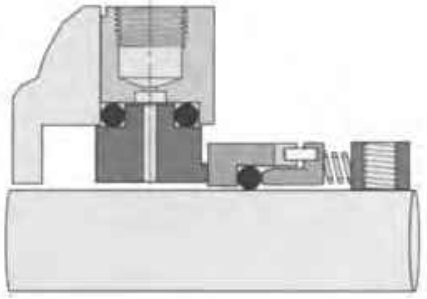
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TABLE 3 Classification by design and orientation (*Continued*)

Single spring pusher  **FIGURE 19**	Definition: A pusher seal design where the mechanical load is provided by a single coil spring.	Advantages: Large wire cross section more tolerant of corrosion and erosion, low spring rate more tolerant of axial shaft motion Disadvantages: Requires more axial space, can provide uneven loading of seal faces
Multiple spring pusher  **FIGURE 20**	Definition: A pusher seal design where the mechanical load is provided by multiple coil springs.	Advantages: Compact package, more uniform face loading Disadvantages: Small wires subject to erosion and less tolerant of corrosion

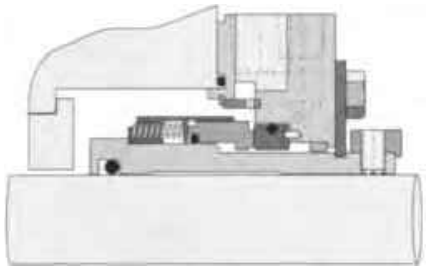
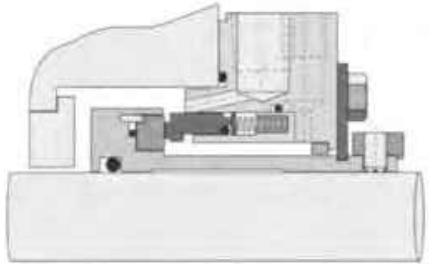
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Internally mounted seal  FIGURE 21	Definition: Seal is located inside the seal chamber	Advantages: Pressurized from OD of seal, better circulation of fluid around seal faces, allows for bushing and drain in seal gland to contain leakage Disadvantages: More components exposed to process fluid
Externally mounted seal  FIGURE 22	Definition: Seal is located outside of seal chamber	Advantages: Few seal components exposed to process fluid, requires no space in seal chamber Disadvantages: Faces are pressurized on the ID, leakage can not be contained, low pressure rating

(*Continued*)

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TABLE 3 Classification by design and orientation (*Continued*)

Rotating flexible element  **FIGURE 23**	Definition: The flexible element rotates with the shaft	Advantages: Good circulation around spring components, tends to be self-cleaning Disadvantages: Potential for erosion of seal components, less tolerant of misalignment
Stationary flexible elements  **FIGURE 24**	Definition: The flexible element is stationary with the gland	Advantages: More tolerant of misalignments, able to operate at higher shaft speeds Disadvantages: Requires more radial space, may be more difficult to design adequate flush

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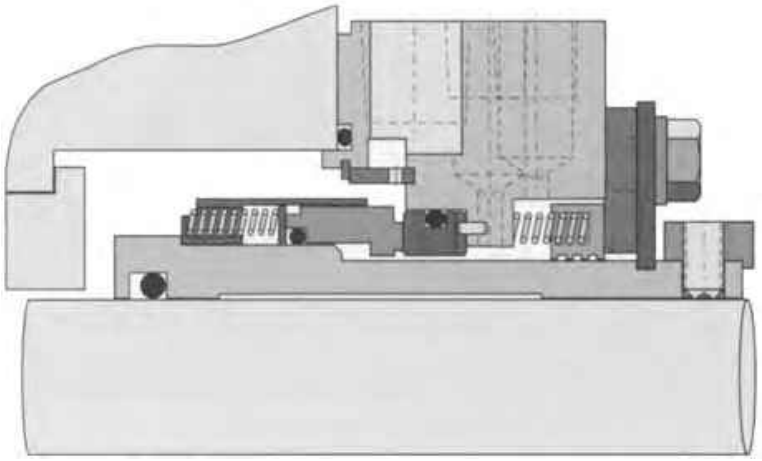


FIGURE 25 Type A seal

TYPE A A Type A seal (Figure 25) is an inside mounted, balanced pusher seal. The default design has a rotating flexible element although a stationary flexible element is optional. The default design utilizes multiple springs (in alloy C-276) although there is an option for a single coil spring (316 SS). The materials of construction are carbon versus silicon carbide faces, FKM gaskets, and a 316 SS sleeve and gland. The operating temperature range for the seal is −40°C to 176°C (−40°F to 350°F). The Type A seal is considered the default design for most applications within API 662.

TYPE B A Type B seal (Figure 26) is an inside mounted, balanced general-duty bellows seals. The default design has a rotating bellows assembly although a stationary bellows is optional. The default materials of constructions are alloy C-276 bellows diaphragms, carbon versus silicon carbide faces, FKM gaskets, and a 316 SS sleeve and gland. The operating temperature range for the seal is −40°C to 176°C (−40°F to 350°F).

TYPE C A Type C seal (Figure 27) is an inside mounted, balanced high-temperature bellows seal. The default design has a stationary flexible element although a rotating flexible element is optional. The default materials of constructions are alloy 718 bellows diaphragms, carbon versus silicon carbide faces, flexible graphite gaskets, and a 316 SS sleeve and gland. There is also a requirement for a bronze anticoke device and a floating carbon throttle bushing. The operating temperature range for the seal is −40°C to 400°C (−40°F to 750°F).

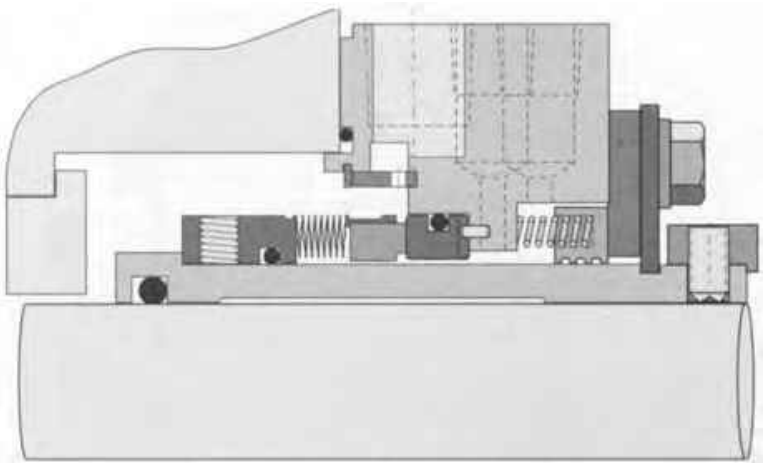


FIGURE 26 Type B seal

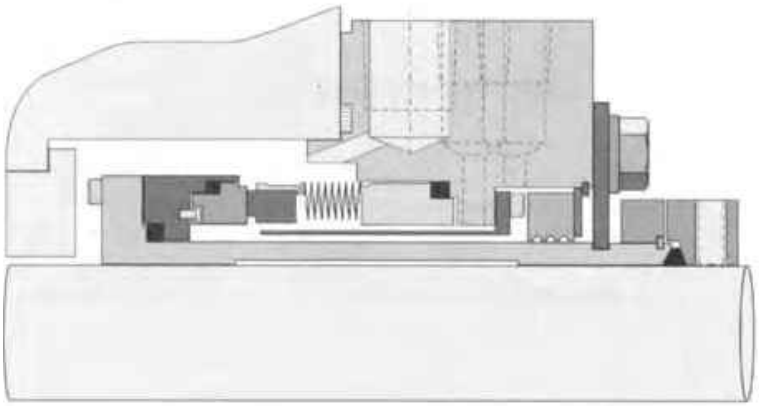


FIGURE 27 Type C seal

Classification by Arrangements and Configurations Mechanical seals are often applied where only one mechanical seal is installed into the seal chamber and the seal operates on the pump process fluid. While this is acceptable for many installations there may be conditions where a single seal cannot be used. This may include applications where the pump fluid is unreliable or is not suitable for a mechanical seal. It can also include applications where leakage from a failed seal would present an unacceptable environmental or safety hazard. To address these issues, arrangements are available where multiple seals are used. The most common seal arrangements are listed below.

SINGLE SEALS Single seals (Figure 28) consist of one set of seal faces. It is the simplest design and, in general, requires minimal outside support systems or auxiliary equipment. The majority of single seals operate on the process fluid although an outside seal flush may be utilized. Leakage past a single seal will go to an atmospheric drain or vaporize and go directly into the atmosphere. If the application contains fluids which are regulated emissions, carcinogens, above the auto-ignition temperature, or present other hazards, the user should consider the consequences of both normal leakage and leakage under failure conditions. This arrangement is defined as an Arrangement 1 seal in API 682.

DUAL UNPRESSURIZED SEALS Dual unpressurized seals (Figure 29) are seal assemblies designed to contain two mechanical seals in series. In this arrangement, the inner (or primary) seal operates on the process fluid under seal chamber process conditions. The seal outer (or secondary) seal is intended to operate as a backup seal under low-pressure conditions. Under normal operation, seal leakage past the inner seal would collect between the two seals and be routed to a flare or recovery system. In the event of a failure of the

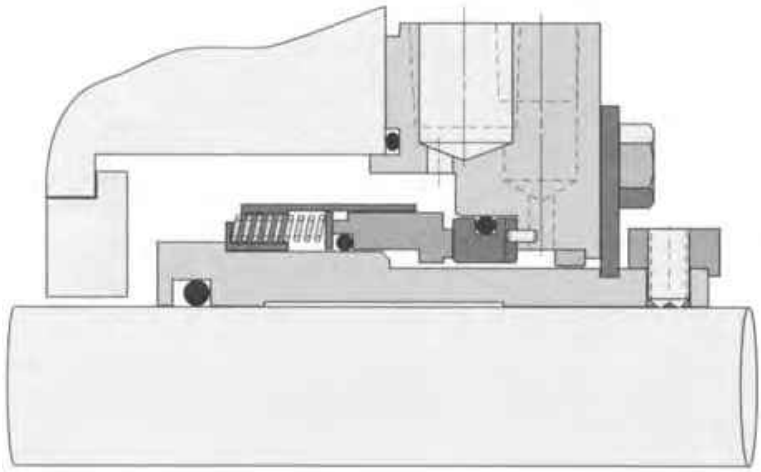


FIGURE 28 Single seal

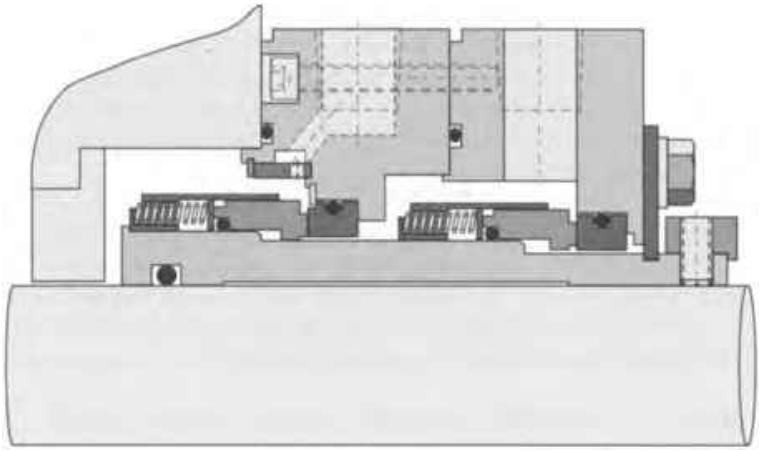


FIGURE 29 Dual unpressurized seal

inner seal, the outer seal could contain the process fluid under full seal chamber pressure. In either scenario, leakage of process fluid to atmosphere would be minimal.

The cavity between the primary and secondary seal may be filled with a liquid or operated dry. If the cavity is filled with a liquid, the arrangement is referred to as a dual unpressurized seal with a liquid buffer fluid. The buffer fluid serves the dual purpose of providing the seal environment for the outer seal and a collection point for inner seal leakage. The outer seal is a standard liquid mechanical seal with the same rating as the inner seal. During operation, the buffer fluid is circulated from the secondary seal cavity to a reservoir that provides cooling and monitoring of the buffer fluid level.

If the cavity between the inner and outer seal is operated without a liquid, the arrangement is referred to as a dual unpressurized seal with a containment seal. A containment seal is a dry-running backup seal specifically designed to operate without liquid lubrication. The cavity between the inner and outer seal is referred to as the containment seal cavity. This cavity must be maintained under low-pressure conditions (less than 0.7 bar or 10 psi). Leakage from the inner seal is routed to the flare or a recovery system. In the event of failure of the inner seal, the outer seal could contain the process fluid under full seal chamber pressure. Leakage of process fluid to atmosphere would be minimal under normal seal operation or in the event of an inner seal failure. This arrangement is defined as an Arrangement 2 seal in API 682.

DUAL PRESSURIZED SEALS Dual pressurized seals (Figure 30) are used when, for some circumstances, it is not desirable to have any process leakage released to atmosphere. There may be other applications where the process fluid is unsuitable or too unreliable for mechanical seals. In these cases, a dual pressurized seal arrangement is used. A dual pressurized

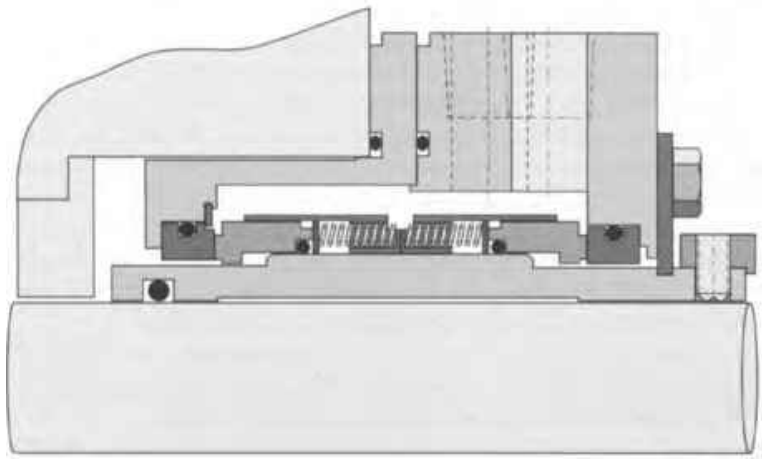


FIGURE 30 Dual pressurized seal

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seal consists of two sets of seal faces arranged in a face-to-back, back-to-back, or face-to-face orientation. The cavity between the inner and outer seal is referred to as the barrier fluid cavity. The barrier fluid is maintained at a pressure higher than found in the seal chamber that effectively causes both the inner and outer seal to operate on a fluid film of the barrier fluid. The pressurization also prevents any process fluid from entering the barrier fluid cavity and leaking to atmosphere.

If the barrier fluid is a liquid, it will be circulated from the seal to an external reservoir (or accumulator) and back to the barrier cavity. The support system will not only provide pressurization of the barrier fluid but also cooling and monitoring of the barrier fluid liquid level. If the barrier fluid is a gas, the seal is often referred to as a dual gas seal. The barrier gas is provided by a gas control panel that cleans the gas, regulates the pressure, and monitors the flow rate to the seal. This arrangement is defined as an Arrangement 3 seal in API 682.

Special Applications There are other special applications that do not fit neatly into the above arrangements. These would include dual seals designed to pressure share (or breakdown) the process fluid pressure so the application can be more reliably sealed. They may also include seal arrangements with three or more sets of seal faces. While these examples are not common, they do illustrate that special seal arrangements can be developed to meet unusual application requirements.

CONTROL OF SEAL ENVIRONMENT

Mechanical seals operate most reliably when the seal chamber conditions promote a clean, cool fluid film between the faces. In practice, the process environment is often less than ideal. Fortunately, methods of improving the seal environment are available which can circulate, clean, and cool the seal chamber fluid. Seals are often installed in equipment that was designed for packing. This can limit the selection for sealing options or provide poor fluid circulation around the seal. There are seal chamber designs and pump retrofits available which can provide the required space for reliable seal performance. Providing a good environment for the seal is as critical to success as selecting the correct seal.

Seal Chamber Options Historically packing was used to provide the seal in centrifugal and rotary pumps. Packing is generally manufactured in rings with a square cross section ranging from approximately 8 mm (5/16 in) for small chemical duty pumps to 20 mm (3/4 in) for larger heavy-duty pumps. Unfortunately, these small cross sections reflect the small radial clearances found in a typical stuffing box. In addition, the effectiveness of the packing arrangements also depended upon the number of rings used in the installation. On high-pressure pumps, this resulted in many rings of packing and a very deep stuffing box. Packing not only provided sealing for the process fluid but also provided mechanical support for the shaft and was sometimes critical for proper control of shaft deflections. Simply removing packing and installing a mechanical seal often resulted in poor seal performance.

To provide a better environment for mechanical seals, pump manufacturers began offering seal chamber designs specifically to accommodate larger seals and provide better circulation of fluids in the seal chamber. One class of pumps (ASME B73 series) increased the radial clearance from 8 mm to 11 mm (5/16 in to 7/16 in) in their stuffing boxes to 19 mm to 25 mm (3/4 in to 1 in) in their seal chambers. This allowed for the option to install a wider range of seals and provide a better environment. Lab testing has shown that seal chambers provide a better control of mechanical seal temperatures and lower overall temperatures compared to comparable stuffing boxes. Many end users recognize this improvement and specify only seal chambers when new pumps are purchased with mechanical seals. Unfortunately, many end users continue to use stuffing boxes on sealed pumps previously installed in the field even though upgrade options are available.

Seal chamber designs are defined in pump standard such as API 610/ISO 13709 and ASME B73. These designs can be broken into three main categories: cylindrical bore, tapered bore, and tapered bore with flow modifiers (Figure 31).

Cylindrical bore seal chambers maintain the same basic shape as a stuffing box although the radial clearance between the shaft and seal chamber bore has been increased. This design is used for general seal applications and works well with most piping plans. Some

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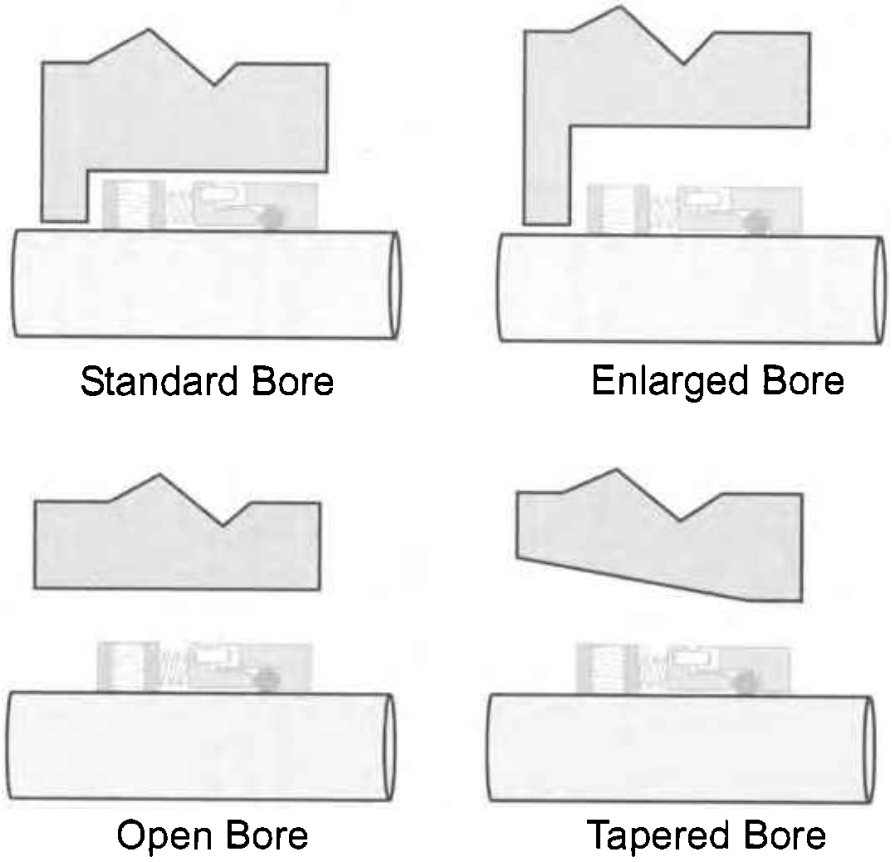


FIGURE 31 Comparison of seal chambers

piping plans require isolation of the seal chamber from the pump process. This may be due to cooling the process fluid (e.g., Plan 21 or 23), cleaning the process fluid (e.g., Plan 31), or introducing an external fluid (e.g., Plan 32). In these cases, a cylindrical bore seal chamber with a throat bushing is required to get the full benefit of the piping plan.

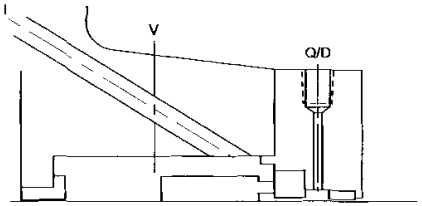
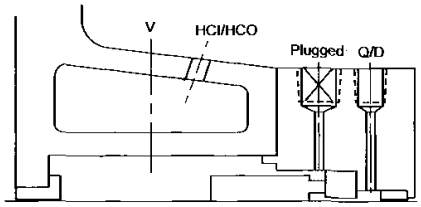
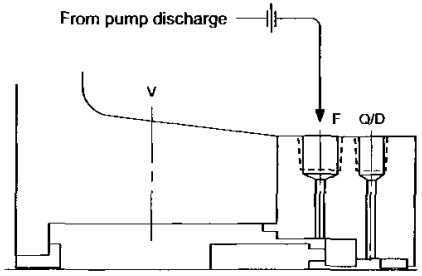
Many pumps are applied where the natural circulation of the fluid in the seal chamber provides all of the cooling requirements for the mechanical seal. To assist in this circulation, seal chambers have been designed which are tapered from the face of the seal chamber back to the impeller. This design is appropriately referred to as a taper bore seal chamber. This design provides good circulation around the mechanical seal and should be used when no external piping plan is used (Plan 02). Tapered bore seal chambers are also inherently self-venting and self-draining.

While tapered bore seal chambers provided good circulation, lab testing and field experience showed that the circulation flows were not as simple as first thought. Areas inside the seal chamber could collect solid particles from the process fluids. To assist in removing these particles from the seal chamber, the bore of the taper was modified to provide protrusion to redirect the rotational flow of the fluid into axial flow. This eliminated any stagnant areas and effectively cleaned the fluid in the seal chamber. These seal chambers are generically referred to as tapered bore seal chambers with flow modifiers and they are available from a number of pump OEMs.

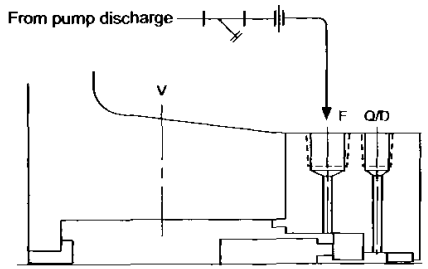
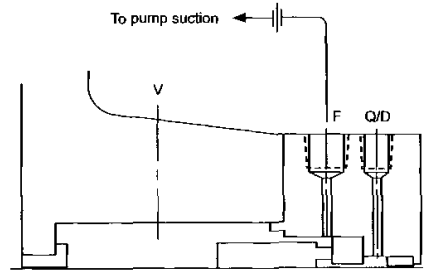
Seal Piping Plans To further improve the environment around the seal, fluid is circulated through the seal chamber. The condition of this fluid may be modified by cleaning and cooling the fluid or by increasing or decreasing the seal chamber pressure. Dual pressurized and dual unpressurized seals have further requirements for handling the barrier fluid or buffer fluid, respectively. There are also considerations for modifying the environment on the atmospheric side of the seal or detecting atmospheric side leakage. All of these methods of improving the seal environment have been captured in standardized seal piping plans. These have been widely published and are generally maintained in API 682 and ASME B73. A list of the current piping is shown below:

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TABLE 4 Seal piping plans

Plan 01  **FIGURE 32**	Definition: Internal recirculation from a high pressure region of the pump to the seal chamber	Advantages: Built-in from factory Disadvantages: Flow is not easy to control or modify, injection is generally not directed at the seal faces, difficult to troubleshoot
Plan 02  **FIGURE 33**	Definition: Dead-end seal chamber with no flushcirculation. Taper bore seal chambers are generally recommended to improve seal chamber circulation	Advantages: Simple system requiring no external piping or auxiliaries Disadvantages: Generally limited to lower pressure, lower speed applications
Plan 11  **FIGURE 34**	Definition: Circulation of process fluid from the pump discharge (or other high pressure region of the pump) through an orifice into the seal chamber	Advantages: Provides flow to carry away seal generated heat, can be used to increase seal chamber pressure, easy to modify flow rate by changing orifice Disadvantages: Orifice can become plugged restricting circulation, process fluid must not solidify under atmospheric temperatures or heat tracing may be required

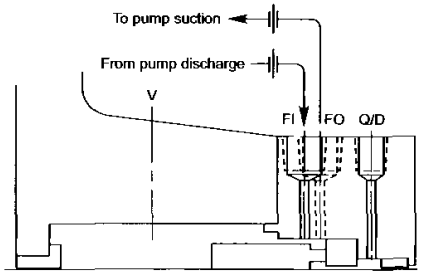
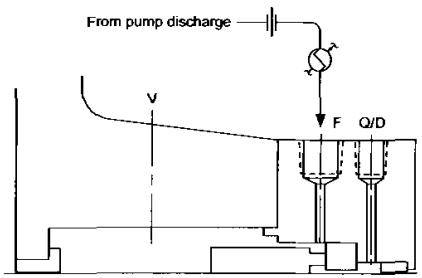
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Plan 12  **FIGURE 35**	Definition: Circulation of process fluid from the pump discharge (or other high pressure region of the pump) through a strainer and an orifice into the seal chamber	Advantages: Provides screening of process fluid to prevent orifice plugging, capable of keep large debris out of seal chamber Disadvantages: Strainer can become plugged restricting circulation, not recommended for new applications
Plan 13  **FIGURE 36**	Definition: Circulation from seal chamber through an orifice to the pump suction or other low-pressure region of the pump	Advantages: Used in vertical pumps where seal chamber is near discharge pressure, can be used to reduce pressure in seal chamber, can be used to provide clean flush in some pump constructions Disadvantages: May not be useable in some pump constructions due to lack of pressure differential, orifice can become plugged restricting circulation

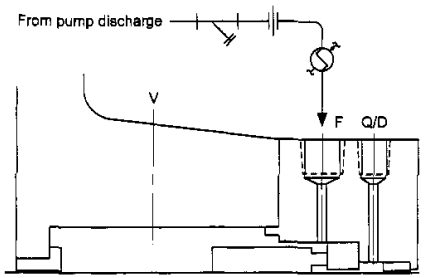
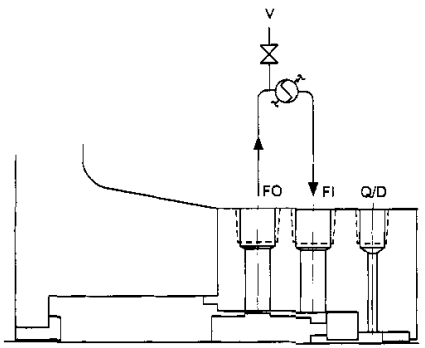
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TABLE 4 Seal piping plans (*Continued*)

Plan 14  **FIGURE 37**	Definition: Combination of Plan 11 and Plan 13, fluid circulation from pump discharge to seal chamber and from seal chamber to pump suction	Advantages: Provides continual flow into and out of the seal chamber Disadvantages: Difficult to determine resulting pressure in seal chamber
Plan 21  **FIGURE 38**	Definition: Circulation of process fluid from the pump discharge (or other high pressure region of the pump) through an orifice and seal cooler into the seal chamber	Advantages: Reduces the temperature of injected fluid which reduces its vapor pressure and increases its viscosity Disadvantages: Reinjection of cold fluid back into pump/system, inefficient method of cooling fluid, will cause fouling of water-cooled seal coolers in high temperature applications

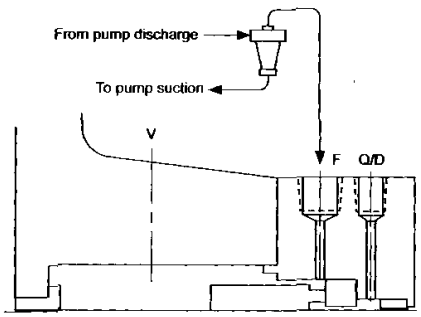
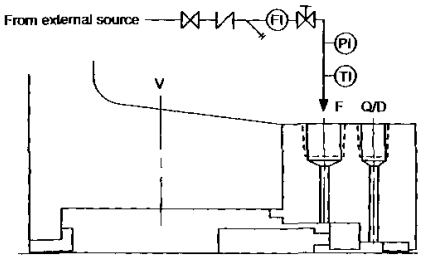
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Plan 22  **FIGURE 39**	Definition: Circulation of process fluid from the pump discharge (or other high pressure region of the pump) through a strainer, an orifice, and seal cooler into the seal chamber	Advantages: Reduces the temperature of injected fluid which reduces its vapor pressure and increases its viscosity, provides screening of process fluid to prevent orifice plugging, capable of keep large debris out of seal chamber Disadvantages: Reinjection of cold fluid back into pump/system, inefficient method of cooling fluid, will cause fouling of water-cooled seal coolers in high temperature applications, strainer can become plugged restricting circulation, not recommended for new applications
Plan 23  **FIGURE 40**	Definition: Circulation of process fluid from the seal chamber through a seal cooler and back into the seal chamber	Advantages: Provides efficient cooling with minimum size seal cooler and minimum cooling water requirements, reduces the temperature of circulated fluid which reduces its vapor pressure and increases its viscosity, does not inject cooled process fluid back into pump/system Disadvantages: Requires careful venting piping prior to start-up

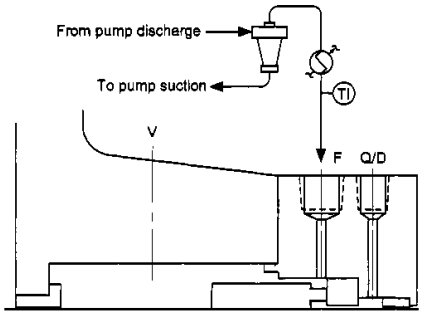
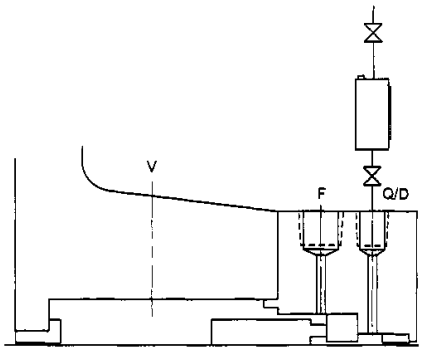
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TABLE 4 Seal piping plans (*Continued*)

Plan 31  **FIGURE 41**	Definition: Circulation of process fluid from the pump discharge (or other high pressure region of the pump) through a cyclone separator with clean fluid directed into the seal chamber and dirty fluid directed into the pump suction	Advantages: Provides cleaner injection fluid to seal chamber Disadvantages: Not effective in many applications, may clog or plug in operation, requires similar pressure in seal chamber and cyclone return line to pump suction
Plan 32  **FIGURE 42**	Definition: Injection of flush fluid from an outside source into seal chamber	Advantages: Flush fluid is independent of pump process Disadvantages: Seal reliability depends upon reliability of flush source, flush must be compatible with process fluid and pump conditions (especially temperature), may be expensive due to cost in flush fluid

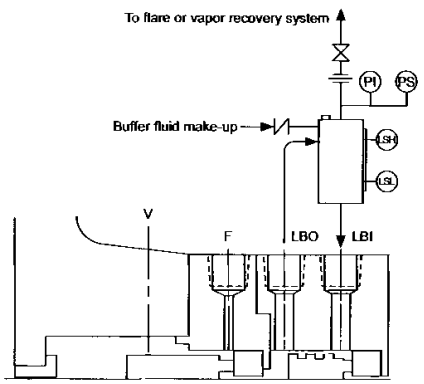
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Plan 41  **FIGURE 43**	Definition: Circulation of process fluid from the pump discharge (or other high pressure region of the pump) through a cyclone separator and seal cooler into the seal chamber	Advantages: Provides cleaner injection fluid to seal chamber, reduces the temperature of injected fluid which reduces its vapor pressure and increases its viscosity Disadvantages: Not effective in many applications, may clog or plug in operation, requires similar pressure in seal chamber and cyclone return line to pump suction, reinjection of cold fluid back into pump/system, inefficient method of cooling fluid, will cause fouling of water-cooled seal coolers in high temperature applications, strainer can become plugged restricting circulation, not recommended for new applications
Plan 51  **FIGURE 44**	Definition: External reservoir providing a dead-end blanket on the atmospheric side of the seal	Advantages: Can modify the atmospheric side of the seal to prevent icing or crystallization Disadvantages: Useful only in vertical pumps

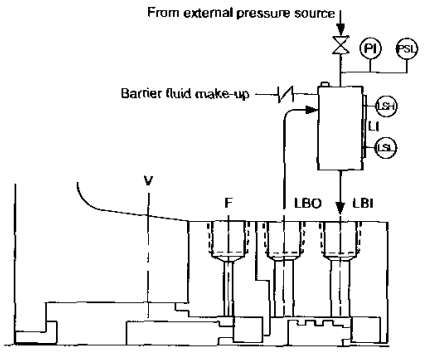
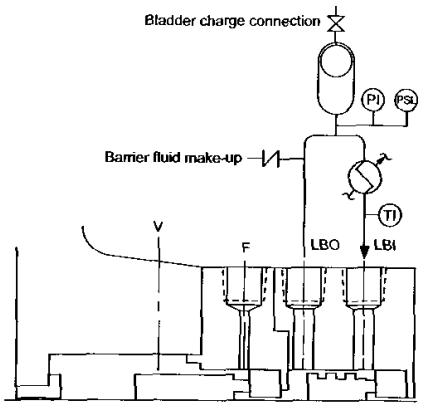
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TABLE 4 Seal piping plans (*Continued*)

Plan 52  **FIGURE 45**	Definition: External reservoir supplying buffer fluid to a dual unpressurized seal assembly	Advantages: Captures leakage from inner seal, buffer fluid provides additional cooling to inner seal, operated at low pressure, directs inner seal leakage to flare or vapor recovery system, minimizes process leakage to atmosphere Disadvantages: Requires external reservoir, buffer fluid, and cooling water (in many cases), requires monitoring of buffer fluid level
Plan 53 Plan 53 represents a series of piping plans that provide a pressurized liquid barrier fluid to a dual pressurized seal assembly. The system not only provides pressurization but also cooling of the barrier fluid as well as instrumentation to monitor system pressure and barrier fluid levels. In the past, a Plan 53 was assumed to consist of a reservoir pressurized with an inert gas. Variations started to appear which were informally designated as a Plan 53 Modified. To help provide a better definition of the variations and allow users to specify them, three versions of Plan 53 was defined		

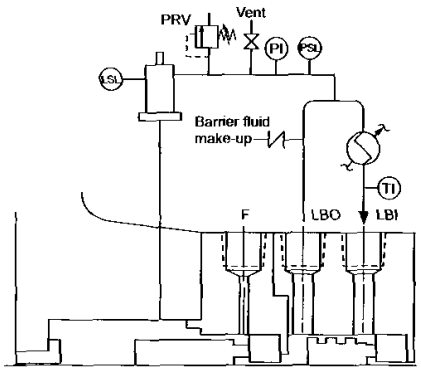
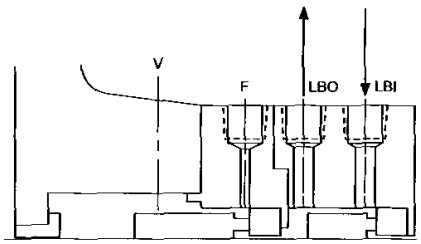
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Plan 53A  **FIGURE 46**	Definition: Pressurized external reservoir providing barrier fluid to a dual pressurized seal, pressurization is provided by inert blanket of gas (usually Nitrogen) controlled by a pressure regulator, barrier fluid level is monitored through sight glass or instrumentation.	Advantages: Fairly simple pressurization scheme, easy monitoring of fluid level changes Disadvantages: Interface between pressurization gas and barrier fluid promotes gas absorption, generally limited to less than 14 bar (200 psi), requires continuous supply of pressurization gas
Plan 53B  **FIGURE 47**	Definition: Pressurized external system providing barrier fluid to a dual pressurized seal, pressurization is provided by a pre-charged bladder accumulator, barrier fluid "level" is monitored by system pressure.	Advantages: Suitable of high pressure applications, does not require continuous external pressurization, can be operated in remote areas Disadvantages: Barrier fluid pressure varies with seal leakage, refill rates determined by size of accumulator

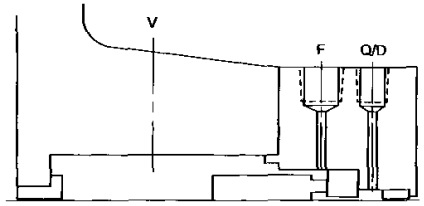
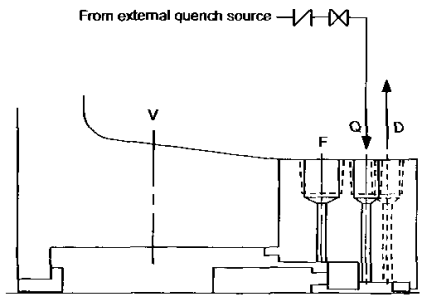
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TABLE 4 Seal piping plans (*Continued*)

Plan 53C  **FIGURE 48**	Definition: Pressurized external system providing barrier fluid to a dual pressurized seal, pressurization is provided by a piston transmitter which increases the barrier fluid pressure by a fixed ratio, barrier fluid level is monitored by piston extension	Advantages: Automatically provides correct pressurization of barrier system, handles process upsets or pressure spikes, can operate at high pressures Disadvantages: Limited volume in transmitter, transmitter is exposed to process fluid, limited material selection for transmitters, not suitable for dirty applications or where process fluid may freeze, crystallize, or plate out
Plan 54  **FIGURE 49**	Definition: Pressurized barrier fluid provided by an outside source, source may be a closed circulation system or a once-through process stream	Advantages: Easy control of flush flow rate, ability to handle high pressures and high heat loads Disadvantages: Seal reliability depends upon reliability of barrier fluid system or source, circulation systems may become complex

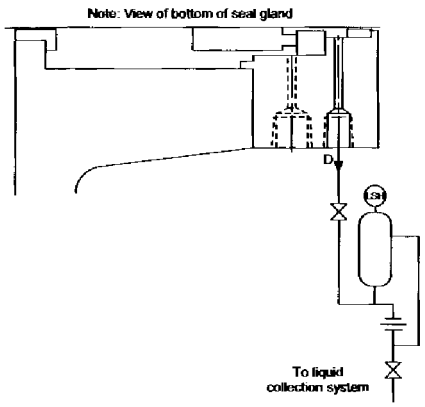
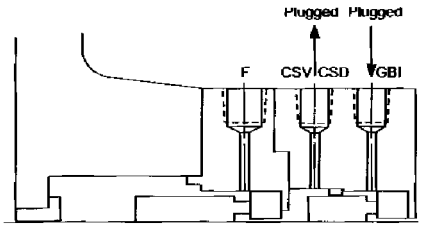
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Plan 61  **FIGURE 50**	Definition: Atmospheric side connections (quench and drain ports) plugged for purchaser's use	Advantages: Ports available for future use Disadvantages: Ports may be left plugged and provide no benefits
Plan 62  **FIGURE 51**	Definition: External quench provided to the atmospheric side of seal (usually steam, nitrogen, or water)	Advantages: Modify the environment on the atmospheric side of the seal, minimize coking of hot process fluids, prevent atmospheric crystallization of some process fluids (e.g., caustic) Disadvantages: Wet steam quenches create problems in hot services, leakage of liquid quench to atmospheric side of seal

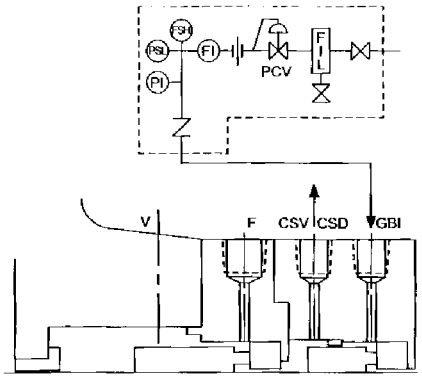
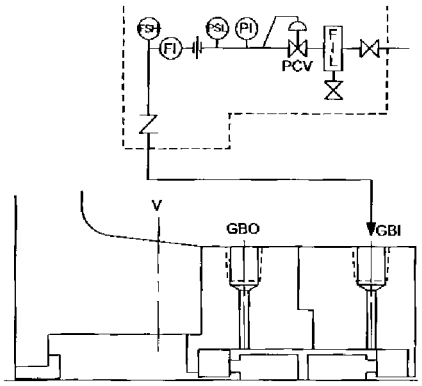
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TABLE 4 Seal piping plans (*Continued*)

Plan 65  **FIGURE 52**	Definition: External drain from atmospheric side of seal to monitor seal leakage	Advantages: Provide remote alarm of leakage from single seal Disadvantages: Requires a high leak rate to alarm system, leakage which will solidify or crystallize under atmospheric conditions may require system to be heat traced
Plan 71  **FIGURE 53**	Definition: Connection to containment seal cavity plugged for purchaser's use	Advantages: Ports available for future use Disadvantages: Ports may be left plugged and provide no benefits

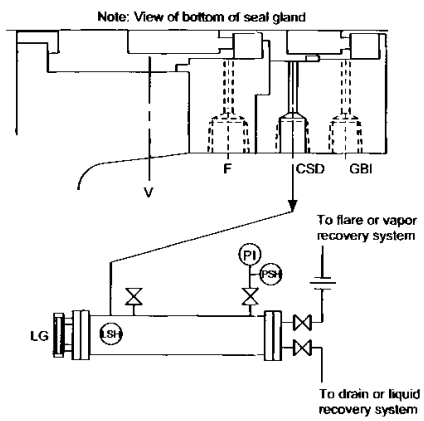
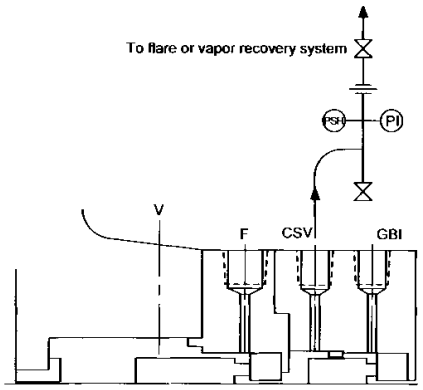
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Plan 72  **FIGURE 54**	Definition: Supply of low-pressure buffer gas to the containment seal cavity of a dual unpressurized seal, requires use of Plan 72 or 74.	Advantages: Dilute and remove process fluid from containment seal cavity, minimize process emissions to atmosphere Disadvantages: Requires external buffer gas and gas control panel
Plan 74  **FIGURE 55**	Definition: Supply of high-pressure barrier gas to the barrier cavity of a dual pressurized seal	Advantages: Provide environment for dual gas seals, prevent process leakage to atmosphere Disadvantages: Requires external buffer gas and gas control panel, pressure must be maintained even during standby conditions

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TABLE 4 Seal piping plans (*Continued*)

Plan 75  **FIGURE 56**	Definition: External drain from containment seal cavity to collect and monitor liquid or mixed phase process leakage from inner seal, may be used alone or with a Plan 72	Advantages: Allows monitoring of both liquid and vapor phase leakage, provides alarm for inner seal failure, directs process leakage to flare, drain, or recovery system Disadvantages: Leakage which will solidify or crystallize under atmospheric conditions may require system to be heat traced
Plan 76  **FIGURE 57**	Definition: External venting of containment seal cavity to collect and monitor vapor phase process leakage from the inner seal, may be used alone or with a Plan 72	Advantages: Allows monitoring of both liquid and vapor phase leakage, provides alarm for inner seal failure, directs process leakage to flare or vapor recovery system Disadvantages: Requires a high leak rate to alarm system

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Other Piping Plans While the list of defined plans covers most applications, there will always be applications that require modifications to defined plans or require new methods of piping the seal system. Historically, users and seal OEMs would use terms like Plan 53 modified to document that the installed plan is not according to the stand definition. Other users and seal OEMs have come up with additional plan numbers that sound similar to the standard plans. There is great danger in this since there is no accepted definition and different users may use the same description to define different plans. If the plan offered is not in accordance with the standard plans defined in API 682 or ASME B73, they should be accompanied by a complete description of the function and operation of the plan as well as a P&ID to define the offering.

Auxiliary Equipment Several of the standard piping plans require the use of auxiliary equipment. This equipment includes reservoirs, seal coolers, gas control panels, cyclone separators, and other devices, which help contain and condition the fluid in the piping plan. There are no standardized requirements for these auxiliaries other than found in API 682. In many cases, individual end users have company specifications that will need to be followed.

RESERVOIRS Reservoirs (Figure 58) are used in Plan 52 and Plan 53 systems. The purpose of a reservoir is to maintain a supply of buffer or barrier fluid for the seal plan and provide temperature control through the use of internal cooling coils if required. Reservoirs are commonly supplied with capacities from 8 L (2 gal) for small low duty pumps to as large as 76 L (20 gal) for large high temperature applications. The most common sizes have 12 L (3 gal) and 20 L (5 gal) capacities. These are commonly supplied with features such as site glasses to monitor fluid levels, pressure indicators to locally monitor pressure, and level switches to indicate a change in level.

Reservoirs are constructed from pipe and are available with either fixed or flanged ends. The pipe size and wall thickness dictates the pressure rating for the reservoir. API 682 specifies that 12 L (3 gal) reservoirs be constructed from DN 150 (6 in) schedule 40 pipe and 20 L (5 gal) reservoirs use DN 200 (8 in) schedule 40 pipe. Other sizes and wall thickness may be required for specific applications. API 682 also provides guidance on the sizing and location for all connections and ports to the reservoir.

SEAL COOLERS Seal coolers (Figure 59) are sometimes referred to as heat exchangers. While they do certainly serve the same function, seal coolers are designed to provide effective cooling to low volumes of fluid while offering low resistance to flow as required in many piping plans. Many plants require that heat exchangers comply with standards with such as TEMA, which is inappropriate for the size coolers required for seal systems. Seal coolers are therefore specifically designed for the requirements of mechanical seal systems. Seal coolers are most commonly used in Plan 21 and Plan 23 piping plans.

Seal coolers are required to have a low-pressure drop to insure adequate circulation in Plan 23 systems. Seal coolers should also be designed and mounted to promote thermosyphoning in standby services. The systems must also be ventable and drainable on both the process and cooling water side.

GAS CONTROL PANELS Dual gas seals (Figure 60) require the supply of a barrier gas to pressurize the seal assembly (Plan 74). Some containment seals will also use a buffer gas sweep to improve seal performance (Plan 72). In each of these cases, the gas provided must be conditioned, controlled, and monitored to insure successful operation. Gas seal control panels are designed to serve these functions. A typical panel will include a coalescing filter, pressure regulator, flow meter, flow switch, pressure switch, pressure indicator, and check valve. The list of components may vary depending upon the installation needs.

Piping Practices While piping plans are described in schematics and P&IDs, the success of a plan in an actual application is a function of how the plan is designed and constructed. This includes the size and type of interconnecting piping or tube, the routing of the piping, the use of fittings, and the mounting of auxiliary equipment. In many piping

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FIGURE 58 Typical seal reservoir

plans, the flow is generated by the differential pressure in the pump. In other cases, system flows are generated by pumping rings on the seal so the system design becomes critical and must reduce friction losses. There are general guidelines that can help guide the user in creating a successful application.

The most critical piping systems involve the connection of the seal to an auxiliary such as a reservoir or seal cooler. Piping should be designed to minimize the length of the piping runs. This requires that auxiliary be located close to the seal and at the correct elevation. The actual piping or tubing should run directly from the seal to the auxiliary with a minimum of bends, elbows, or fittings. The piping should be continuous sloping upward from the seal to the auxiliary to aid in venting or draining of the system. Piping or tubing sizes should be large to minimize friction losses. An example of good and bad piping systems is shown in Figure 61.



FIGURE 59 Typical seal cooler

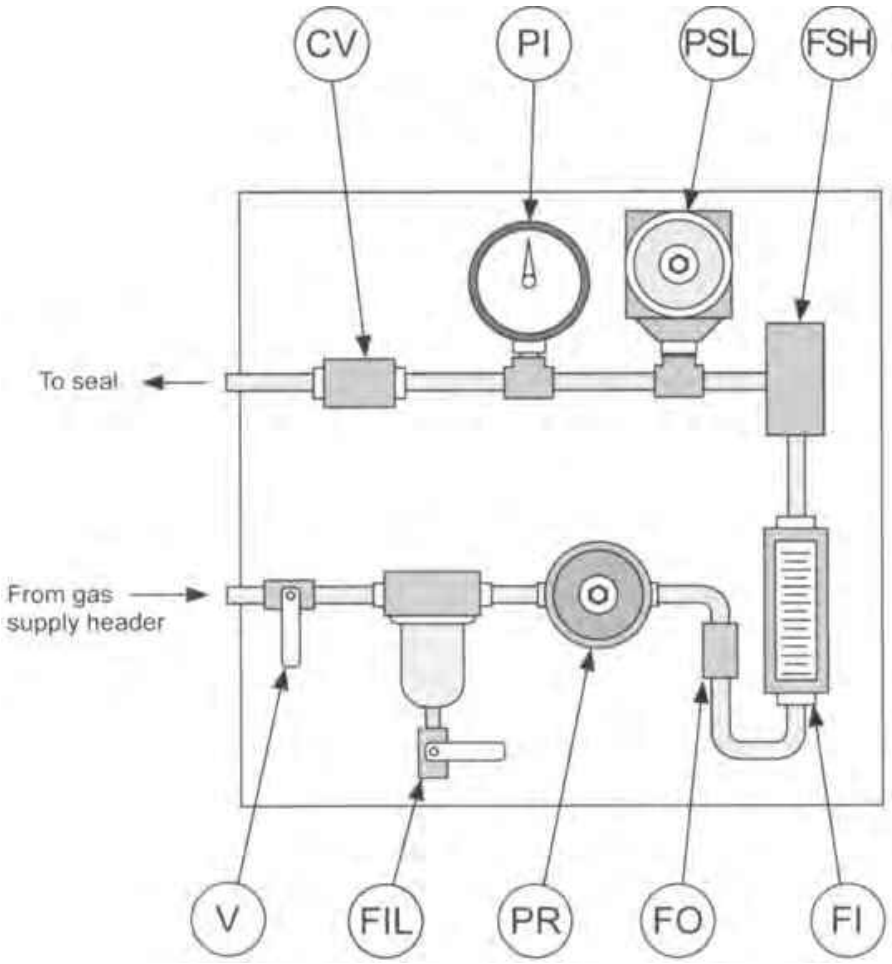
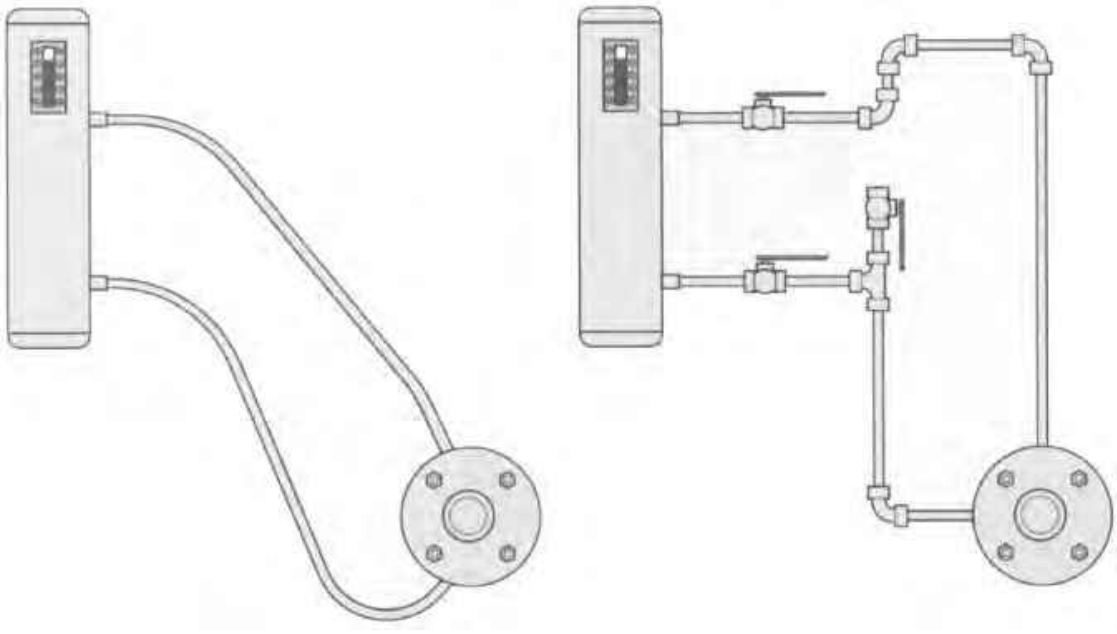


FIGURE 60 Typical gas control panel



Good piping practices Bad piping practices

FIGURE 61 Examples of good and bad piping practices

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SUMMARY

Mechanical seals are a critical component in modern centrifugal pumps. They can provide years of reliable service and meet the most stringent emissions regulations. They are, however, highly engineered devices that require careful selection and operation to achieve these results.

The following cutaway views illustrate some of the available options discussed.

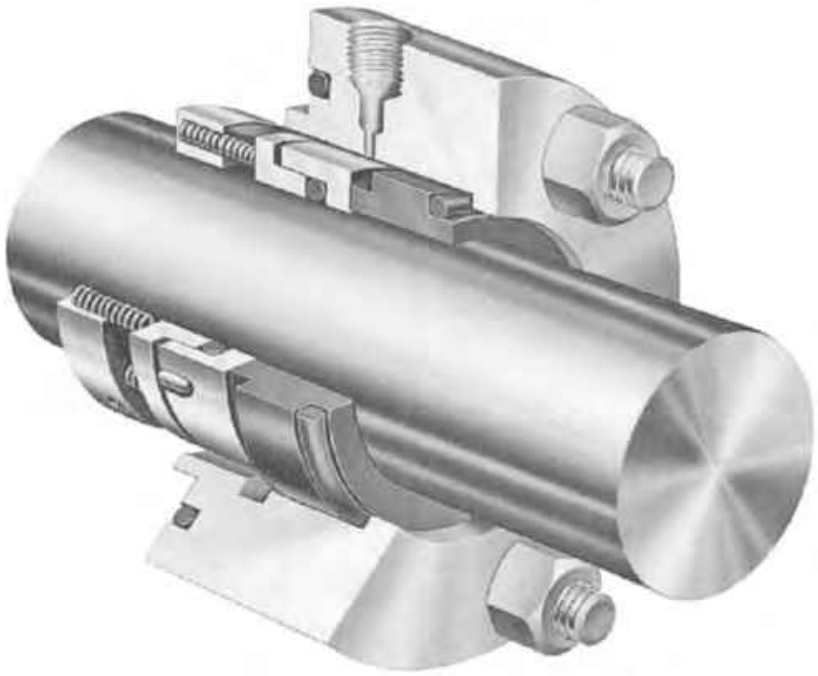


FIGURE 62 Component pusher seal (Flowserve Corporation)

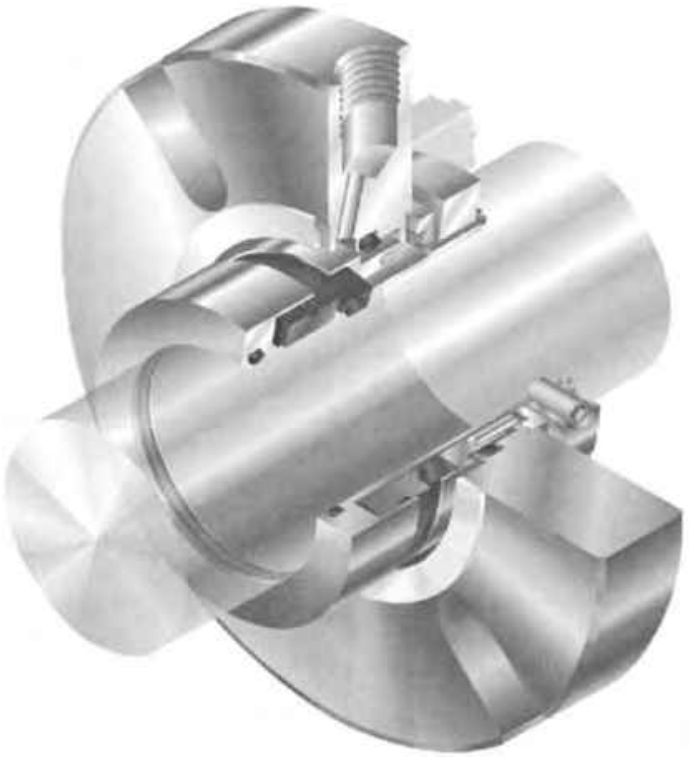


FIGURE 63 Cartridge stationary pusher seal (Flowserve Corporation)

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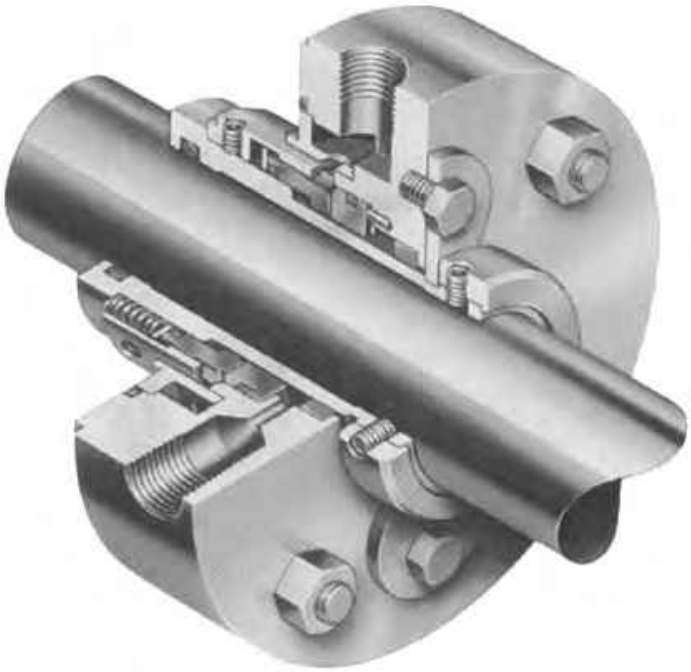


FIGURE 64 Cartridge rotating pusher seal (Flowserve Corporation)

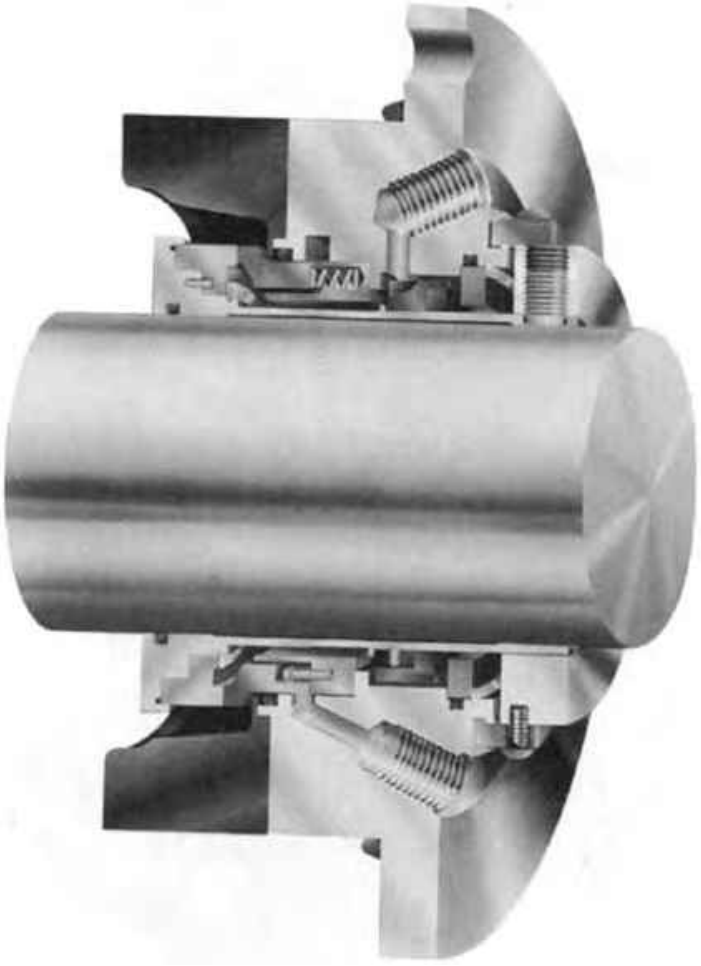


FIGURE 65 Slurry seal (Flowserve Corporation)

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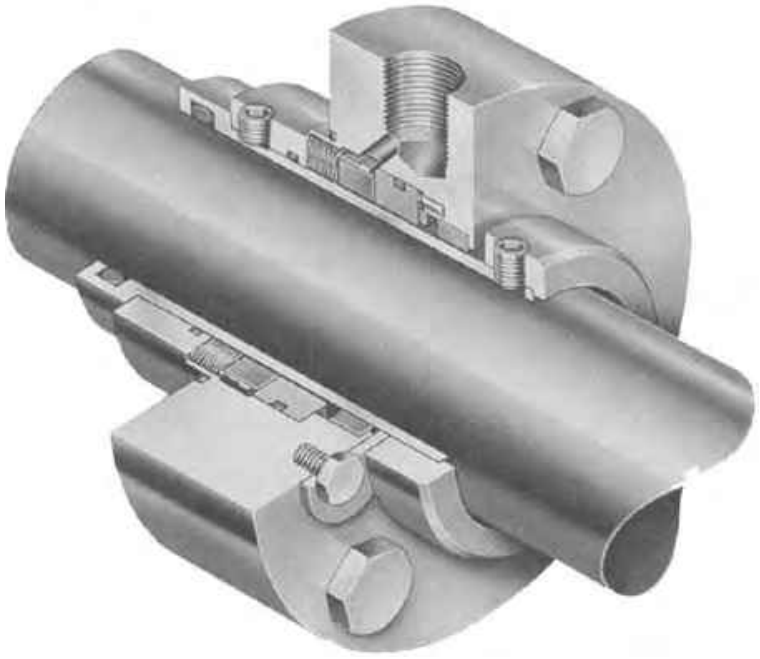


FIGURE 66 Cartridge rotating bellows seal (Flowserve Corporation)

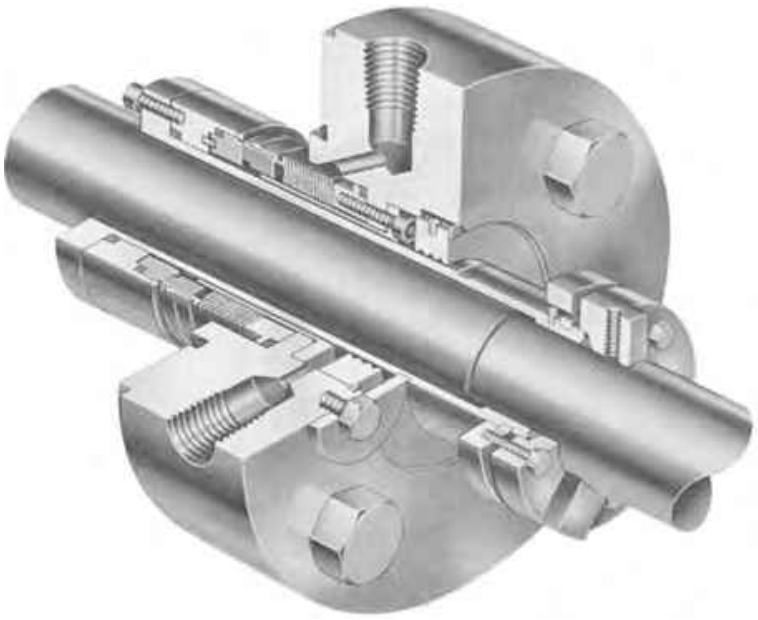


FIGURE 67 Cartridge stationary bellows seal (Flowserve Corporation)

REFERENCES AND FURTHER READING

American Petroleum Institute. "Centrifugal Pumps for Petroleum, Petrochemical and Natural Gas Industries." API Standard 610, 10th ed., Washington, DC, 2004.

American Petroleum Institute. "Pumps-Shaft Sealing Systems for Centrifugal and Rotary Pumps." API Standard 682, 3rd ed., Washington, DC, 2004.

Davidson, M. "The Effects of Seal Chamber Design on Seal Performance." 6th International Pump Users Symposium Proceedings, Turbomachinery Laboratory, Texas A&M University. College Station, TX. 1989.

HI., "Mechanical Seals for Pumps: Application Guidelines." Hydraulic Institute. Parsippany, NJ. 2006.

Huebner, M. B. "Improving the Reliability of Mechanical Seals." *Chemical Engineering Progress*. Nov. 2005.

Huebner, M. B. "Material Selection for Mechanical Seals: Part I." *Pump Engineer*. July 2005.

Huebner, M. B. "Material Selection for Mechanical Seals: Part II." *Pump Engineer*. Sept. 2005.

Huebner, M. B. "Mechanical Seal Techniques for Light Hydrocarbon Services." The Institution of Mechanical Engineers, Mechanical Sealing Technology Seminar, London, UK, Apr. 2007.

Huebner, M. B. "Overview of API 682 and ISO 21049." 23rd International Pump Users Symposium Proceedings, Turbomachinery Laboratory, Texas A&M University. College Station, TX. Mar. 2007.

Lebeck, A. O. "Principles and Design of Mechanical face Seals." John Wiley Interscience, New York, 1991.

STLE. "Guidelines for Meeting Emission Regulations for Rotating Machinery with Mechanical Seals." SP-30, Society of Tribologists and Lubrication Engineers, Park Ridge, IL. Apr. 1994.

Summers-Smith, J. D. "Mechanical Seal Practice for Improved Performance." The Institution of Mechanical Engineers, Suffolk, UK. 1992.

Young, L., Burroughs, J. N., Huebner, M. B. "Use of Metal Bellows Noncontacting Seals and Guidelines for Steam Turbine Applications." 28th Turbomachinery Symposium Proceedings, Turbomachinery Laboratory, Texas A&M University. College Station, TX. Sept. 1999.

Young, L., Huebner, M. B. "The Use of Wavy Face Technology in Various Gas Seal Applications." 15th International Pump Users Symposium Proceedings, Turbomachinery Laboratory, Texas A&M University. College Station, TX. Mar. 1998.

SECTION 5.3
CENTRIFUGAL PUMP INJECTION-TYPE SHAFT SEALS

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TIMOTHY L. WOTRING

Injection-type shaft seals (sometimes called *packless stuffing boxes*) are designed to control leakage from hot-water pumps. Cool water is injected into each seal to either suppress or regulate the hot leakage, which would otherwise flash upon reaching the outside of the pump.

Injection-type shaft seals provide high reliability and yet require little maintenance. They are used primarily in power plant boiler-feed and reactor-feed pump applications where shaft peripheral speeds are high (3600 rpm and up) and pumping temperatures are greater than 250°F (120°C). Under these conditions, conventional packing or mechanical seal-type stuffing boxes may not be suitable or desirable.

Injection shaft seals are either *serrated throttle bushings* or *floating ring seal* designs that regulate the flow, temperature, and pressure of the controlled leakage. The flow of the cool injection and of any hot water in the seals depends on operating pressures, but it is restricted by close seal clearances and is regulated by injection control valves.

The operating temperature of the seals is controlled by allowing cool injection water to surround the outside of the seal. Ports in the seal enable the cool injection water access to the shaft to either overcome or mix with hot water in the seal so that the resulting seal leakage is cool.

The seal designs must provide sufficient pressure breakdown between the pump suction or balance device chamber pressures inside the pump and the atmospheric conditions outside the pump. Upon reaching the outside of the seal, the cool leakage is piped away by gravity drain for eventual return to the power plant feedwater system.

SERRATED THROTTLE BUSHINGS

The construction of a serrated (sometimes called *labyrinth* or *grooved)* bushing (see Figure 1) varies from one pump manufacturer to another. However, the serrated designs basically involve a rotating shaft running with a reasonably small clearance, such as 0.002 to

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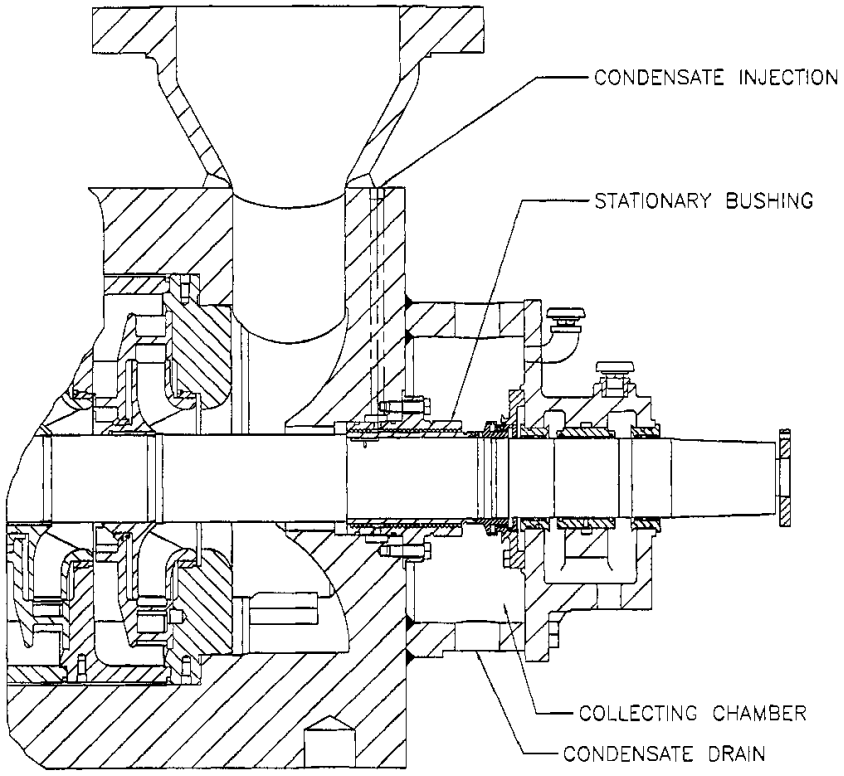


FIGURE 1 A cutaway view of a modern double-case boiler feed pump showing serrated throttle bushing shaft seals (Flowserve Corporation)

0.003 in/in (0.002 to 0.003 mm/mm) of the shaft diameter within a solid stainless steel (typically 12% chrome) stationary bushing installed in the pump casing end cover. Grooved serrations are applied to the hardened stainless steel rotating surface (usually a shaft sleeve) or to the stationary bushing or to both rotating and stationary surfaces to effect a high throttling action or stuffing box leakage reduction.

The serrations create a more effective pressure breakdown than a smooth axial surface and enable increased running clearances to let foreign particles, such as grit, pass through or flush out in the grooves. The increased clearances also enable more tolerance for any displacement between rotating and stationary components that might result from assembly misalignment or from distortions induced by transient temperatures. The grooved running surface area at clearances is greatly reduced relative to that of a smooth axial surface, and therefore possible metal surface contact between rotating and stationary parts is reduced during transient operations.

Seal leakage could be reduced by decreasing the running clearance in the injection seal, but the smaller clearance would limit the capability of the serrated seal to conform to radial shaft movement. Reduced clearances in any injection seal design may lead to problems with thermal distortions caused by a loss of available cool injection water and may also increase galling by particles trapped in the seal running fits.

FLOATING RING DESIGN

A segmented throttle bushing made of many floating rings will enable conformity with the radial shaft movement and smaller running clearances to reduce seal leakage. The construction of a floating ring stuffing box (see Figure 2) varies from one pump manufacturer to another, but the unit basically consists of a stack of hardened, martensitic stainless steel rings (typically 17% chrome), instead of a stationary bushing. The rings run against a

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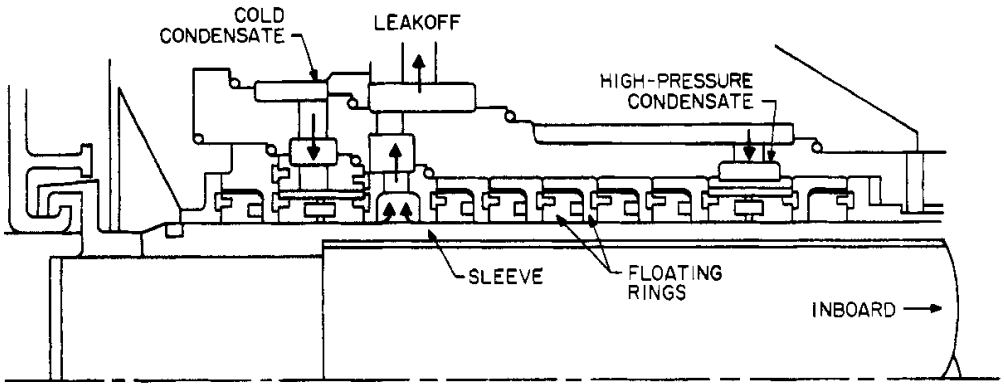


FIGURE 2 Floating ring seal design with shaft sleeve. Pegs prevent rotation of spring-loaded rings but let them float radially (From *Power*, August 1980, McGraw-Hill, New York, copyright 1980)

smooth, hardened, martensitic stainless steel (typically 12 to 17% chromium) rotating shaft sleeve surface.

The separate rings are all contained within a housing installed in the pump casing end cover. Each ring is loaded against an adjacent stainless steel ring spacer so that a stationary seal is produced in the axial direction, but the ring is still able to move radially with the shaft. An axial loading of the rings is provided by hydraulic pressure during operation and by springs during idle pump periods. The rings are locked against rotation usually by pegs and/or a slot arrangement.

A small radial clearance, usually 0.001 to 0.0015 in/in (0.001 to 0.0015 mm/mm) of the shaft diameter, is provided between the rings and the shaft sleeve to provide an adequate throttling of seal leakage across the limited number of seal rings. The length of each ring varies with the diameter of the injection-type seal but is generally about 0.50 in (13 mm). The radial clearance enables a reduced injection flow but increases the sensitivity of the seal to foreign particles in the seal leakage.

The individual seal rings are permitted to move radially (float), finding an equilibrium running position relative to the shaft. The capability to float up to 0.016 in (.4 mm) radially increases the tolerance for shaft misalignment as the shaft passes through the seal area. The multiplicity of seal rings further reduces the effect of any angular displacement between the rotating and stationary components that might arise from errors in the original assembly or from distortions caused by temperature changes during pump load variations.

Some injection-type shaft seal designs use a combination of serrations and floating rings in the same stuffing box. For either type of seal, careful maintenance during the assembly or disassembly requires accurate alignment of the rotating and stationary components as well as careful handling to avoid scratching the components. Cleanliness is of utmost importance with shaft seals.

CONDENSATE INJECTION REQUIREMENTS

Because boiler-feed and reactor-feed pumps normally have high feedwater temperatures, 250 to 500°F (120 to 260°C), with consequent vapor pressures higher than external atmospheric conditions, the seal leakage must be cooled to avoid flashing in the stuffing box or outside the pump. This cooling is usually accomplished by injecting cold condensate from a power plant condensate pump directly into the seal to cool the stuffing box components as well as the seal leakage.

The amount of condensate injection water required will depend on several factors, including: (1) whether the design is a serrated or floating ring; (2) the type of seal control system; (3) the diameter, clearance, and rotating speed of the running fit; and (4) the internal pump pressure and temperature. A typical example of stuffing box flows would be a

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5500-rpm, boiler-feed pump with a temperature-controlled seal system (described later) and a serrated 5-in (127-mm) diameter running seal with about 0.015 in (0.38 mm) of diametrical clearance. The approximate flows would be as follows:

1. Total injection per seal: 5 to 15 gpm (0.3 to 1.0 L/s)
2. Leakage from internal pump to seal: 0 to 5 gpm (0 to 0.3 L/s)
3. Drainage from each seal: 8 to 20 gpm (0.5 to 1.3 L/s)

Given a seal with a floating ring design (having typically half the clearance of seals with a serrated bushing design), the previous conditions would require only 50 to 60% of the injection water needed by the serrated bushing design with the larger clearance.

During pump standby periods, higher injection and leakage flows are required because of reduced seal throttling caused by low or zero speed, by high internal pump pressures, or by a pump balance device that induces higher internal leakage during standby. The injection flows to both stuffing boxes on a multistage pump may not be the same as a result of pump balance action at one end of the pump.

Cold condensate, 85 to 110°F (30 to 45°C), is usually available from the power plant condensate pump or secondary condensate booster pump discharge. The variation in condensate supply pressure relative to internal boiler-feed pump pressure makes it necessary to use injection control systems for satisfactory seal operation.

Condensate injection systems vary to accommodate the wide range of pressures and temperatures found in different power plant feedwater system layouts. The four basic types of injection flow systems are: (1) manual, (2) differential pressure-controlled, (3) differential temperature-controlled, and (4) constant drain temperature control.

Manual System Manual flow control requires setting and readjusting a valve by hand for each stuffing box each time a change occurs in pump operating conditions due to varying plant loads. Although it is possible to use such a control system, it is not usually recommended because of the inability to automatically compensate for rapid changing conditions.

Differential Pressure-Controlled System The differential pressure-controlled shaft seals operate as cold condensate maintained at a pressure greater than boiler-feed pump suction is injected into the central portion of the seal. By maintaining an injection pressure above the pump seal chamber pressure, a small portion of this injection water flows into the pump proper, and most of the injected water flows out of the seal into a collection chamber adjacent to the pump bearing bracket. The leakage in the collection chamber, which is vented to the atmosphere, is drained by gravity for an eventual return to the main feedwater system.

The differential pressure-controlled system has a pneumatic injection control valve. This valve in the condensate injection line is governed by an air signal received from a differential pressure control monitor that maintains the preset pressure differential, 10 to 25 lb/in^2 (70 to 170 kN/m^2), between the injection pressure and the internal pump pressure. This system is not always favored, however, because of a feedback instability tendency resulting from operating pressure changes affecting the valve position, which changes pressure, and so on. Another unfavorable factor for hot-water service is the introduction of cold condensate into a hot pump at all times, including pump idle periods, affecting pump prewarming conditions.

Differential Temperature-Controlled System The differential temperature-controlled system operates by maintaining a seal drain temperature differential of 25°F (14°C) above the injection water temperature. The temperature differential ensures a continuous out-leakage of hot boiler feed water. The hot out-leakage is desirable, ensuring that cold water cannot be injected into the pump and thus eliminating thermal distortions that occur with the differential pressure-controlled system. The valve controllers are identical to the controllers used for the differential pressure-controlled system.

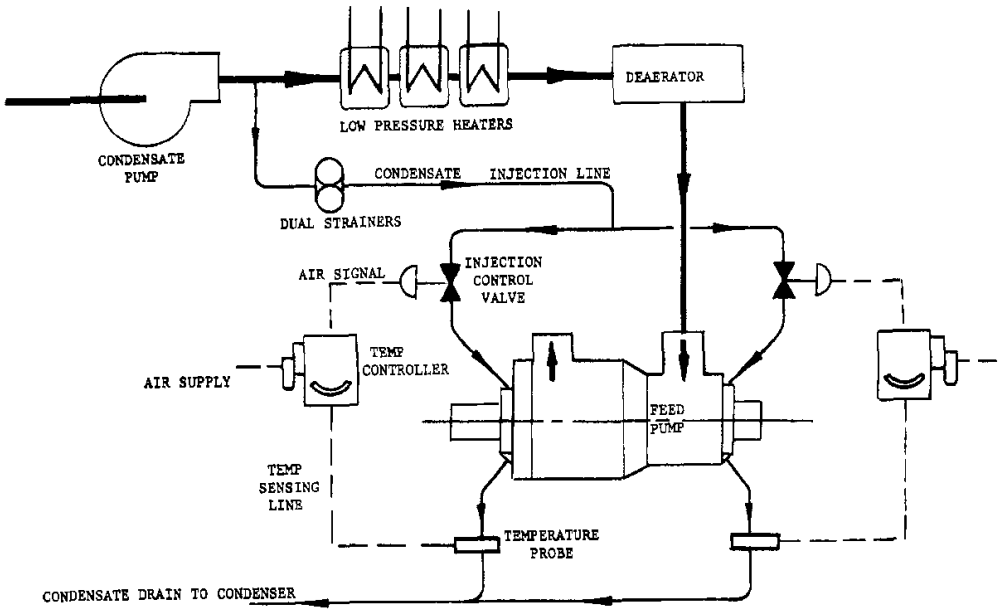


FIGURE 3 Temperature-controlled system regulating condensate injection to feed pump shaft seals

Constant Drain Temperature-Controlled System The constant drain temperature-controlled seal system (see Figure 3) controls hot out-leakage by throttling the injection flow to maintain a preset seal drain temperature. This system has a temperature-sensing probe in each seal drain line. Each probe is connected to an indicating temperature controller, which provides an air signal to a pneumatic control valve in the condensate injection line for control of the seal injection flow rate. Electronic control systems are often used where thermocouples or RTDs sense drain temperatures. The signal output is then processed through an I/P controller that adjusts the position of the pneumatically operated throttle valve as required.

The cold condensate is injected into the stuffing box central portion and allowed to mix with hot water entering the seal from inside the pump. The drain temperature is maintained at a preset 140 to 150°F (60 to 66°C) to preclude flashing in the stuffing box or in the drains. Note that with this system some hot water enters the seal. Therefore, cold condensate does *not* enter the hot pump and does not adversely affect pump warming conditions, especially during extended idle periods. The required condensate injection pressure is at least equal to the internal stuffing box pressure plus interim frictional loss between the condensate supply source and the point of hot-water mixture. Note that this system *may* enable satisfactory operation even when the condensate supply pressure is nearly equal to boiler-feed pump suction pressure. In addition to providing a rapid response to variations in operating pump conditions, this type of control will always supply just enough injection water to maintain the recommended drainage temperature.

Intermediate Leakoff System The intermediate-leakoff shaft seal system (see Figure 4) has many variations but basically uses a bleedoff from a central portion of the stuffing box. This system may be used to reduce internal stuffing box pressure if high boiler-feed pump suction pressure exists. To create a positive leakoff flow, the intermediate bleedoff flow is piped back to a plant feedwater system low-pressure point, such as a plant condenser, heater, or booster pump suction where the pressure is less than the boiler-feed pump suction pressure. However, the back pressure of the leakoff destination must be above the bleedoff vapor pressure to suppress flashing in the leakoff lines. This back pressure may be the leakoff destination pressure or may be created by an orifice or a valve.

Cold condensate injection into the stuffing box is controlled by a pressure differential monitor maintaining a preset pressure above the bleedoff pressure (refer to Figure 4). A stuffing box drain temperature control can also be used. Note that the cold condensate

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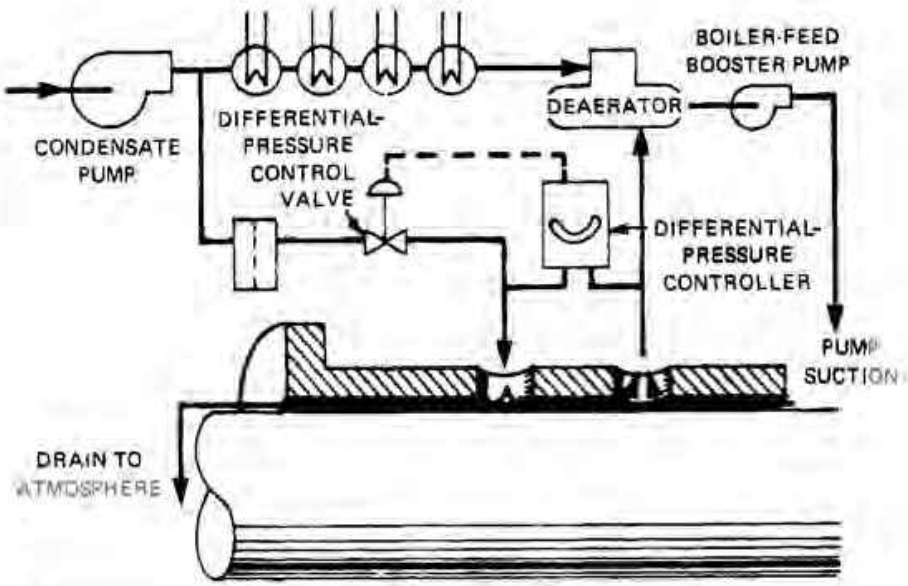


FIGURE 4 Typical intermediate-leakoff shaft seal system (From *Power*, September 1980, McGraw-Hill, New York, copyright 1980)

injection pressure need not equal or overcome a high feed pump suction pressure. The condensate injection temperature must still be 85 to 110°F (30 to 45°C) to keep the drain temperature below the flashing condition. Condensate injection shaft seals without an intermediate bleedoff but that are subject to suction pressures in excess of about 250 lb/in^2 (1725 kN/m^2) must be extremely long for a proper pressure breakdown. Longer shaft seals require thicker pump case end covers, affecting pump cost, and a longer rotating element, which could adversely affect rotor dynamics. The intermediate-leakoff shaft seal is effective where there is high feed pump suction pressure imposed by boiler-feed booster pumps or in a closed feedwater system with no deaerating open heater, wherein a condensate pump discharge can be fully imposed on the feed pump at low plant loads.

In plant systems with feedwater heaters between a booster pump and a feed pump, a high pressure condensate from the cooler booster pump is injected into the seal to enable a cooler intermediate leakoff to help prevent flashing (refer to Figure 2).

INJECTION SOURCES

Many power plant feedwater systems with deaerating direct-contact heaters (open cycle) usually have the boiler-feed pump drawing water directly from the deaerator. These open cycles with constant-speed condensate pumps always have cold condensate supply pressures in excess of boiler-feed pump suction pressure because of the increased available condensate pump head at low system flows and the interim system frictional loss at high system flows.

If a boiler-feed booster pump or a closed feedwater system with no open heaters and resultant higher boiler-feed pump suction pressures is used, the pump manufacturer may elect to require condensate injection seal water booster pumps or may use an intermediateleakoff packless shaft seal with a condensate injection overcoming only the intermediate leakoff pressure. If the condensate pumps are variable-speed units that enable the condensate injection pressure to drop to an equal feed pump suction pressure at low loads with little or no feedwater system frictional drop, then condensate injection seal water pumps are required. In this situation, at least one pump manufacturer offers an optional pumping ring configuration in their packless seal that can increase the seal water pressure in the stuffing box to overcome any pump suction pressure.

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AUXILIARY EQUIPMENT

The condensate injection piping should be conservatively sized based on the maximum injection flow requirements to obtain a low-pressure drop between the injection source and the seal injection control valve. These control valves may be equipped with limit stops to prevent full closure and enable a continuous cool injection to the seals under almost all operating conditions. In some installations, isolating lines are furnished around the valves to enable a continued injection flow even during control valve maintenance. The valves can also be designed to remain open during a failure, such as a loss of station air to the pneumatic controls, and to close only with an air supply. Injection control systems that are not properly maintained might result in cold water entering a hot pump. Should this problem occur while a boiler feed pump is in the hot standby mode or when turning gears, thermal gradients will occur, leading to contact among the close running fits within the pump. Proper maintenance and operation of the injection control systems is necessary to ensure reliable operation of the pump itself. The air supply filter regulators for each control must be furnished with relatively dry clean air at station supply pressure.

The condensate injection supply to the seals must be clear and free of foreign matter to prevent damage to stuffing box components. It is therefore necessary to install filters in the injection line prior to the control valve. To keep damaging fine mill scale, oxide particles, abrasives, and other materials from entering the small seal clearances, several pump manufacturers recommend 100-mesh (150-micron) dual strainers. If dual strainers with isolating valves are used, each filter can be cleaned without interrupting injection flow during pump operation. Pressure gages should be installed before and after each filter to permit the operator to monitor filter pressure drop. A differential pressure switch and alarm for each filter are preferable to alert the operator to clean the strainer when pressure drop becomes excessive.

The condensate injection shaft seals should always be filled with cool water before and during pump operation, even during reverse pump rotation. Some pump manufacturers stipulate that condensate injection *must* be continuous without any interruption during all operation modes.

The clearances in the condensate injection shaft seal may double over the service life of the internal wearing parts. With double clearances, the leakage will approximately double. This factor should be considered when sizing the return drain piping back to the plant condenser if frictional losses are to be kept to a minimum. The drain line should be pitched at least a quarter-inch per foot (20 mm/m). The collecting chamber at the pump stuffing box is vented to the atmosphere, and the only head available to evacuate the chamber is the static head between the pump and the point of return. This head must always be well in excess of the frictional losses (even after the leakage is doubled). Otherwise, the drains may back up, the collection chambers may overflow, and the adjacent bearing brackets may flood, with subsequent possible intrusion of water into the pump bearings and lubricating oil.

The seal collection chambers have especially large connections to assure proper drainage, provided no back pressure exists. Two types of condensate drain systems can be used to dispose of the drain coming from the collecting seal chambers. One system uses traps that are piped directly to the plant condenser if sufficient static head exists for positive drain flow. The second system collects the drain in a condensate storage tank into which various other drains (from other pumps shaft seals and so on) are also directed. As this vented storage tank is under atmospheric pressure, it must be set at a reasonable elevation below the pump centerline so that the static elevation difference will overcome frictional losses in the drain piping. A separate condensate transfer pump, under control of the storage tank liquid level control system, can then pump the condensate drains from the storage tank into the plant condenser. The storage tank should have its own overflow protection system that enables outside drainage if, for some reason, proper drainage cannot be achieved. For example, the top of the tank vent pipe should be below the pump centerline to help preclude the possibility of drainage backing up to the level of the pump seal collection chambers. Note that this storage tank should also be large enough for an adequate drainage collection to help prevent backups.

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PACKLESS SHAFT SEALS WITHOUT INJECTION

The packless shaft seals that have been so successfully applied to boiler-feed, reactor-feed, and booster pump services are applicable to a number of other services. For instance, they are very suitable for cold-water condensate booster pumps and for high-pressure pumps applied to hydraulic descaling or hydraulic press work. In such services, there is no need to bring in an injection supply water to the breakdown seals (unless the pumped water is not clear and free of gritty material), because the water handled by the pump is already cold with no danger of flashing as it leaves the pump stuffing boxes.

C • H • A • P • T • E • R • 6

PUMP BEARINGS

SECTION 6.1
CENTRIFUGAL PUMP BEARINGS

W. G. GOODMAN
C. C. HEALD

PUMP BEARINGS

A pump bearing system consists of bearings to carry radial loads and at least one bearing to carry axial or thrust loads. For horizontally mounted pumps, there are two basic pump rotor arrangements: the pumping element overhangs the bearing system (Figure 1) or the pumping element is between bearings (Figure 2). The function of bearings in centrifugal pumps is to keep the shaft or rotor in correct alignment with the stationary parts under the action of radial and transverse loads. Bearings that give radial positioning to the rotor are known as *radial* or *line bearings,* and those that locate the rotor axially are called *thrust bearings*. In most applications, the thrust bearings actually carry both axial and radial loads.

Loads on Bearings Centrifugal pump bearings are subject to radial and axial loads that result from internal pressures acting on the surface areas of the impeller. The resultant force acting perpendicular to the shaft is considered radial load and the resultant force acting along the shaft axis is considered axial load. Loads in a centrifugal pump vary considerably and are approximately proportional to the developed pressure for a given pump design. Pressure variations occur because of changes in liquid specific gravity, varying operating speed, changes in impeller diameter and at various operating points on the performance curve. Centrifugal pump bearings must be carefully selected to operate reliably when carrying any of these varying loads.

Pump designers utilize various techniques to balance radial and axial loads. Radial thrust loads are described in Section 2.2.1, along with the influence of impeller and casing design on the magnitude of radial unbalance. Axial thrust loads are also described in Section 2.2.1, along with methods of balancing thrust loads to reduce the residual unbalanced axial loads that must be carried by thrust bearings.

Types of Bearings Rolling-element bearings are used for the majority of pump applications. They were originally called "antifriction" bearings because the rolling elements "roll" (and theoretically don't slide) such that friction is minimized. These bearings can be used

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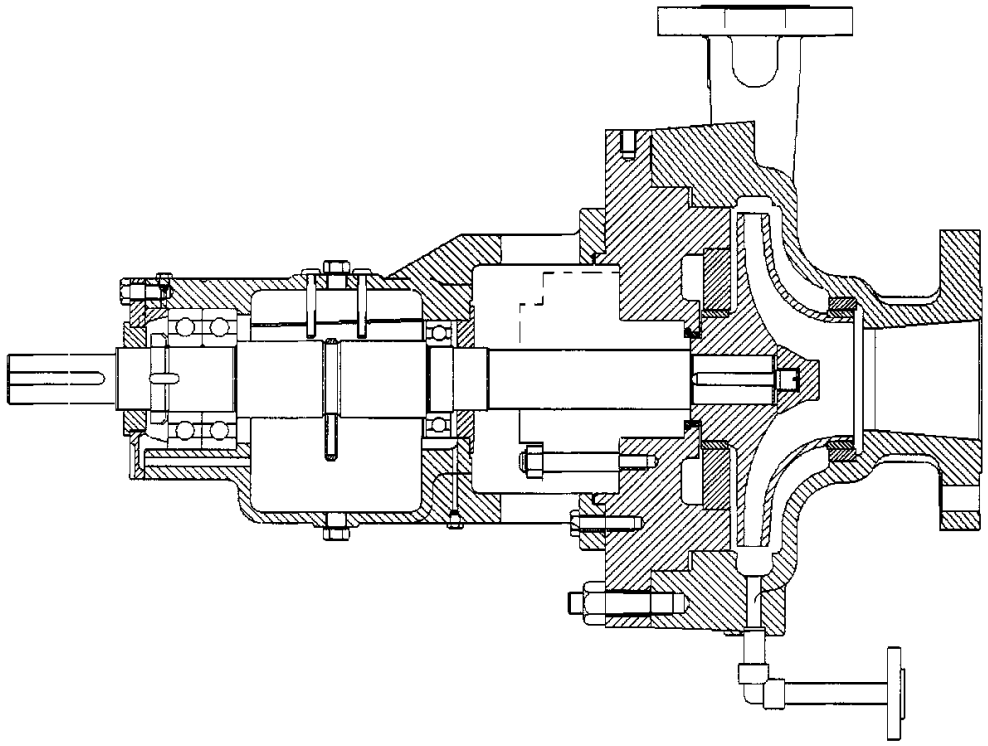


FIGURE 1 Overhung impeller pump and pumping element

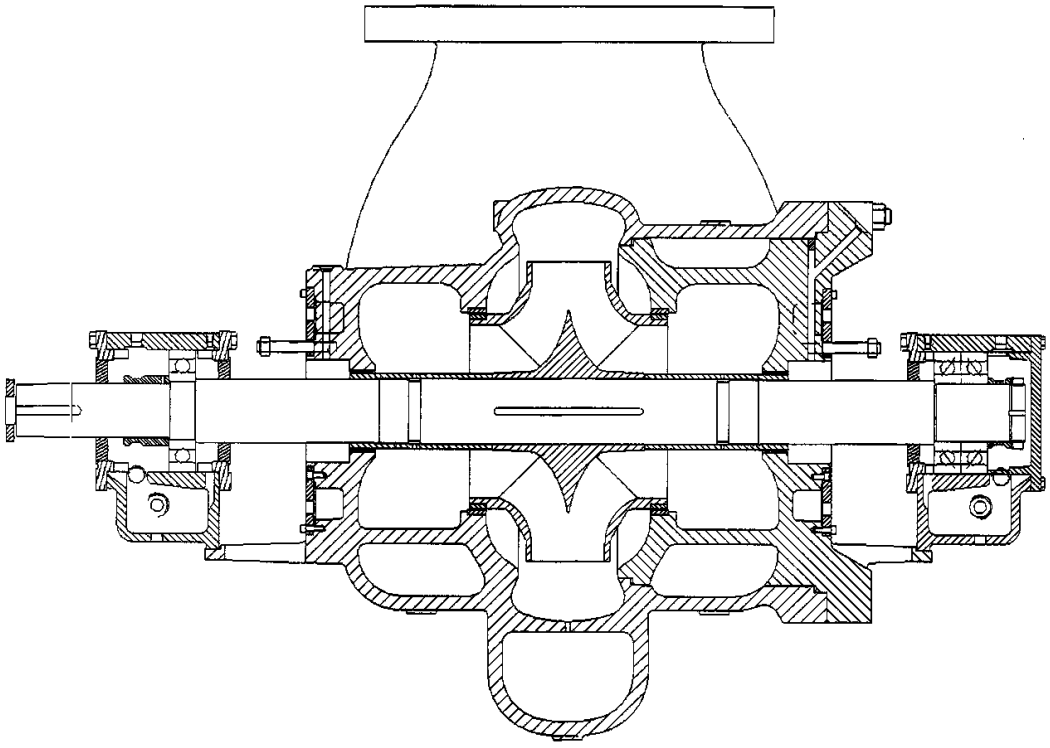


FIGURE 2 Between bearings pump and pumping element

to carry purely radial loads, both radial and axial loads, or purely axial loads, depending on their design and how they are applied or mounted (installed). Ball bearings and various types of roller bearing designs (cylindrical, spherical, and taper roller) are commonly used. Bearing manufacturers' catalogues usually provide all information necessary to properly select and apply these bearings.

Ball bearings come in many configurations, such as single and double row with various contact angles, to handle purely radial loads, combined radial and axial loads, and purely axial loads. Sometimes ball bearings are mounted in pairs (duplex) or triples (triplex) to increase load-handling capability or to provide axial thrust handling capability in both directions, as in the case of an angular contact bearing. The shape of the rolling element is that of a ball. Ball bearings are considered to be a point-contact bearing, meaning that the location where the load is transferred from the bearing race to the rolling element is a small circular area. Ball bearings have a relatively low load rating because the small area results in relatively high contact stress for a given load (stress = force/area).

Cylindrical roller bearings are used for purely radial loads for applications that require a higher load carrying capability than a ball bearing. These bearings are available only in single rows. The shape of the rolling element is that of a cylinder. Cylindrical roller bearings are considered to be a line-contact bearing because the contact between the bearing race and the rolling element is in the shape of a line. This shape provides a larger contact area that reduces the contact stress for a given load as compared to a ball bearing.

Tapered roller bearings accommodate both a radial and axial load. The shape of the rolling element is that of a truncated cone. These bearings are available only in a single row configuration but must be mounted in pairs since the axial setting or preload is critical to proper operation. They are also a line-contact bearing and, therefore, have a relatively high load carrying capability.

Spherical roller bearings also accommodate both radial and axial loads, though the radial load rating is considerably higher than the axial load rating. These bearings are available only in a double row configuration. The shape of the rolling element is that of a wooden stave barrel. The contact patch is ovular in shape resulting in high load carrying capability.

When bearing loading or rotating speeds exceed, capabilities of a rolling-element bearing, hydrodynamic bearings are generally selected. Hydrodynamic sleeve (journal) bearings can only accommodate purely radial loads (see Section 6.1). The shaft rides on a film of oil as it rotates inside of a babbit-lined sleeve. These bearings are a line-contact bearing and have a very high load carrying capability since the bearing contact area is quite large. Contact area is a misnomer since, during operation, the shaft is supported by a layer of oil (oil film) that further increases the "contact area." The bearings are called *hydrodynamic bearings* because, during operation, an oil "wedge" is formed between the shaft and the sleeve, which lifts the shaft away from the bearing metal. Thus "contact" between the shaft and the bearing metal does not actually occur during normal operation.

Tilting pad thrust bearings are also hydrodynamic bearings and can only accommodate purely axial loads. These bearings are normally used in conjunction with hydrodynamic sleeve bearings to accommodate the radial load. Axial loads from the shaft are transferred through a disk mounted on the shaft to the bearing. A compliment of pie shaped pads is mounted in the bearing housing facing each side of the disk to handle axial loads in either direction. These pads are allowed to pivot individually which allows an oil wedge to form on each of the pads, transferring the load to the housing.

Vertically suspended, magnetic drive, canned motor, and certain types of multistage pumps utilize internally mounted product lubricated sleeve bearings. These bearings are often made from rubber, bronze, carbon, or silicon carbide. The fluid flowing between the bearing and the shaft creates a dynamic centering effect, known as the Lomakin effect. Magnetic drive pumps, canned motor pumps, and some multistage designs also utilize product-lubricated thrust bearings or thrust balancing devices for axial thrust, whereas vertically suspended and horizontal multistage pumps normally handle axial thrust with externally mounted rolling-element or hydrodynamic thrust bearings.

Clearance and Preload Single-row deep-groove ball bearings are available with several classes of radial internal clearances. As standard, they are manufactured with Normal

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radial internal clearance, but a popular alternative for severe duties, especially applications involving pumping temperatures higher than ambient, is a larger than Normal, or C3, radial internal clearance. Pumps designed for petroleum, petrochemical, and natural gas industries, such as covered by API Standard 610 (ISO 13709) require a C3 internal clearance as standard. To meet this requirement, bearings should be specified to be in accordance with ISO 5753, Group 3. Greater internal clearances may reduce the temperature rise of the bearing lubricant and allow the bearings to run cooler under load. However, there may be a trade-off in that higher pump vibration velocities may occur with greater bearing internal clearances. It is important that the temperature rise of the bearings under maximum load conditions does not exceed the capability of the lubricant to properly perform and maintain the required viscosity, and thus film thickness, within the bearings.

In centrifugal pumps, bearing preload is a term applied to two bearings mounted in a pair to handle both radial and axial thrust loads. Tapered roller bearings and angular contact ball bearings are typically mounted to a predetermined setting or preload. This setting may be necessary to maintain proper contact of the rolling elements with the bearing races during operation to avoid skidding of the unloaded bearing in the pair. For angular contact ball bearings, the setting ranges from various clearances (non preloaded) to various amounts of preload. The determination of clearance or preload is based on a balance between providing sufficient clearance to operate at a low temperature and sufficient contact to avoid skidding of the rolling elements in the lightly loaded bearing in the pair. The pump manufacturer is responsible for determining the amount of preload or clearance necessary to meet the requirements of the standard or specification to which the pump is manufactured and sold.

It should be noted that bearing preload occurs in two other ways: (a) when the bearings are mounted on the shaft with an interference fit (as required by many standards, particularly for dynamic loading applications), and (b) when, for hot liquid pumping applications, the shaft heats more than the bearing races. Also, bearing preload determined by interference fit must be evaluated as the bearings may heat up in service and it may either decrease or increase.

The balance between bearing preload and clearance is not always easy to achieve. To address instances in ball bearings where ball skidding is observed or can be anticipated—in the unloaded bearing of the duplex-mounted pair, at least one bearing manufacturer provides a matched set of bearings consisting of one 40° contact angle bearing paired with one 15° contact angle bearing. Properly selected and mounted, the 15° contact angle bearing should be less susceptible to ball skidding when operating unloaded (when the axial thrust is in a direction such that it is all absorbed by the 40° contact angle mate). Care must be taken with this bearing type to ensure the direction of axial thrust is known and the bearing mounted accordingly. Also, it must be realized that the 15° contact angle bearing has less axial load capability than the 40° contact angle bearing, and bearing selection and sizing must account for this difference in capability. If axial bearing loads in both directions can be of the same magnitude, this option may not be feasible.

Retainers Bearing retainers or "cages" are used to guide and separate the rolling elements as they revolve in the bearing races. Bearing retainers are available in various materials and styles depending on bearing type and application. Pressed steel, pressed brass, machined bronze, and nonmetallic retainers are commonly used. Pressed steel and nonmetallic retainers are successfully used on single and double row ball bearings, cylindrical roller bearings, and tapered roller bearings. Pressed brass is used successfully on low speed angular contact bearings. Machined brass is successfully used on spherical roller and high-speed angular contact bearings. Spherical roller and high-speed angular contact bearings require a strong retainer with closely fitted rolling element pockets to properly guide the rolling elements.

Nonmetallic retainers should not be used for high temperature applications and are not allowed by some heavy-duty pump standards such as API 610 (ISO 13709).

Arrangements Each pump shaft is supported by two radial bearings and a thrust bearing; the thrust bearing is normally a double acting thrust and radial bearing. For pumps

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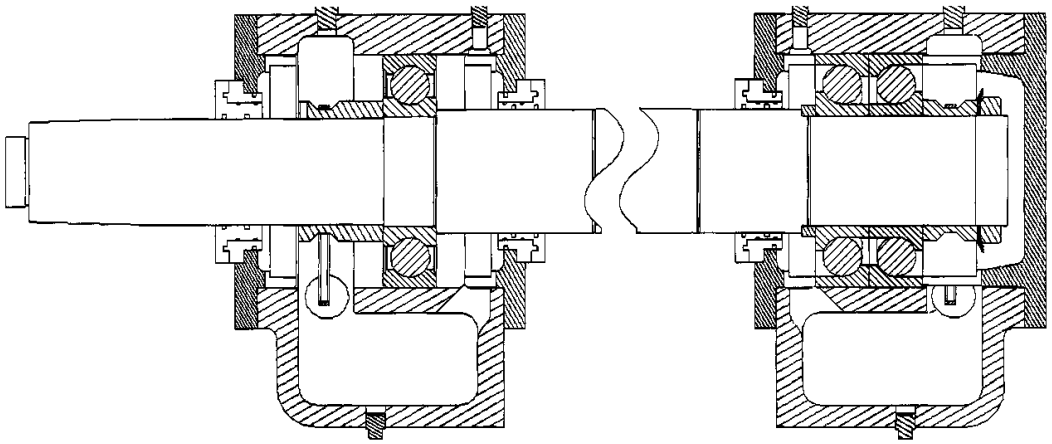


FIGURE 3 Rolling-element radial and thrust bearings

with externally mounted bearings, bearings are usually applied in one of the following arrangements: rolling-element radial and thrust (Figure 3), hydrodynamic radial and rolling-element thrust (Figure 4), or hydrodynamic radial and thrust (Figure 5). Rolling-element bearings are normally provided on simple machines when the application of the bearing is within the speed and load limitations of the bearing. Hydrodynamic bearings are better suited for higher speeds and loads and are, therefore, used on larger, more critical machines. Though both rolling-element and hydrodynamic bearings have proven reliability, rolling-element bearings tend to fail more rapidly once failure has begun. Even when worn, hydrodynamic bearings will seldom fail suddenly, which allows for a planned shutdown to rebuild the pump.

Table 1 shows selection criteria commonly used for the various bearing arrangements (Reference 1).

Lubrication Rolling-element bearings may be lubricated with grease or oil. Sealed bearings are available in certain bearing types that are pregreased from the bearing manufacturer, but most industrial pumps utilizing grease lubrication are regreasable. Grease is an acceptable lubricant for many applications since grease will protect the bearing surfaces from oxidation and regreasing purges contaminants from the bearing. Grease-lubricated bearings tend to operate at a higher temperature than oil-lubricated bearings, particularly until excess grease is purged from the bearing after regreasing.

Oil lubrication is also a widely used method of lubrication. There are various methods of oil lubrication: flood oil, oil splash, and ring oil, each using an oil sump that is integral with the pump bearing housing, and oil mist.

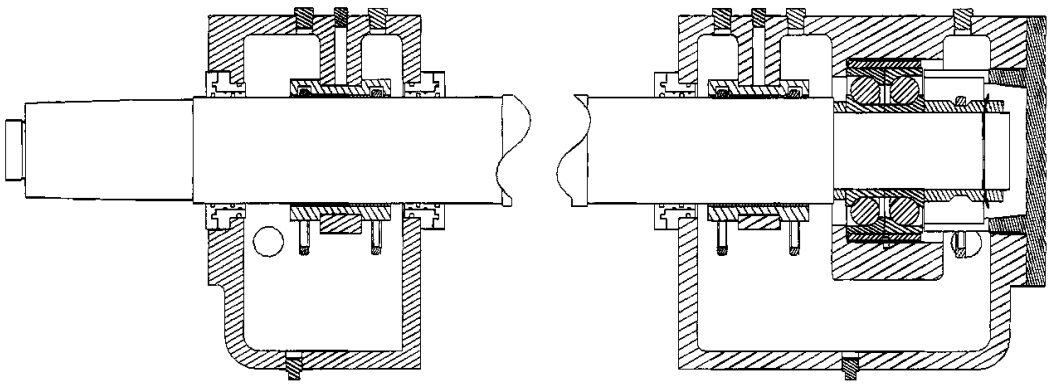


FIGURE 4 Hydrodynamic radial and rolling-element thrust bearings

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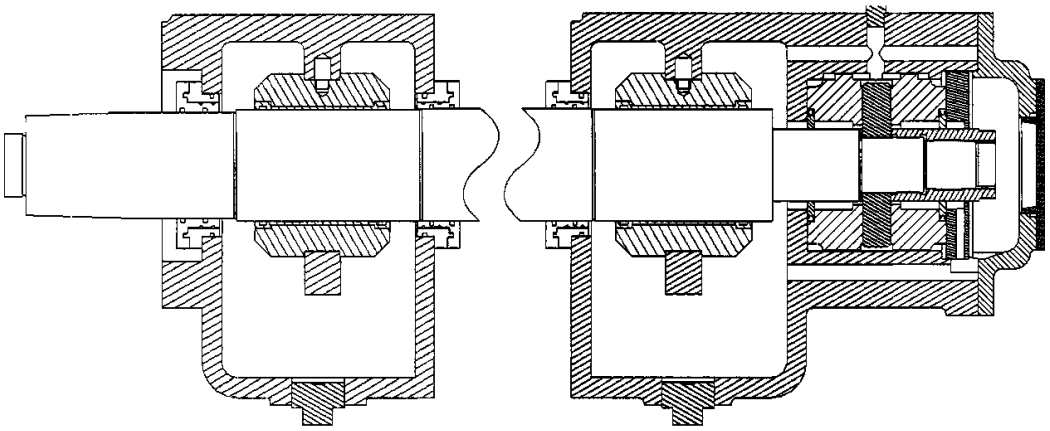


FIGURE 5 Hydrodynamic radial and thrust bearings

The oil level in a flood oil system is high enough to contact the rolling elements with the oil setting normally halfway up the bottom most rolling element. If two bearings are in the same housing such as for an overhung pump, the radial and thrust bearing must be the same size or nearly the same size in a flood oil system. Oil is delivered to the bearing as the rolling elements revolve and dip into the oil.

An oil splash system is similar to a flood oil system except the oil level is below the rolling elements and there is a flinger or thrower, normally a disk, mounted to the shaft inside the bearing housing. The outer portion of the thrower dips into the oil as the shaft rotates and creating a stream of oil and oil droplets that "splash" around inside the bearing housing and into the bearing. Bearings in this arrangement normally operate at a lower temperature because the bearing is not flooded with oil and because there is greater oil circulation thereby increasing cooling area. Since the oil flinger is rotating at the same speed as the shaft, foaming of the oil can be a problem for higher speed applications.

Ring oil systems are similar to splash systems except, instead of an oil flinger, an oil ring is used. The oil ring normally rides in a groove in the shaft and dips into the oil in the sump. As the ring rotates, oil is drawn up and splashes around the inside of the frame and into the bearing. The advantage of the oil ring is that it operates at a slower speed than the shaft and does not tend to cause oil foaming. Some bearing housings utilize troughs or channels that catch the oil stream coming from the oil rings and positively direct the oil to the bearings.

Oil mist systems are widely used and are particularly suitable for high-speed rolling-element bearings. There are two basic types of oil mist systems, pure oil mist and purge

TABLE 1 Bearing arrangement selection

Condition	Type and Arrangement of Bearing
Radial and thrust bearing speed and life are within limits for rolling-element bearings	Rolling-element radial and thrust
Radial bearing speed or life is outside limits for rolling-element bearings and thrust bearing speed and life is within limits	Hydrodynamic radial and rolling-element thrust, or hydrodynamic radial and thrust
Radial and thrust bearing speed or life is outside limits for rolling-element bearings	Hydrodynamic radial and thrust

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oil mist. These systems continuously provide fresh oil in the form of a mist from a centralized oil mist generator. Experience has shown that bearing housings equipped with oil mist lubrication systems result in extended bearing life. The bearings tend to run cooler, seldom fail because of lack of lubrication, and inherently provide a bearing environment that is free of the such common contaminants as dirt, dust, and water.

Pure oil mist systems do not utilize the oil sump other than to direct the mist and any condensed oil to a drain once it has passed through the bearing(s). Any internal troughs or passages, such as oil returns for non-oil mist systems, must be plugged to avoid short-circuiting the mist from flowing through the bearing(s). Normally one oil mist reclassifier is used at each pump bearing housing to ensure that a proper oil mist and air mixture is delivered to each bearing, even though there may be more than one inlet fitting on the bearing housing—one for each bearing to be lubricated. Oil mist injectors are normally located between the bearing and bearing housing end cover and the mist is forced to pass through the bearing to either a vent or a drain that is located on the opposite side of the bearing. If a directed oil mist stream is required to ensure that mist circulates through the bearing (e.g., to combat "windage" from the bearing), directional reclassifiers may be installed adjacent to those bearings. This is usually required only for larger or high-speed angular contact bearings because they can act like mini pumps that can push air against the oil mist attempting to enter the bearing.

Purge oil mist systems are used in conjunction with sump oil, splash oil, or ring oil systems. A central oil mist connection is provided in the bearing housing for injecting the oil mist and provide a low-pressure oil-saturated environment for the bearings. A reclassifying fitting is located in this feed line to ensure a proper oil mist. This low pressure created by the oil mist within the bearing housing tends to prevent water, moisture, dirt, dust, and other contaminants from entering the bearing environment. In addition to the oil mist inlet connection, a connection is provided in the bearing housing to draw off excess oil that accumulates from condensed oil mist during operation in order to maintain a constant oil level in the sump. After the oil mist passes through the bearings, the residual mist that is not condensed escapes through the bearing end covers.

Hydrodynamic bearings are lubricated with oil. A hydrodynamic radial and rolling-element thrust bearing arrangement typically utilizes oil ring lubrication. There are grooves in the upper half of the sleeve bearing that allow the oil ring to ride on the shaft. In this case, the oil ring delivers oil to the sleeve bearing directly as it rotates. The rolling-element thrust bearing has a separate oil ring to provide lubrication.

A hydrodynamic radial and hydrodynamic thrust bearing arrangement requires an external lubricating oil system. The additional heat load generated by the oil shear from the hydrodynamic thrust bearing requires a larger oil reservoir and external cooling. This lubrication system uses a separate oil pump to circulate cooled and filtered oil to the pump bearings and back to the reservoir. Most systems have a primary oil pump that is shaft driven from the pump outboard bearing housing with a backup electric motor driven pump on the lubrication system assembly or skid. Often oil for the driver bearings and other drive train components, including gears, is integrated into the same lubrication system.

Cooling A water-cooled oil sump utilizes a plain or finned tube that runs through the oil sump. Cooling water is circulated through the tube, conducting heat from the oil (Figure 6). Some designs use a separate water jacket that bolts to the outside of the oil sump to draw heat from the oil through the bearing housing wall into the water jacket. Tube style heat exchangers are preferable because they are easier to disassemble and clean any mineral deposits, or even replace, than are jackets. Water cooling is not effective nor is it applied on oil mist or grease-lubricated systems.

Air-cooling is becoming more popular as cooling water is in limited supply or of poor quality in many parts of the world. It is also less expensive to operate. In its simplest form, a fan is mounted on the shaft on the outboard side of the bearing housing to direct air over the bearing frame toward the pump, thus creating forced convection of heat from the bearing frame. Sometimes a heat deflector is also utilized which is a finned disc mounted on the shaft between the seal chamber and the bearing housing (Figure 7). This device conducts heat from the shaft before it reaches the bearing housing, absorbs heat radiated

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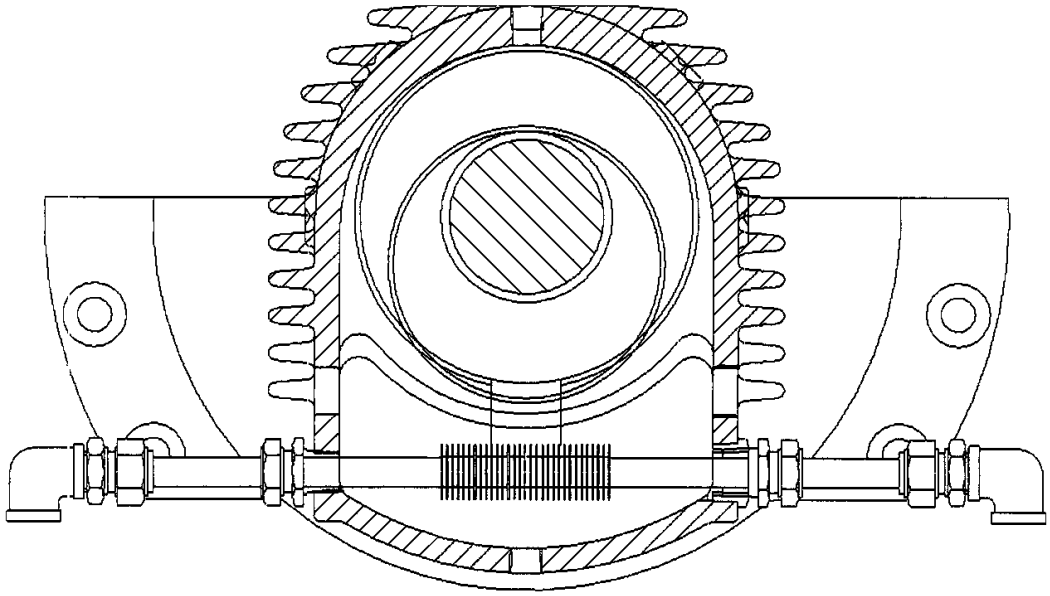


FIGURE 6 Finned cooling tube running through the oil sump

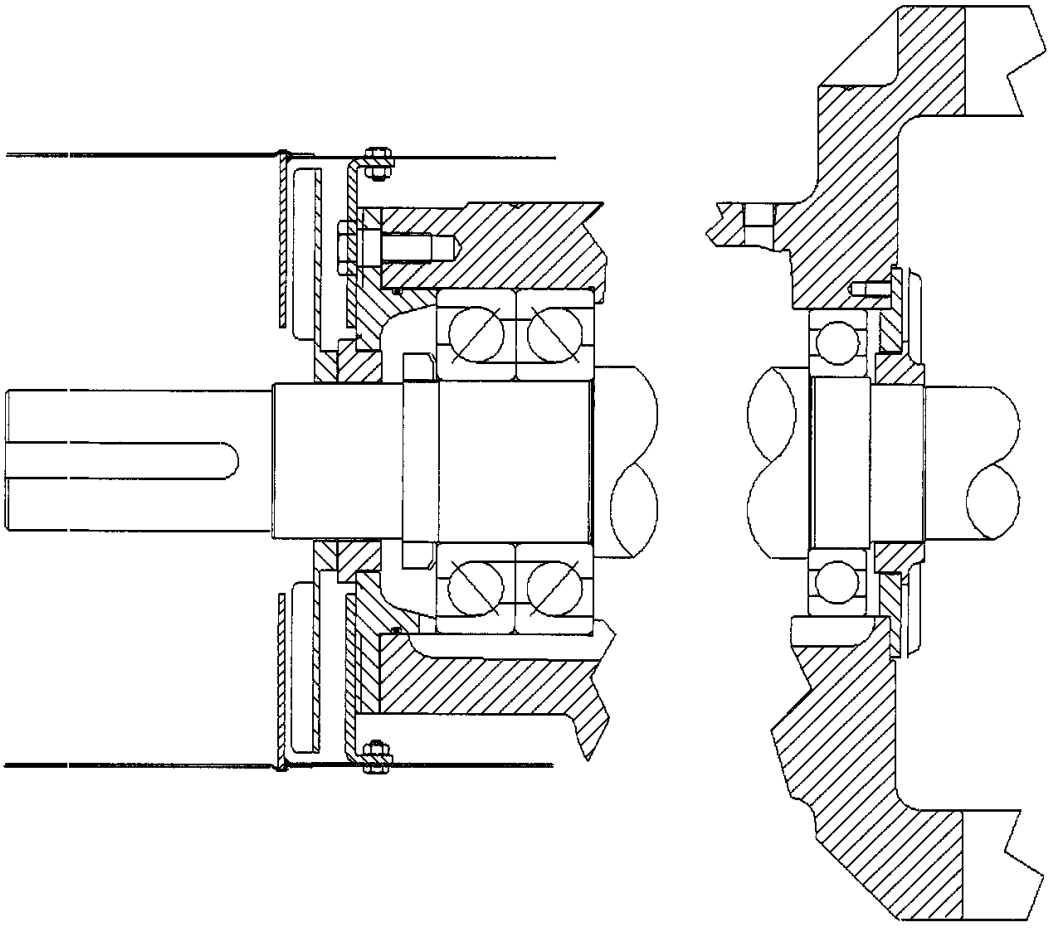


FIGURE 7 Finned disc mounted on the shaft between the seal chamber and bearing housing to deflect and remove heat

from the seal chamber and also creates forced convection in the seal chamber area. Air-cooling can be used on sump oil, oil mist and grease-lubricated bearing systems.

External lubricating oil systems incorporate either an oil-to-water or oil-to-air heat exchanger. The oil-to-water heat exchanger is a tube-type system with the lubricating oil on the tube side and the water on the shell side. Small oil-to-air heat exchangers consist of an electric motor driven fan, forcing convection over a coil of finned tubes containing the circulating oil. Larger systems utilize a large radiator and fan much like you would see on a diesel engine.

Sizing Generally, bearings should be sized for continuous operation under all specified conditions. Many specifications, such as API 610 (ISO 13709), require the thrust bearing be sized to carry loads at both design internal clearances and also at twice design internal clearances. Any thrust loading from gears and any axial force transmitted through flexible couplings also need to be added to thrust from the pump rotor to give a total load requirement for the thrust bearing. Thrust bearings should also provide full-load capabilities if the pump's normal direction of rotation is reversed. Also, if a sleeve-bearing motor (without a thrust bearing) is directly connected to the pump shaft with a coupling, the coupling-transmitted thrust should be assumed to be the maximum motor thrust.

Rolling-element bearing life (basic rating life, L_{10h}) for each bearing or bearing pair) shall be calculated in accordance with ISO 281. Minimum bearing life is specified by various industry standards such as ASME B73, API 610, and ISO 13709. ASME B73 requires 16,000 h and API 610 (ISO 13709) requires 25,000 h with continuous operation at rated conditions and at least 16,000 h at maximum radial and axial loads and rated speed.

It is not uncommon in some municipal specifications to find a requirement of 100,000 h bearing life. This may be suitable and achievable for pumps with low operating speeds, but it could result in severely oversizing bearings for pumps operating at 2 pole motor speeds. Bearing manufacturer catalogues provide maximum operating speeds for rolling-element bearings, but general practice is to select bearings that are below 75% of published values for centrifugal pumps. The higher the operating speed, the more dynamic loads are placed on the bearing-rolling elements, which require near-ideal steady state conditions to operate properly and achieve predicted life. Since pump shafts have variable loads and are subject to deflection and vibration, it is best to select bearings with a speed margin.

Mounting Most rolling-element bearings should be retained on the shaft with a light interference fit and fitted into the housing with a small diametral clearance. Note that some roller bearing designs, such as cylindrical roller types with separable races, require a light interference fit in the housing bore to work properly. Guidelines for the amount of interference fit or clearance recommended for each bearing type and loading are given in ABMA Standard 7. Bearings should be mounted directly on the shaft whenever possible. Bearing carriers that are sometimes used between the bearings and the shaft can be a problem if tolerances and clearances are not carefully controlled and should be avoided. It is preferred that bearings are located axially on the shaft using shoulders, collars, or other positive locating devices such as split rings. Snap rings and spring-type washers are generally not acceptable as they usually fit into grooves in the shaft that contain sharp radii, resulting in severe shaft stress risers. Thrust bearings should always be locked to shafts with a matching bearing nut with a tongue-type lock washer.

Bearings should be heated with an induction-type bearing heater or in a heated oil bath prior to mounting on the shaft. A torch or flame should never be used to heat a bearing. It is difficult to uniformly heat the bearing and to control the metal temperature with a flame, which likely will result in distorting the bearing or changing the metal properties. Heating the bearing to 250°F is sufficient to provide a slip fit of the bearing onto the shaft. If nonmetallic retainers are allowed, extra care must be exercised in the heating process. Thermally insulated gloves must be used when handling hot bearings.

Operation Heat is generated during bearing operation for various reasons. The lubricant is subject to shear forces as it circulates in the bearing. Excess oil in the bearings increases churning and shearing resulting in elevated operating temperatures. Energy from

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deflection of the bearing metal occurs on both the bearing races and the rolling element itself, so a highly loaded bearing will run hotter than a lightly loaded bearing. Friction occurs from the rolling element as it revolves in the retainer pocket. Friction occurs from any relative motion or sliding of the rolling element to the bearing race. Bearings whose rolling elements are skidding run very hot. Highly preloaded bearings also tend to run hotter than lightly preloaded or non-preloaded bearings, provided no skidding is occurring.

The operating speed, the type of rolling element, angular operation of a rolling element, the type of bearing retainer, the number of rolling elements, and ambient temperature can also influence operating temperature. Higher speeds result in increased lubricant shear and retainer friction. Roller-type bearings increase lubricant shear, retainer friction, and sliding friction compared to ball-type bearings. Bearings whose rolling elements operate at an angle such as angular contact ball bearings and spherical roller bearings are more prone to sliding and skidding which increases friction. These types of bearings also utilize a heavy-duty retainer that operates in close proximity to the rolling element, resulting in more lubricant shear and friction. Maximum capacity bearings (when allowed) have a higher quantity of rolling elements than standard-capacity bearings, which increases lubricant shear and friction. Bearing operating temperature is approximately proportional to ambient temperature.

As can be surmised from the above discussion, bearing operating temperatures vary widely. Bearing manufacturers generally claim a maximum allowable bearing metal operating temperature of 250°F (121°C), but this is rarely utilized in practice. The practical limit of operation is considered to be 200°F (93°C), though there are many successful applications operating at higher temperatures. 180°F (82°C) is the upper temperature limit of mineral-type oils to achieve a reasonable lubricant life. Mineral oils operating above this temperature oxidize very rapidly, resulting in what has become known in the industry as black oil. Synthetic oils are utilized successfully at higher bearing temperatures. The following operating temperature limits have been established for API Standard 610 (ISO 13709) pumps, based on an ambient temperature of 110°F (43°C):

- For ring-oiled or splash systems, the oil sump temperature should not exceed 180°F (82°C). Bearing outer ring temperatures should not exceed 200°F (93°C).
- For pressurized lubrication systems, the outlet oil temperature should not exceed 160°F (70°C) and bearing metal temperatures should not exceed 200°F (93°C).
- During a shop test with a pressurized system, under the most severe operating conditions specified, the bearing oil temperature rise should not exceed 50°R (28 K).

REFERENCES

1. American Petroleum Institute Standard 610. "Centrifugal Pumps for Petroleum, Petrochemical and Natural Gas Industries." 10th Edition, October 2004, www.api.org
2. Specifications for Centrifugal Pumps for Chemical Process: ANSI/ASME B73.1M–1991, Horizontal End Suction; American Society of Mechanical Engineers, New York, NY www.asme.org
3. ISO 281, "Rolling bearings—Dynamic load ratings and rating life."
4. ISO 5753, "Rolling bearings—Radial internal clearance."
5. ISO 13709:2003, "Centrifugal pumps for petroleum, petrochemical and natural gas industries."

SECTION 6.2
OIL FILM JOURNAL BEARINGS

WILBUR SHAPIRO

PRINCIPLES OF OPERATION

A journal bearing is essentially a viscous pump, and it derives load capacity by pumping the lubricant through a small clearance region. In Figure 1, the fluid is dragged along by the rotating journal. To generate pressure, the resistance to pumping must increase in the direction of the flow. In the figure, the journal moves to form a converging tapered clearance in the direction of the rotation or flow.

The eccentricity e is the total displacement of the journal from its concentric position. The attitude angle γ in Figure 1 is the angle between the load direction and the line of centers. Note that, because of the necessity to form a converging wedge, the displacement of the journal is not along a line that is coincident with the load vector. A positive pressure is produced in the converging region of the clearance. Downstream from the minimum film thickness, which occurs along the line of centers, the film becomes divergent. The resistance decreases in the direction of pumping, and either negative pressures occur or the air in the lubricant gasifies or cavitates and a region of atmospheric pressure occurs in the bearing area. This phenomenon is known as *fluid film bearing cavitation*. It should be clearly distinguished from other forms of cavitation that take place in pumps, such as in the impeller, for example. Here the fluid is traveling at a high velocity and the inertia forces on each fluid element dominate. Implosions occur in the impeller and can cause damage.

In a bearing, the viscous forces dominate and each fluid particle moves at a constant velocity in proportion to the net shearing forces on it. Thus, cavitation in a bearing is more of a change of the phase of the lubricant that occurs in a region of lower pressure that permits the release of entrained gases. Generally, bearing cavitation does not cause damage.

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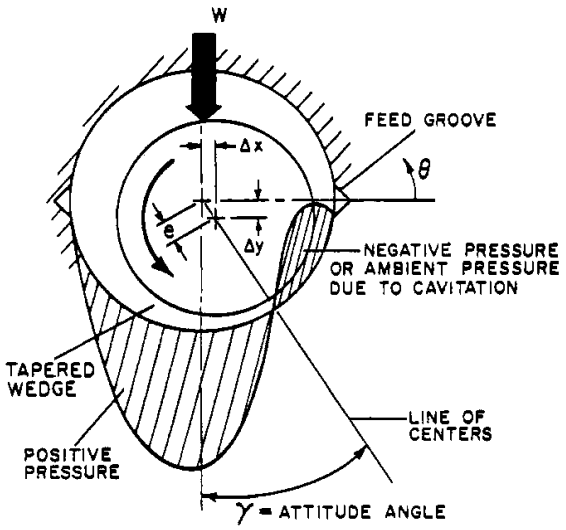


FIGURE 1 Two-groove cylindrical bearing

Regimes of Lubrication Whether or not a fluid film can be formed, journal rotation is dependent on several factors including the surface speed, viscosity, and load capacity. A parameter[1] is often used to determine a particular regime of lubrication, $ZN/\bar{P}$, where

Z = viscosity of lubricant, cP (Pa · s)
N = rotating speed, rpm
$\bar{P}$, = average pressure of the bearing, lb/in^2 (bar)*

A plot of the coefficient of friction versus $ZN/\bar{P}$ generally has the form shown in Figure 2. At low values of $ZN/\bar{P}$, a combination of viscosity, speed, and load places a bearing in a boundary lubricated regime where typical coefficients of friction are 0.08 to 0.14. Boundary lubrication implies intimate contact between the opposed surfaces. As the value of the parameter increases as a result of the increased speed, increased viscosity, or lowered load, there is a dramatic reduction in the coefficient of friction. In this region, there is a mixed film lubrication and the coefficient of friction varies between 0.02 and 0.08. By mixed film lubrication, it is meant that the journal is partly surrounded by a fluid film and is partly

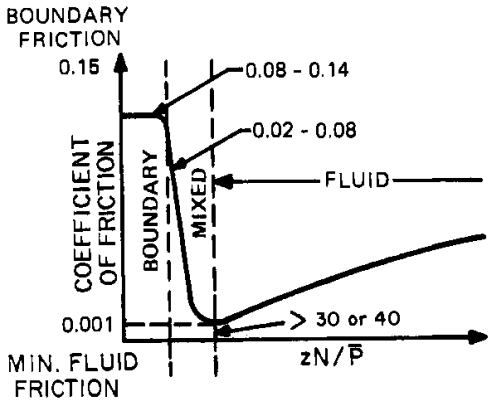


FIGURE 2 Coefficient of friction versus $ZN/\bar{P}$ (Ref. 1)

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

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supported by rubbing contact between the opposed members. As $ZN/\bar{P}$ increases further, a situation of full fluid film lubrication prevails.

A general rule of thumb is that $ZN/\bar{P}$ should be 30 (0.44) or greater for a fluid film to be generated. Note that as $ZN/\bar{P}$ continues to increase beyond the full film demarcation, the coefficient of friction rises, but at a relatively low rate and generally remains in regions of low coefficients of friction.

EXAMPLE

$$Z = 30 \text{ cP } (0.03 \text{ Pa} \cdot \text{s})$$

$$N = 150 \text{ rpm}$$

$$\bar{P} = 200 \text{ lb/in}^2 \text{ (13.6 bar)}$$

$$\frac{ZN}{\bar{P}} = \frac{30 \times 150}{200} = 22.5$$

The bearing is not fluidborne and is operating in the mixed film regime. At what speed will the bearing become hydrodynamic? For hydrodynamic operations, $ZN/\bar{P} = 30$. Therefore,

$$N = \frac{30\bar{P}}{Z} = \frac{30 \times 200}{30} = 200 \text{ rpm}$$

THEORETICAL FOUNDATIONS

The foundation of a fluid film-bearing analysis emanates from the boundary layer theory of fluid mechanics. The governing differential equation was first formulated by Osborne Reynolds in 1886 and is known as Reynolds' equation in his honor. It has been only in the last 30 years or so that general solutions have been obtained, and this has been primarily due to the use of numerical methods applied to the digital computer. References 2 and 3 go into the details of contemporary numerical solutions and are recommended for those interested in the analytical aspects of lubrication.

Principal Assumptions Reynolds' equation can be derived from the Navier-Stokes equation of fluid mechanics, and a number of textbooks are available that comprehensively describe the derivation.[4] The primary assumptions are as follows:

- Laminar flow conditions prevail, and the fluids obey a Newtonian shear stress distribution where the shear stress is proportional to the velocity gradient.
- Inertial forces, resulting from acceleration of the liquid, are small relative to the viscous shear forces and may be neglected.
- The pressure across the film is constant since the fluid films are so thin.
- The height of the fluid film is small relative to other geometric dimensions, and so the curvature of the fluid film can be ignored.
- The viscosity of the liquid remains constant. In most cases, this is a reasonable assumption since it has been repeatedly demonstrated that, if the average viscosity is used, little error is introduced and the complexity of the analysis is considerably reduced.

Derivaton of Reynold's Equation of Lubrication Assume that the rotating journal has a peripheral velocity U. Consider an elemental volume in the clearance space of the bearing and establish equilibrium (see Figure 3). Note that since inertial forces are neglected, the volume is in equilibrium by the pressure and shear forces acting upon it, so there is no acceleration. As shown in Figure 4, p is the pressure and τ is the shear stress acting upon the volume. Summing forces in the x direction

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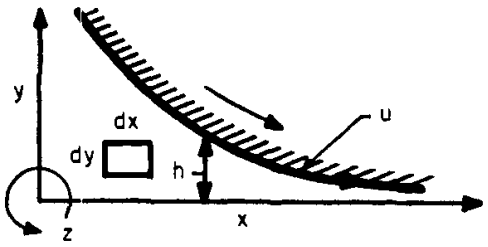


FIGURE 3 Fluid control volume

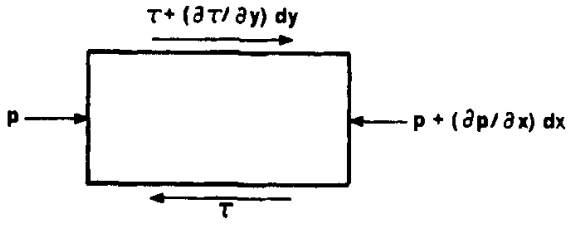


FIGURE 4 Force equilibrium on fluid element

$$p\,dy\,dz - \left(p + \frac{\partial p}{\partial x}\,dx\right)dy\,dz - \tau\,dx\,dz + \left(\tau + \frac{\partial \tau}{\partial y}\,dy\right)dx\,dz = 0 \tag{1}$$

$$-\frac{\partial p}{\partial x}\,dx\,dy\,dz + \frac{\partial \tau}{\partial y}\,dx\,dy\,dz = 0$$

or

$$\frac{\partial p}{\partial x} = \frac{\partial \tau}{\partial y} \tag{2}$$

For a Newtonian fluid in a laminar flow, the shear stress is directly related to the velocity gradient with the proportionality constant being the absolute viscosity μ (see Figure 5):

$$\tau = \mu \frac{dv}{dy} \tag{3}$$

$$\frac{\partial \tau}{\partial y} = \mu \frac{\partial^2 v}{\partial y^2}$$

where v is the fluid velocity. Substituting Eq. 3 into Eq. 2, we obtain

$$\frac{\partial p}{\partial x} = \mu \frac{\partial^2 v}{\partial y^2} \tag{4}$$

Integrating with respect to y twice produces the following equation:

$$v = \frac{1}{\mu}\,\frac{\partial p}{\partial x}\,\frac{y^2}{2} + C_1 y + C_2 \tag{5}$$

where C_1 and C_2 are constants of integration. The boundary conditions are

$$v = 0, y = 0 \text{ and} \tag{6}$$

$$v = u, y = h$$

Substituting the boundary conditions of Eq. 6 into Eq. 5 results in the following expression for v:

$$v = \frac{1}{2\mu}\,\frac{\partial p}{\partial x}\,(y^2 - h^2 y) + \frac{uy}{2} \tag{7}$$

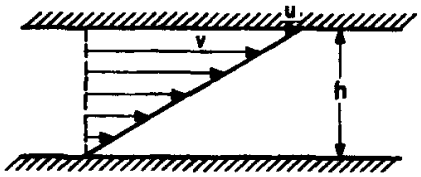


FIGURE 5 Laminar velocity distribution across film

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The velocity in the z direction would be similar, except that the surface velocity term would be omitted because no surface velocity exists in the z direction. In addition, the pressure gradient would be with respect to z.

Now let us consider the flow across the film due to this velocity. Note that Eq. 7 is the velocity computed in the x direction, which is in the direction of rotation of the journal:

$$q_x = \int_0^h v_x \, dy = \int_0^h \left[\frac{1}{2\mu} \frac{\partial p}{\partial x} (y^2 - h^2 y) + \frac{uy}{2} \right] dy \tag{8}$$

After integrating,

$$q_x = -\frac{1}{12\mu} \frac{\partial p}{\partial x} h^3 + \frac{1}{2} uh \tag{9}$$

Note that q_x is the flow per unit width across the film. The flow in the axial direction is

$$q_z = -\frac{1}{12\mu} \frac{\partial p}{\partial z} h^3 \tag{10}$$

Now let us consider a flow balance through an elemental volume across the film (see Figure 6). The net outflow through the volume equals the net reduction in volume per unit time:

$$\left(q_x + \frac{\partial q_x}{\partial x} dx \right) dz - q_x \, dz + \left(q_z + \frac{\partial q_z}{\partial z} dz \right) dx - q_z \, dx = -\frac{\partial h}{\partial t} dx \, dz \tag{11}$$

Thus, this gives us

$$\frac{\partial q_x}{\partial x} + \frac{\partial q_z}{\partial z} = -\frac{\partial h}{\partial t} \tag{12}$$

Substituting Eqs. 9 and 10 into Eq. 12, we obtain

$$-\frac{\partial}{\partial x} \left(\frac{1}{12\mu} h^3 \frac{\partial p}{\partial x} \right) - \frac{\partial}{\partial z} \left(\frac{1}{12\mu} h^3 \frac{\partial p}{\partial z} \right) = -\frac{\partial h}{\partial t} - \frac{u}{2} \frac{\partial h}{\partial x} \tag{13}$$

and the final equation becomes

$$\frac{\partial}{\partial x} \left(\frac{h^3}{\mu} \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial z} \left(\frac{h^3}{\mu} \frac{\partial p}{\partial z} \right) = 6\mu \frac{\partial h}{\partial x} + 12 \frac{\partial h}{\partial t} \tag{14}$$

Equation 14 is the general form of Reynolds' equation used for laminar, two-dimensional lubrication problems.

Reynolds' equation is a flow balance equation. The left-hand side represents pressure-induced flows in the x and z directions through the differential element. The first term on the right-hand side represents the shear flow of the fluid induced by the surface velocity of the journal u. Note that this term contains the derivative of clearance with respect to distance. If this term is zero, then there is zero pressure produced by hydrodynamic action,

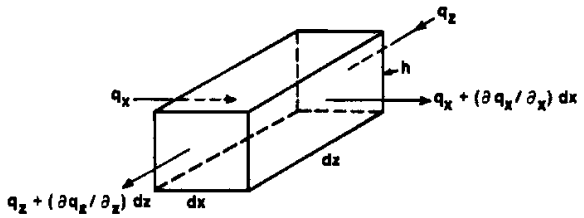


FIGURE 6 Flow balance across control volume

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and the term $\partial h/\partial x$ is the mathematical representation of the tapered wedge. The second term on the right-hand side refers to a time rate of change of the film thickness, which can be translated to a normal velocity of the center of the journal. It produces pressure by a fluid velocity normal to the bearing surfaces that attempts to squeeze fluid out of a restricted clearance space. This phenomenon is called the squeeze film effect in bearing terminology. Since it is proportional to the velocity of the center of the journal, it is the phenomenon that produces viscous damping in a bearing.

The solution to Reynolds' equation (refer to Eq. 14) provides the pressure at all points in the bearing. The application of the digital computer has enabled a rapid solution of Reynolds' equation over a grid network representing the bearing area.[2,3]

Once the pressures have been obtained, a numerical integration is applied to determine the performance parameters (in other words, the load capacity):

$$w = \int\int prd\theta\, dr \tag{15}$$

The flow across any circumferential line is

$$q_\theta = \oint\left(-\frac{1}{12\mu}\frac{\partial p}{r\partial\theta}h^3 + \frac{1}{2}uh\right)dz \tag{16}$$

The flow across any axial line is

$$qz = \oint\left(-\frac{1}{12\mu}\frac{\partial p}{\partial z}h^3\right)rd\theta \tag{17}$$

The viscous frictional moment is obtained by integrating the shear stress over the area and can be shown to be

$$M_f = \int\int\left[\frac{1}{r}\frac{\partial p}{\partial\theta}\frac{h}{2} + \frac{\mu r\omega}{h}\right]r^2 d\theta\, dz \tag{18}$$

where ω = journal surface speed, rad/s.

Typical computer program output includes the following:

- Pressure distribution throughout the grid network
- Load capacity
- Side leakage and carryover flows
- Viscous power losses
- Righting moments due to misalignments
- Attitude angles
- Cross-coupled spring and damping coefficients due to displacements and velocity perturbations of the journal center
- Clearance distribution

Turbulence Equation 14 is for laminar conditions. For very high speed bearings, operations beyond the turbulent regime may occur and Reynolds' equation must be modified. The turbulent theory has been developed, and the literature on this topic can enable performance predictions for turbulent bearings.[5,6]

The onset of turbulence is determined by examining the bearing's Reynolds number, which is the ratio of inertia to viscous forces and is defined as

$$Re = \frac{\rho uh}{\mu} \tag{19}$$

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where Re = Reynolds number
ρ = fluid density, lb · s²/in⁴ (kg/m³)
u = surface velocity, in/s (m/s)
h = local film thickness, in (m)
μ = viscosity, lb · s/in² (Pa · s)

A reasonable approximation is to use the concentric clearance c for h. In terms of N rpm and journal diameter D, the Reynolds number is

$$Re = \rho \frac{\pi DN}{60} \frac{c}{\mu} \tag{20}$$

The criterion for turbulence in journal bearings is that $Re \geq 1000$.

EXAMPLE As an example, consider the following:

Journal diameter $D = 5$ in (127 mm)
Bearing length $L = 5$ in (127 mm)
Radial clearance $c = 0.0025$ in (0.064 mm)
Operating speed $N = 5000$ rpm
Lubricant viscosity $\mu = 2 \times 10^{-6}$ lb · s/in² (14×10^{-3} Pa · s)
Lubricant density $\rho = 7.95 \times 10^{-5}$ lb · s²/in⁴ (8.66×10^{-11} kg · s²/mm⁴)

$$Re = \rho \frac{\pi DN}{60} \frac{c}{\mu}$$

$$= 7.95 \times 10^{-5} \times \frac{\pi \times 5 \times 5000}{60} \times \frac{0.0025}{2 \times 10^{-6}} = 130$$

Thus, the bearing is operating in the laminar regime. To become turbulent (assuming constant viscosity), the operating speed would have to increase to approximately 38,500 rpm.

The example cited is for a relatively high-speed, oil-lubricated bearing for pump applications. In general, most pump bearings operate in the laminar regime. Exceptions might occur when water is used as tbe lubricant because it its much less viscous than oil.

Evaluation of Frictional Losses It is often desirable to obtain a quick estimate of viscous drag losses that the journal bearings produce. If we consider shear forces again, we return to the laminar flow equation:

$$F = \mu A \frac{\mu}{h} \tag{21}$$

where F = viscous shear force, lb (N)
μ = viscosity, lb · s/in² (Pa · s)
A = surface area, in² (m²)
u = journal velocity, in/s (m/s)
h = film thickness, in (m)

To obtain friction, we multiply both sides of Eq. 21 by the journal radius R. Then the viscous frictional moment is

$$M = \mu AR \frac{u}{h}$$

The frictional horsepower loss is

$$FHP = \frac{NM}{63{,}000}$$

where N = rotating speed, rpm
M = moment, lb · in (N · m)

Also

A = surface area of bearing = πDL, in^2 (m^2)

u = surface speed = $\frac{\pi DN}{60}$ in/s (m/s)

Substituting, we obtain

$$FHP = \frac{\mu L D^3 N^2}{766{,}000h} \tag{22}$$

In obtaining approximate losses for estimation purposes, the concentric clearance c is substituted for the local film thickness h.

If we consider the previous example where D = 5 in (127 mm), L = 5 in (127 mm), c = 0.0025 in (0.064 mm), N = 5000 rpm, and $\mu = 2 \times 10^{-6}$ lb · s/in^2 (14×10^{-3} Pa · s), the horsepower loss is

$$FHP = \frac{(2 \times 10^{-6})(5)(5)^3(5000)^2}{(766{,}000)(0.0025)} = 16.32 \text{ hp}$$

Note that for thrust bearings, the frictional horsepower loss is

$$FHP = \frac{\mu N^2}{h} \frac{OD^4 - ID^4}{6.127 \times 10^6} \tag{23}$$

where OD = outside diameter
ID = inside diameter

A general rule of thumb is that the frictional horsepower in a thrust bearing is approximately twice that in a journal bearing.

BEARING TYPES

Cylindrical Bearing The most common type of journal bearing is the plain cylindrical bushing shown schematically in Figure 1. It can be split and have lubricating feed grooves at the parting line. A ramification is to incorporate axial grooves to enable better cooling and to improve whirl stability (described in more detail below in the discussion of cylindrical bearings with axial grooves). The principle advantages of cylindrical bearings are (1) simple construction and (2) a high-load capacity relative to other bearing configurations.

This type of bearing also has several disadvantages:

- *Whirl Instability:* This is prone to subsynchronous whirling at high speeds and also at low loads. Whirling is an orbiting of the journal (shaft) center in the bearing, a motion that is superimposed upon the normal journal rotation. The orbital frequency is approximately half the rotating speed of the shaft. The expression *half-frequency whirl* is commonly used. The reason for the occurrence of this whirl and more details concerning bearing dynamics are presented in the section on bearing dynamics.
- *Viscous Heat Generation:* Because of the generally large and uninterrupted surface area of this bearing, it generates more viscous power loss than some other types.

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- *Contamination:* The cylindrical bearing is more susceptible to contamination problems than other types because contaminants that are dragged in at the leading edge of the bearing cannot easily dislodge because of the absence of grooves or other escape paths.

The advantages of simplicity and load capacity make the plain journal a leading candidate for most applications, but performance should be carefully investigated for whirl instability and potential thermal problems. Cylindrical bearings are generally used for medium-speed (500 in/s [200 mm/sec] surface speed) and medium- to heavy-load applications (250 to 400 lb/in^2 [17 to 28 bar] on a projected area).

Cylindrical Bearing with Axial Grooves A typical configuration of this type of bearing is a plain cylindrical bearing with four equally spaced longitudinal grooves extending most of the way through the bearing. Usually, a slight land area exists at either end of the groove to force the inlet flow to each groove into the bearing clearance region (see Figure 7), rather than out the groove ends. This configuration is a little less simple than the plain cylindrical bearing, and because the grooves consume some land area, this configuration has less load capacity than the plain bushing. Since oil is fed into each of the axial grooves, this bearing requires more inlet flow but also will run cooler than the plain bushing. The grooves act as convenient outlets for any contaminants in the lubricant, and thus the grooved bearing can tolerate more contamination than the plain cylindrical bearing.

In general, this bearing can be considered as an alternate to a plain bearing if the former can correct a whirl or overheating problem.

Elliptical and Lobe Bearings Elliptical and lobe bearings have noncircular geometries. Figure 8 shows two types of three-lobe bearings with the clearance distribution exaggerated so that the lobe geometry is easily discernible. An elliptical bearing is simply a two-lobe bearing with the major axis along the horizontal axis.

The lobe bearing shown in Figure 8a is a symmetric lobe bearing where the minimum concentric clearance occurs in the center of each lobed region. Thus, at the leading edge region, a converging clearance produces positive pressure, but downstream from the minimum film thickness, a divergent film thickness distribution can be found with resulting negative, or cavitation, pressures.

The canted lobe in Figure 8b, on the other hand, generally develop positive pressure throughout the lobe because the bearing is constructed with a completely converging film thickness in each lobed region. This design has excellent whirl resistance (superior to that of the symmetric lobe bearing) and a reasonably good load capability. A 2:1 ratio between leading and trailing edge concentric clearance is generally a reasonable compromise with respect to performance.

Elliptical and lobe bearings are often used because they provide better resistance to whirls than cylindrical configurations. They do so because they have multiple load-producing pads that assist in preventing large-attitude angles and cross-coupling (see the section on bearing dynamics). Elliptical and lobe bearings are generally used for high-speed, low-load applications where whirls might be a problem.

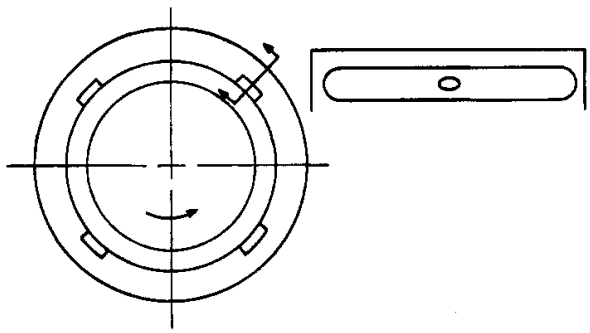


FIGURE 7 Cylindrical bearing with axial grooving

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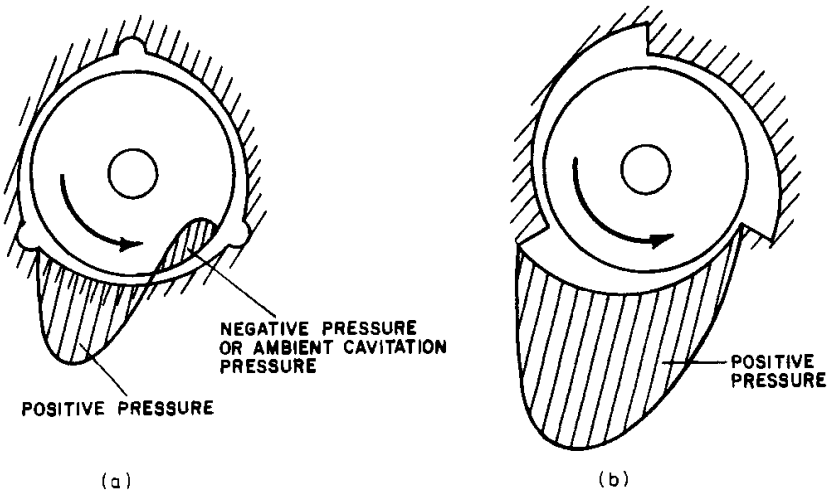


FIGURE 8 (a) symmetric lobe bearing and (b) canted lobe bearing

Elliptical, or two-lobe, bearings generally have poor horizontal stiffness because of the large clearances along the major diameter of the ellipse. The split elliptical configuration, however, is easier to manufacture than the other types because it is two cylindrical bearing halves with material removed along the parting line. Lobe bearings are usually clearance- and tolerance-sensitive. The other types of lobe bearings are complicated to manufacture.

Tilting-Pad Bearings Tilting-pad bearings are used extensively, especially in high-speed applications, because of their whirl-free characteristics. They are the most whirl-free of all bearing configurations.

An important geometric variable for tilting-pad bearings is the preload ratio, defined as shown in Figure 9.

The preload ratio equals

$$\text{Preload ratio} = PR = \frac{c - c'}{c} = 1 - \frac{c'}{c} \tag{24}$$

where c = machined clearance
c' = concentric pivot film thickness

The variable c' is an installed clearance and is dependent upon the radial position of the pivot. Figure 10 displays two pads. Pad 1 has been installed such that the preload ratio

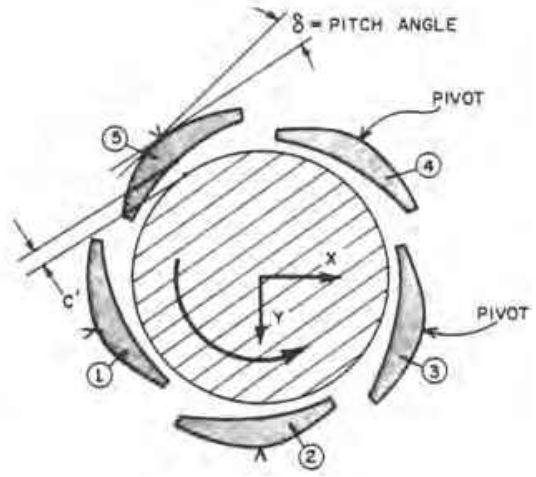


FIGURE 9 Five-pad tilting pad bearing

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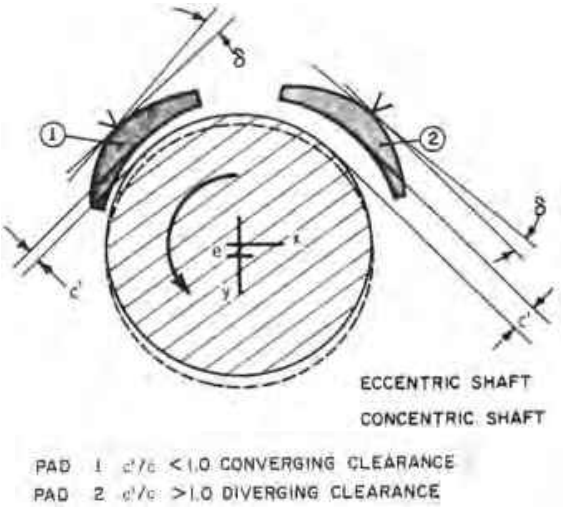


FIGURE 10 Tilting-pad bearing preload

is less than one. For pad 2, the preload ratio is one. The solid line represents the position of the journal in the concentric position. The dashed portion of the journal represents its position when a load is applied to the bottom pads (not shown). Pad 1 is operating with a good converging wedge, even though the journal is moving away from it. Pad 2, on the other hand, is operating with a completely diverging film, which means that it is totally unloaded. Thus, bearings with installed pad preload ratios of one or greater will operate with unloaded pads, which reduces overall stiffness of the bearing and results in a deterioration of stability because the unloaded pads do not aid in resisting cross-coupling influences. In the unloaded position, they are also subject to flutter instability and to a phenomenon known as *leading edge lockup,* where the leading edge is forced against the shaft and is maintained in that position by the frictional interaction of the shaft and the pad. This is especially prevalent in bearings that operate with low-viscosity lubricants, such as gas or water bearings. Thus, it is important to design bearings with preload, although for manufacturing reasons it is common practice to produce bearings without preload.

Tilting-pad bearings have some other characteristics that are both positive and negative:

- They are not as clearance-sensitive as most other bearings.
- Because the pads can move, they can operate safely at a lower minimum film thickness than other bearings.
- They do not provide as much squeeze film damping as rigid configurations.
- Generally, they are more expensive than other bearings.
- For high-speed applications, their pivot contacts can be subjected to fretting corrosion.

Hybrid Bearings A hybrid bearing, schematically shown in Figure 11, derives a load capacity from two sources: (1) the normal hydrodynamic pressure generation and (2) an external high-pressure supply that introduces oil into recesses machined into the bearing surface via restrictors (orifices or capillaries upstream of the recesses). External pressure significantly enhances load capacity. Also, these bearings have excellent low- or zero-speed load capabilities. They are sometimes used as startup devices to lift off the rotor. When self-sustaining hydrodynamic speeds are attained, the external pressure is shut off. The characteristics of externally pressurized, or hybrid, bearings include the following:

- High load and stiffness capabilities
- An external flow that assists in cooling

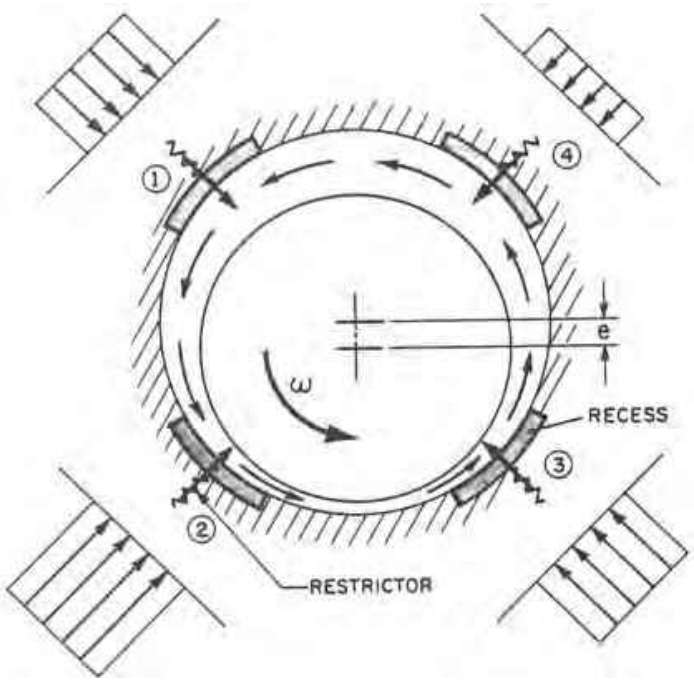


FIGURE 11 Cross-coupling influences in hybrid bearings

- Their clearances and tolerances are generally more liberal than in hydrodynamic bearings
- They require external fluid-supply systems
- They are applied when there is not a sufficient generating speed or when a high-load capacity and stiffness are required
- They are sometimes applied to prevent a whirl, but rotational speeds can unbalance recess pressures, introduce cross coupling, and promote a whirl

STEADY STATE PERFORMANCE

Computer-generated performances have been obtained for most of the bearing types previously discussed. Information has been plotted in a nondimensional format so that no restrictions exist on operating conditions, lubricant properties, and so on. Use of the charts will be subsequently demonstrated by numerical example.

Viscosity One of the key parameters in determining the performance of a bearing is the lubricant viscosity. Viscosity characteristics of commonly used Society of Automotive Engineers (SAE) grades of oil are shown in Figure 12. The units of viscosity are *microreyns*, where the reyn has the units of lb · s/in^2 and comes from the ratio of shear stress to the velocity gradient across the film, as indicated by Eq. 21. Other units of viscosity are *centipoises*, *Saybolt seconds universal* (SSU), and *centistokes*. The conversion factors are as follows:

$$\mu(\text{reyns}) = Z \times 1.45 \times 10^{-7} \tag{25}$$

$$\nu(\text{centistokes}) = 0.22\,(\text{SSU}) - \frac{180}{\text{SSU}} \tag{26}$$

$$Z(\text{centiposes}) = \nu(\text{centistokes}) \times \text{SG (sp. gr.)} \tag{27}$$

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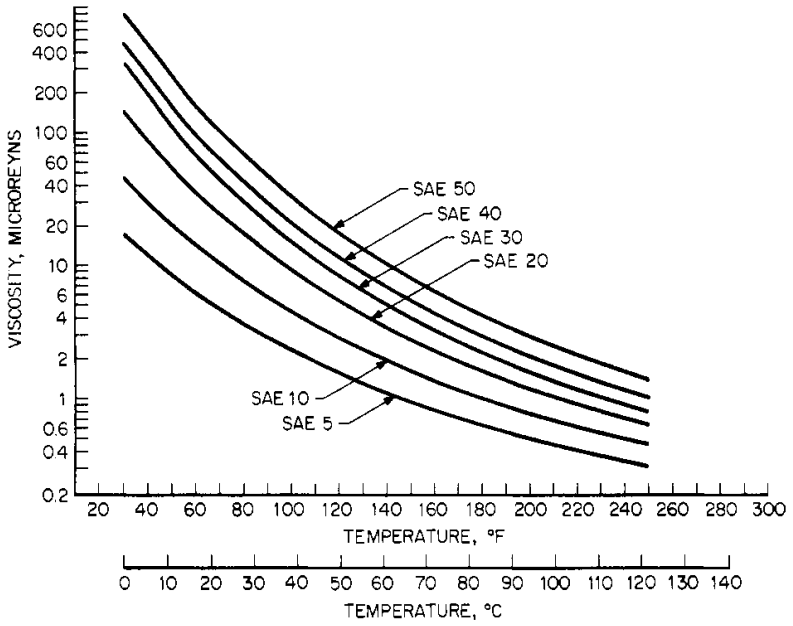


FIGURE 12 Viscosity characteristics for SAE oil grades

Performance Curves Performance plots have been generated for the following types of bearings:

- *Two-groove cylindrical bearings*
- *Symmetric three-lobe bearings:* Each lobe is offset such that in the concentric position the minimum film thickness in the center of each lobed region is half the machined clearance c (see definition following). The pads are each 110° in the angular extent.
- *Canted three-lobe bearing:* The lobing is canted such that in the concentric position the leading edge clearance was twice the trailing edge and the trailing edge film thickness (minimum) in the concentric position was $0.5c$ where c equals the machined clearance. The pads are each 110° in the angular extent.
- *Tilting-pad bearing:* The tilting-pad bearing that information is obtained for is a five-pad bearing with a 60° pad and a preload ratio of 30%.

Two length/diameter ratios are examined for each type of bearing: $L/D = 0.5$ and 1.0. The definition of the nondimensional parameters is as follows:

$$W = \text{nondimensional load parameter} = wc^2/6\mu\omega RL^3 \quad (28)$$

$$P = \text{nondimensional viscous power loss parameter} = 1100cp/\mu(\omega RL)^2 \quad (29)$$

$$Q = \text{nondimensional flow parameter} = 2q/0.26\omega RLc \quad (30)$$

$$HM = \text{nondimensional minimum film thickness} = h_M/c \quad (31)$$

where w = bearing load capacity, lb (N) and
c = reference clearance (machined clearance = radius of bearing − radius of shaft), in (mm)
μ = absolute viscosity, reyns (lb · s/in^2) (cP)
ω = shaft or journal rotational speed rad/s
R = shaft radius, in (mm)
L = bearing length, in (mm)
p = viscous power loss, hp (kW)
q = flow, gpm (m^3/h)

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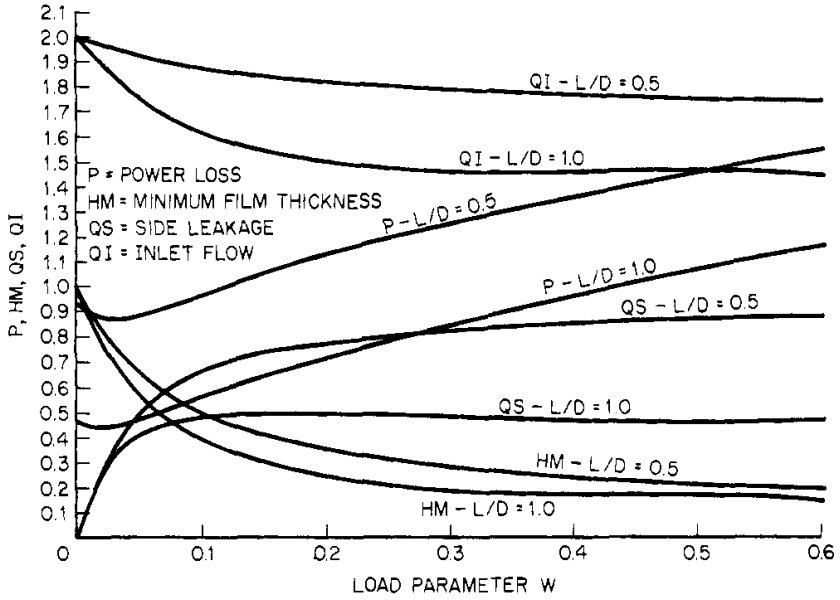


FIGURE 13 Performance characteristics for two-groove cylindrical bearings

q_i = inlet flow to the leading edge of bearing (for multipad bearings, equals the sum of inlet flow to each pad), gpm (m^3/h)
q_s = side leakage flow or flow out of the bearing ends (for multipad bearings, equals the sum of side leakage flow of each pad), gpm (m^3/h)
h_M = minimum film thickness in bearing, in (mm)

Performance curves are shown in Figures 13 through 17.

At times, the nondimensional data can be confusing and lead to erroneous judgments. For example, the nondimensional power loss P is greater for an L/D equal to 0.5 than for an L/D equal to 1.0. However, when the dimensional value of the power loss is being computed, the nondimensional value is multiplied by L^2. Therefore, the power loss for the L/D

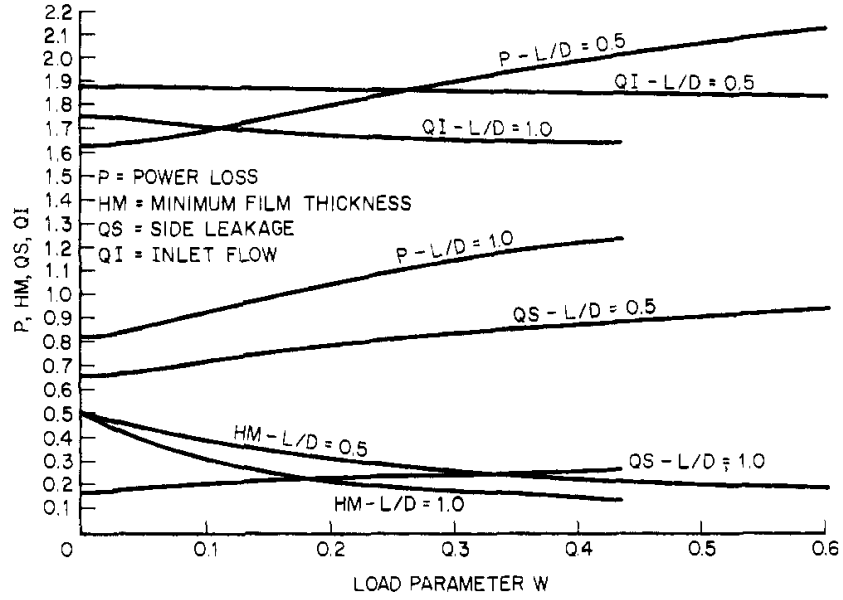


FIGURE 14 Performance characteristics for three-lobe bearings

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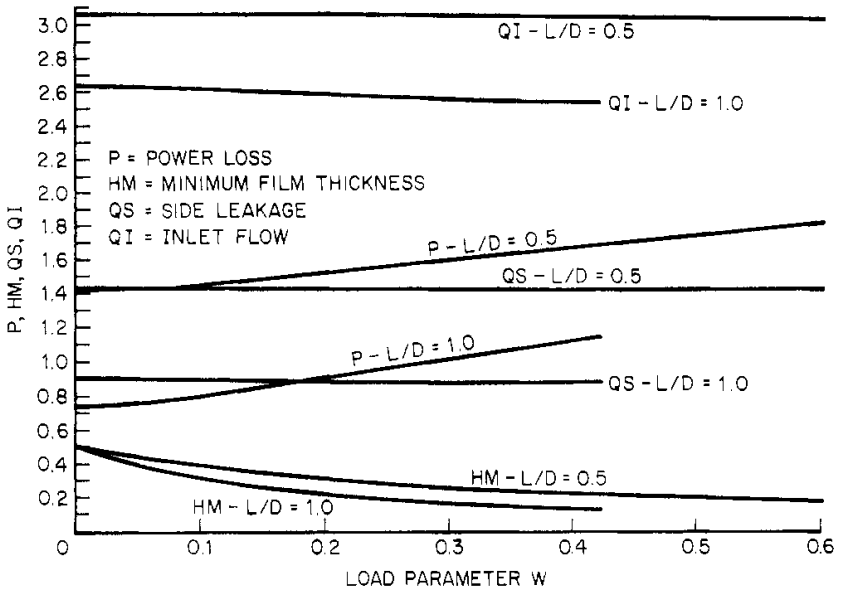


FIGURE 15 Performance characteristics for canted three-lobe bearings

equal to 1.0 will be, as expected, greater than for the L/D equal to 0.5. If the reader uses the data as presented, the dimensional information will prove consistent.

To make comparisons among the bearings, using the nondimensional data is not strictly proper because there may be slight inconsistencies in preloads, the bearings will not be operating at the same average viscosity, and so on. Subsequently, dimensional data derived from the performance curves will be compared, but comparisons of the nondimensional information will provide an indication of performance parameters among the bearings. Comparisons have been made at equal values of the load parameter W and the results are shown in Table 1.

Comparisons of the different bearing types should be made only at the same L/D ratio because of the anomalies (discussed above) of nondimensional parameters that occur at

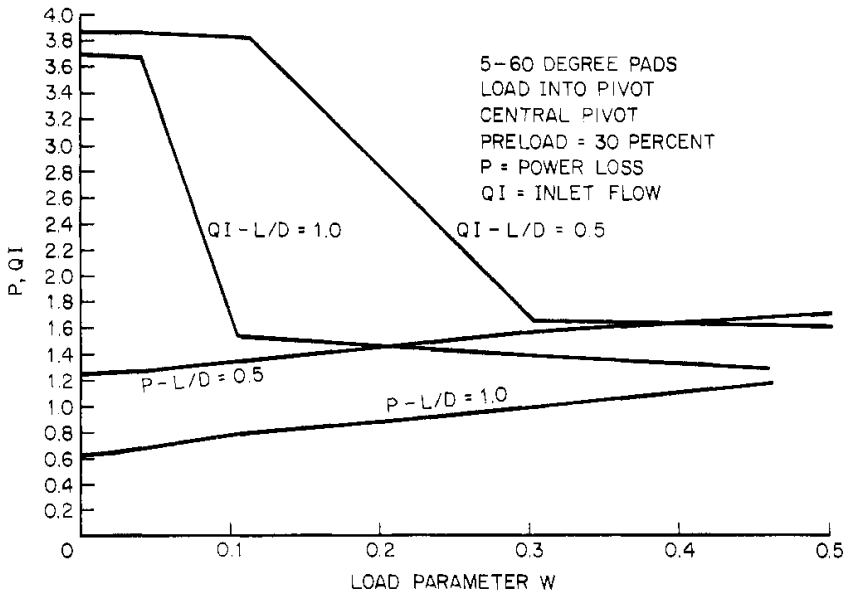


FIGURE 16 Performance characteristics for tilting-pad bearings

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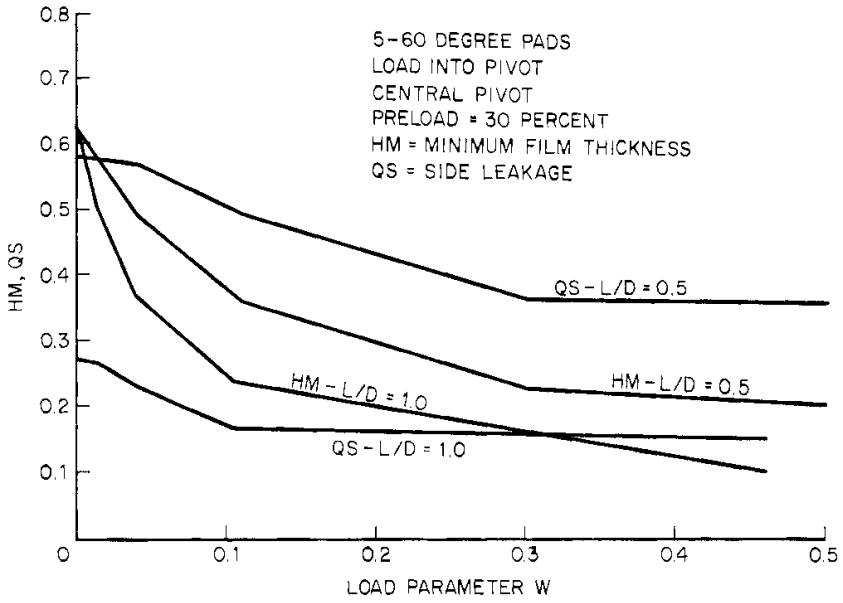


FIGURE 17 Performance characteristics for tilting-pad hearings

different L/D ratios. If we assume that all the reference variables that go into the nondimensional parameters are identical, we can establish the following conclusions:

- The two-groove cylindrical hearing has the highest film thickness and thus the highest load capacity.
- The symmetric three-lobe bearing has the highest power loss.
- The canted three-lobe bearing has the greatest flow requirements.

Note that these comparisons were made on the basis of steady-state performances only. The major reason for applying lobe and tilting-pad bearings is to avoid dynamic instabilities.

Heat Balance Performance is based upon the assumption of a uniform viscosity in the fluid film. Since the viscosity is a strong function of temperature and since the temperature rise of the lubricant due to viscous heat generation is not known a priori, an iterative procedure is required to determine the average viscosity in the film. To determine an average viscosity, there must be a simplified heat balance in the film. The assumption is made that all the viscous heat generated in the film is absorbed by the lubricant as it flows through the film and produces a temperature rise.

TABLE 1 Comparative Results of Bearing Types at $W = 0.2$

	$L/D = 0.5$				$L/D = 1$			
Bearing type	P	HM	QS	QI	P	HM	QS	QI
Two-groove cylindrical	1.13	0.36	0.78	1.81	0.72	0.25	0.50	1.50
Three-lobe	1.80	0.32	0.79	1.85	1.05	0.22	0.22	1.65
Canted three-lobe	1.52	0.31	1.42	3.30	0.91	0.22	0.90	2.58
Tilting-pad	1.50	0.30	0.44	2.82	0.90	0.20	0.16	1.50

QI = nondimensional inlet flow = $2qi/0.26\ wRLc$
QS = nondimensional side leakage flow = $2q_s/0.26\ wRLc$

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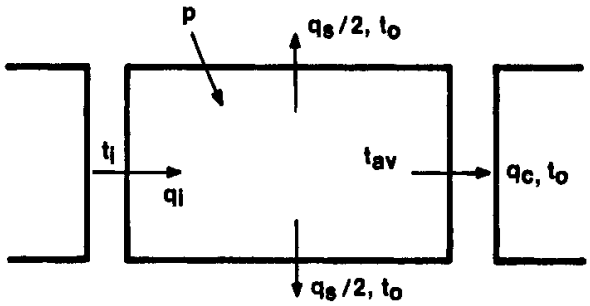


FIGURE 18 Heat balance in a fluid film

Figure 18 shows a developed view of the bearing surface and the parameters involved with conducting the heat balance. The lubricant enters a pad or bearing at the leading edge with an inlet flow q_i and an inlet temperature t_i. As the lubricant enters the bearing surface and flows through it, the lubricant is exposed to a viscous shear, which adds heat to the fluid. In Figure 18, the viscous shear is indicated as a power input p. Some of the fluid flows out of the sides of the bearing, represented by $q_s/2$, and some of the fluid, q_c, is carried either over to the grooving of the next pad or back to the inlet groove for a single pad bearing.

Although the fluid temperature, and thus the viscosity, changes along the length of the pad, it is assumed that the temperature in the film increases to some value t_o for both the side leakage and carry-over fluids. Then the heat balance is conducted as follows:

Heat added to a fluid by a viscous shear equals heat absorbed by the fluid.

$$\begin{gathered}\text{Heat added to fluid by viscous shear} = \text{heat absorbed by fluid} \\ pJ = q_s\rho C_p(t_o - t_i) = q_c\rho C_p(t_o - t_i) = (q_s + q_c)\rho C_p(t_o - t_i)\end{gathered} \tag{32}$$

Since by continuity of flow,

$$q_i = q_s + q_c \tag{33}$$

$$pJ = q_i\rho C_p\Delta t \tag{34}$$

$$\Delta t = \frac{pJ}{q_i\rho C_p} \tag{35}$$

where p = viscous heat generation, hp
J = heat equivalent constant = 0.7069 Btu/hp · s (J/kg · s)
ρ = specific weight of flow, lb/in^3 (kg/m^3)
C_p = specific heat of fluid, Btu/lb · °F (J/kg · °C)
q_i = bearing inlet flow, in^3/s (mm^3/s)
Δt = temperature rise, °F (°C)

If q_i is in gallons per minute [(1 in^3/s = 0.26 gpm) = 16.39 cm^3/s],

$$\Delta t = \frac{0.7069(0.26)p}{q_i\rho C_p} = \frac{0.1838p}{q_i\rho C_p} \tag{36}$$

Before proceeding to a sample problem, a word about the flows q_i and q_s, and the general heat balance philosophy. The flow required by the bearing to prevent starvation is q_i. The minimum make-up flow, or the flow that is lost from the ends of the bearing, is the side leakage q_s. Theoretically then, the only flow that need be supplied to the bearing is q_s. However, if this were true, then the heat balance formulation would require another balance between the carry-over flow q_c at some temperature t_o and the make-up flow q_s coming into the pad at temperature t_i. It would be found that, if we supplied only q_s to the bearing, excessive temperatures would result. In most instances, the amount of flow supplied to a bearing exceeds both the inlet flow and the side leakage flow by a significant

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margin. Designers often determine the flow to be supplied to a bearing system by a bulk temperature rise of the total flow entering the inlet pipe and exiting the exhaust pipe. Thus, the entire bearing is treated as a black box, and the total flow q_T to the bearing system for a given bulk temperature rise Δt_t is

$$q_T = \frac{pJ}{\rho C_p \Delta t_t} \tag{37}$$

Normally, Δt_t is selected between 20 and 40°F. Since q_T will exceed q_i, not all the flow will enter the film. Some will surround the bearing or flow through the feed groove and act as a cooling medium for heat transfer through the bearing walls or for cooling the fluid that does enter the bearing surface q_i.

EXAMPLE This example demonstrates the use of the design curves and how to perform a simplified heat balance in a bearing analysis.
In this example, we know these values:

Bearing type: two-groove cylindrical

Bearing length L = 3 in (76 mm)

Bearing diameter D = 6 in (152 mm)

L/D = 0.5

N = 1800 rpm; $\omega = 1800 \times \pi/30$ = 188.5 rad/s

Bearing load ω = 6000 lb (2721 kg)

Inlet temperature t_i = 120°F (49°C)

Lubricant = SAE 20

Bearing radial clearance c = 0.003 in (0.076 mm)

NOTE: A general rule of thumb is that $c/R = 0.001$.
Assume an average lubricant temperature of 130°F − μ (SAE 20 at 130°F) = 4.5×10^{-6} reyn (lb · s/in^2) (31×10^{-3} Pa · s)

$$\rho = 0.0307 \text{ lb/in}^3 (849.7 \text{ kg/m}^3) \text{ (average value for most oils)}$$

$$C_p = 0.5 \text{Btu/lb} \cdot \text{F}^\circ \text{ (2093 J/kg} \cdot \text{C}^\circ\text{) (average value for most oils)}$$

Compute the following:

$$W = \frac{wc^2}{6\mu\omega RL^3} = \frac{(6000)(0.003^2)}{(6)(4.5 \times 10^{-6})(188.5)(3)(3^3)} = 0.1309$$

From Figure 13 and Eq. 29,

$$P = 1.0 = \frac{1100cp}{\mu(\omega RL)^2}$$

$$P = \frac{(1)(4.5 \times 10^{-6})(188.5 \times 3 \times 3)^2}{(1100)(0.003)} = 3.92 \text{ hp (2.95 kW)}$$

From Figure 13 and Eq. 30,

$$Q_I = 1.85 = \frac{2q_i}{0.26\omega RLC}$$

$$q_i = \frac{(1.85)(0.26)(188.5)(3)(3)(0.003)}{2} = 1.224 \text{ gpm (4.63 l/min)}$$

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From Eq. 36,

$$\Delta t = \frac{0.1838p}{q_i \rho C_p} = \frac{(0.1838)(3.92)}{(1.224)(0.0307)(0.5)} = 38.35\ \text{F°}\ (21.3\ \text{C°})$$

$$t_{av} = \frac{t_i + (t_i + \Delta t)}{2} = \frac{2t_i + \Delta t}{2} = \frac{(2)(120) + 38.85}{2} = 139.2°\text{F}\ (59.6°\text{C})$$

The assumed t_{av} of 130°F was apparently not high enough and therefore we must repeat the calculations with a higher value of assumed t_{av}. Assume $t_{av} = 138°\text{F}$, $\mu = 3.8 \times 10^{-6}$ lb · s/in^2 (26.2×10^{-3} Pa · s) and repeat the calculations:

$$W = \frac{5.8945 \times 10^{-7}}{\mu} = 0.1552$$

$$P = 1.08 = (1.1466 \times 10^{-6})\left(\frac{p}{\mu}\right)$$

$$p = \frac{(1.08)(3.8 \times 10^{-6})}{1.1466 \times 10^{-6}} = 3.579\ \text{hp}\ (2.67\ \text{kW})$$

$$QI = 1.83 = \frac{q_i}{0.6616}$$

$$q_i = 1.211\ \text{gpm}\ (4.58\ \text{l/min})$$

$$\Delta t = 11.974\frac{p}{q_i} = \frac{(11.974)(3.579)}{1.211} = 35.38\ \text{F°}\ (19.65\ \text{C°})$$

$$t_{av} = \frac{(2)(120) + 35.38}{2} = 137.7°\text{F}\ (58.7°\text{C})$$

For all practical purposes, these equal the assumed value of $t_{av} = 138°\text{F}$ (58.9°C).

In conducting the iterative procedure for determining the average fluid temperature, it is sometimes helpful to plot points on a viscosity chart. Referring to Figure 19, suppose

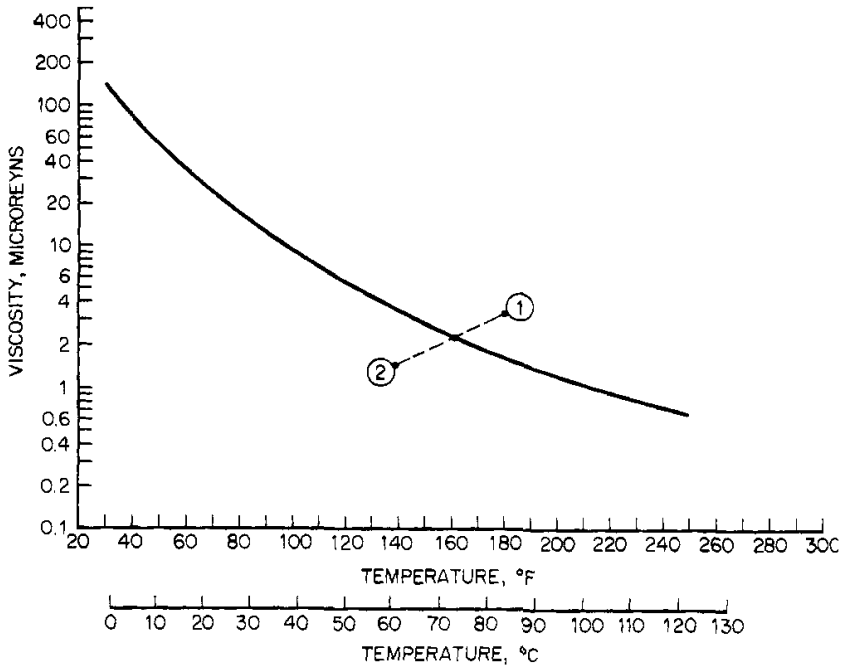


FIGURE 19 Graphical determination of average viscosity for SAE 20 oil

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point 1 is the resultant average viscosity and temperature of the initial guess and point 2 represents the result of the second guess. Then, by drawing a straight line between points 1 and 2 and establishing where it intersects the lubricant viscosity curve, we can determine the convergence to the proper result of average viscosity and temperature.

Now that we have convergence, we can determine the remaining variables, which are the minimum film thickness H_M and side leakage Q_S from Figure 13:

$$HM = 0.41 = \frac{h_M}{c}$$

$$h_M = (0.41)(0.003) = 0.00123 \text{ in } (0.031 \text{ mm})$$

$$QS = 0.74 = \frac{2q_s}{0.26\omega RLC} = \frac{q_s}{0.6616}$$

$$q_s = (0.6616)(0.74) = 0.490 \text{ gpm } (1.85 \text{ l/min})$$

A summary of the total bearing performance is as follows:

Load ω = 6000 lb (2721 kg)
Minimum film thickness h_M = 0.00123 in (0.031 mm)
Viscous power loss ρ = 3.579 (2.669 kW)
Inlet flow q_i = 1.211 gpm (4.58 l/min)
Side leakage q_s = 0.490 gpm (1.85 l/mm)
Fluid temperature rise Δt = 35.38°F (19.65°C)

We can repeat the same calculation for the other types of bearings, using Figures 14 through 17. The results are shown in Table 2.

The two-groove cylindrical bearing operates with the highest film thickness. The symmetric three-lobe has the lowest film thickness and highest temperature rise. It appears that, on the basis of steady-state performance, manufacture, and cost, the two-groove cylindrical bearing is the best choice.

BEARING DYNAMICS[7]

Fluid film bearings can significantly influence the dynamics of rotating shafts. They are a primary source of damping and thus can inhibit vibrations. Alternatively, they provide a

TABLE 2 Comparative Bearing Performance

Bearing type	Δt, °F (°C)	t_{av}, °F (°C)	p, hp	q_i, gpm (cm³/s)	q_s, gpm (cm³/s)	h_M, mils (mm)
Two-groove cylindrical	35 (19.44)	138 (58.9)	3.58	1.21 (76.7)	0.490 (31.05)	1.23 (0.0312)
Symmetric three-lobe	49 (27.22)	144 (62.2)	4.97	1.22 (77.3)	0.516 (32.7)	0.96 (0.0244)
Canted three-lobe	31 (17.22)	135 (57.2)	5.20	2.01 (127.4)	0.940 (59.6)	1.05 (0.0267)
Tilting-pad	26 (14.44)	134 (56.7)	4.88	2.26 (143.2)	0.311 (19.71)	1.02 (0.0259)

The following values were used: ω = 6000 lb (2721 kg), SAE 20 oil, L = 3 in (76 mm), D = 3 in (76 mm), N = 1800 rpm, t_i = 120°F (48.9°C), c = 0.008 in (0.076 mm).

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mechanism for self-excited rotor whirl. Whirl is manifest as an orbiting of the journal at a subsynchronous frequency, usually close to one-half the rotating speed. Whirl is usually destructive and must be avoided.

It is the purpose of this section to provide some insight into bearing dynamics, present some background on analytical methods and representations, and discuss some particular bearings and factors that can influence dynamic characteristics. Dynamic performance data and sample problems are presented for several bearing types.

The Concept of Cross Coupling As mentioned in the opening paragraphs of this section, a journal bearing derives load capacity from viscous pumping of the lubricant through a small clearance region. To generate pressure, the resistance to pumping must increase in the direction of the fluid flow. This is accomplished by a movement of the journal such that the clearance distribution takes on the form of a tapered wedge in the direction of rotation, as shown in Figure 1.

The attitude angle γ in Figure 1 is the angle between the load direction and the line of centers. Thus, the displacement of the journal is not along a line that is coincident with the load vector, and a load in one direction causes not only displacements in that direction, but orthogonal displacements as well.

Similarly, a displacement of the journal in the bearing will cause a load opposing the displacement and a load orthogonal to it. Thus, strong cross-coupling influences are introduced by the mechanism by which a bearing operates. The concept of cross-coupling is significant in dynamic characteristics.

It is the cross-coupling characteristics of a journal bearing that can promote self-excited instabilities in the form of bearing whirl. Motion in one direction produces orthogonal forces that in turn cause orthogonal motion. The process continues, and an orbital motion of the journal results. This orbital motion is generally in the same direction as shaft rotation and subsynchronous in frequency. Half-frequency whirl is a self-excited phenomenon and does not require external forces to promote it.

Cross-Coupled Spring and Damping Coefficients For dynamic considerations, a convenient representation of bearing characteristics is a cross-coupled spring and damping coefficients. These are obtained as follows (refer to Figure 1):

1. The equilibrium position to support the given load is established by computer solution of Reynolds' equation.
2. A small displacement to the journal is applied in the y direction. A new solution of Reynolds' equation is obtained, and the resulting forces in the x and y directions are produced. The spring coefficients are as follows:

$$K_{xy} = \frac{\Delta F_x}{\Delta y} \tag{38}$$

$$K_{yy} = \frac{\Delta F_y}{\Delta y} \tag{39}$$

where K_{xy} = the stiffness in the x direction due to y displacement
ΔF_x = the difference in x forces between displaced and equilibrium positions
Δy = the displacement from the equilibrium position in the y direction
K_{yy} = the stiffness in the y direction due to y displacement
ΔF_y = the difference in y forces between displaced and equilibrium positions

3. The journal is returned to its equilibrium position and an x displacement is applied. Similar reasoning produces K_{xx} and K_{yx}.

The cross-coupled damping coefficients are produced in a similiar manner, except, instead of displacements in the x and y direction, velocities in these directions are consecutively applied with the journal in the equilibrium position. The mechanism for increasing the load capacity is squeeze film in which the last term on the right-hand side of

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Equation 14 is actuated. Thus, for most fixed bearing configurations, eight coefficients exist: four spring and four damping. The total force on the journal is

$$F_i = K_{ij}x_j + D_{ij}\dot{x}_j \tag{40}$$

where F_i = force in the ith direction.

Repeated subscripts imply the following summation:

$$K_{ij}x_j = K_{ix}x + K_{iy}y + \ \ldots$$

It should be realized that the cross-coupled spring and damping coefficients represent a linearization of bearing characteristics. When they are used, the equilibrium position should be accurately determined, as the coefficients are valid for only a small displacement region encompassing the equilibrium position of the journal. This is true because the spring and damping coefficients remain constant for only a small region of the equilibrium position.

Consider the two-groove cylindrical bearing shown in Figure 1, with the geometric and operating conditions indicated in Table 3. The computer solution (also the performance curves in Figure 13) produces the following results:

Bearing load w = 20,780 lb (9,424 kg)

Power loss = 15.51 hp (11.56 kW)

Minimum film thickness h_M = 0.00125 in (0.032 mm)

Side leakage q_s = 0.941 gpm (3.56 l/mm)

The spring and damping coefficients are

Spring coefficients, lb/in (kg/mm):

$$\begin{bmatrix} K_{xx} & K_{xy} \\ K_{yx} & K_{yy} \end{bmatrix} = \begin{bmatrix} -12.14 \times 10^6 & -4.64 \times 10^6 \\ 28.3 \times 10^6 & -20.41 \times 10^6 \end{bmatrix}$$

Damping coefficients, lb · s/in kg · s/in:

$$\begin{bmatrix} D_{xx} & D_{xy} \\ D_{yx} & D_{yy} \end{bmatrix} = \begin{bmatrix} -2.85 \times 10^4 & 2.66 \times 10^4 \\ -2.69 \times 10^4 & -1.11 \times 10^5 \end{bmatrix}$$

The negative signs imply a positive stiffness because the restoring load is opposite the applied load. Note that for this bearing configuration there is very strong cross-coupling, evidenced by the magnitude of the off-diagonal terms.

Critical Mass The cross-coupled spring and damping coefficients provide a convenient way of representing a bearing in a stability analysis. They reduce the fluid film bearing to a spring-mass system (see Figure 20), and consequently stability and dynamics problems are simplified considerably.

Consider a journal of mass M operating in a bearing. The journal can be considered to have two degrees of freedom, x and y. The governing equations are

TABLE 3 Two-Groove Cylindrical Bearing Geometry and Operating Conditions

Journal diameter D	=	5 in (127 mm)
Bearing length L	=	5 in (127 mm)
Active pad angle θ_p,	=	160° (10° grooves on either side)
Radial clearance c	=	0.0025 in (0.064 mm)
Operating speed N	=	5000 rpm
Lubricant viscosity μ	=	2×10^{-6} lb · s/in^2 (13.79×10^{-3} Pa · s)
Eccentricity ratio ε	=	0.5
Load direction is vertical downward		

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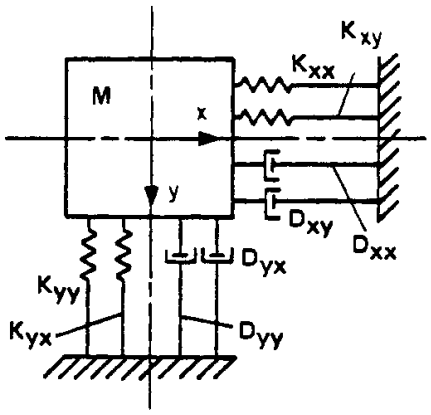


FIGURE 20 Point mass representation of a bearing supported on cross-coupled springs and dampers

$$M\ddot{x} + D_{xx}\dot{x} + D_{xy}\dot{y} + K_{xx}x + K_{xy}y = 0 \tag{41}$$

$$M\ddot{y} + D_{yy}\dot{y} + D_{yx}\dot{x} + K_{yy}y + K_{yx}x = 0 \tag{42}$$

Assume a sinusoidal response to the form

$$x = x_0 e^{\beta t} \tag{43}$$

$$y = y_0 e^{\beta t} \tag{44}$$

where β is a complex variable:

$$\beta = \alpha + i\omega \tag{45}$$

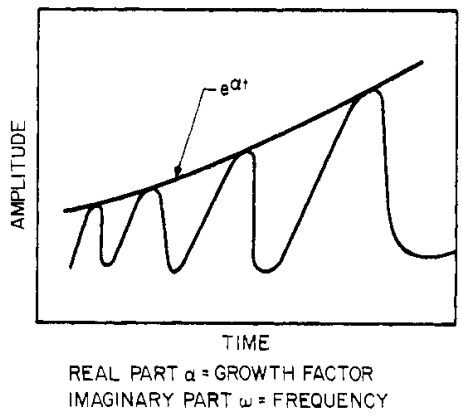
By the Euler expansion of $e^{\beta t}$, another way to write Eqs. 43 and 44 is

$$x = x_o e^{\alpha t}(\cos \omega t + i \sin \omega t) \tag{46}$$

$$y = y_o e^{\alpha t}(\cos \omega t + i \sin \omega t) \tag{47}$$

An interpretation of α and ω is shown in Figure 21. The real part of $\beta = \alpha$ is called the *growth* or *attenuation factor*. The imaginary part is the frequency of vibration. A positive real part means that the response to a disturbance grows in time. The growth factor is similar to the logarithmic mean decrement, which is common in vibration theory:

$$\alpha\tau = \ln \frac{x_{n+1}}{x_n} \tag{48}$$



Interpretation of $\chi = \chi_o e^{\beta t}$
where
$\beta = \alpha + i\omega$
$\chi = \chi_o e^{\alpha t} e^{i\omega t} = \chi_o e^{\alpha t}(\text{COS } \omega t + i \sin \omega t)$

FIGURE 21 Interpretation of the growth factor α and orbital frequency ω

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where τ = period of vibration
x_{n+1} = amplitude at time $n + 1$
x_n = amplitude at time n

Thus, the growth factor α is a measure of the growth or decay of the journal to a small disturbance. A positive growth factor implies an instability.

The solutions to Eqs. 41 and 42 are obtained by substituting Eqs. 43 and 44, which produces the following:

$$\begin{bmatrix} (M\beta^2 + D_{xx}\beta + K_{xx}) & (D_{xy}\beta + K_{xy}) \\ (\beta D_{yx} + K_{yx}) & (M\beta^2 + \beta D_{yy} + K_{yy}) \end{bmatrix} \begin{Bmatrix} x_0 \\ y_0 \end{Bmatrix} = \{0\}$$

To obtain a solution, the determinant of the coefficient matrix must vanish. Expansion produces a polynomial in β that can be solved for the roots of β, which in turn provide the growth factors and frequencies of vibration.

It is possible to obtain a closed-form solution of Eqs. 41 and 42 for the critical mass and resulting orbital frequency. The critical mass M is defined as that mass above which an instability will occur. At the threshold of instability, the real part of $\beta = \alpha$ goes to zero and $\beta = i\omega$.

Substituting into Eq. 49, expanding the determinant and separating real and imaginary components produces the following equations:

$$M^2\omega^4 - \underbrace{M(K_{yy} + K_{xx})}_{A}\omega^2 + \underbrace{(D_{yx}D_{xy} - D_{xx}D_{yy})}_{B}\omega^2 + \underbrace{K_{xx}K_{yy} - K_{xy}K_{yx}}_{C} = 0 \tag{50}$$

$$\underbrace{M(D_{yy} + D_{xx})}_{D}\omega^2 + \underbrace{(D_{xy}K_{yx} + D_{yx}K_{xy} - D_{xx}K_{yy} - D_{yy}K_{xx})}_{E} = 0 \tag{51}$$

The two equations can be solved for M and ω:

$$M = \frac{BED}{E^2 - AED + CD^2} \tag{52}$$

$$\omega = \sqrt{\frac{-(AED + E^2 + CD^2)}{BD^2}} \tag{53}$$

Thus, if the cross-coupled coefficients are known, it is possible to determine the critical mass and the orbital frequency. If the mass acting on the bearing exceeds or equals the critical value, then an instability will occur.

Dynamic Stability of Various Bearing Types Several parameters can be used to establish the stability characteristics of a particular type of bearing. The most significant is the critical mass, derived above. It is also possible to get some feel for stability from purely steady-state performances by examining the bearing attitude angle. The larger the attitude angle, the greater the cross-coupling influence and the worse the stability characteristics. Interpreting attitude angles, however, can prove misleading.

Figure 22 shows plots of the attitude angle γ versus the load parameter W for some of the different types of bearings previously described. The contradiction in these results is that the symmetric three-lobe bearing has higher attitude angles than the two-groove cylindrical bearing even though, as will be subsequently demonstrated, it has superior stability characteristics. The reason for the higher attitude angle is that the bearing is cavitating in the diverging region of the loaded pad, which results in a large shift in the journal. Whirl motion, however, is prevented by the accompanying lobes.

A more direct and accurate approach to establishing bearing stability is to determine the critical mass acting on the bearing. Figure 23 shows the results obtained for the fixed

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CENTRIFUGAL PUMPS

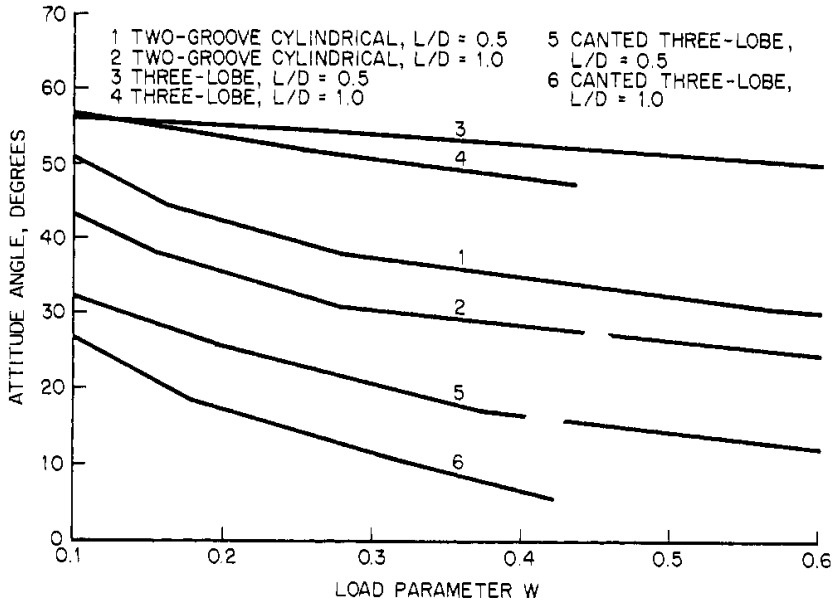


FIGURE 22 Attitude angle versus load parameter for various bearing types

bearing geometries previously discussed. For any particular bearing geometry, if the critical mass attributable to the bearing exceeds that of the data plotted, the bearing will be unstable. Thus, if the plotted point falls to the right of the bearing line, the bearing is stable; if it falls to the left, the bearing is unstable.

The critical mass is defined as follows:

$$M = m\omega Rc^3/24\mu L^5 \tag{54}$$

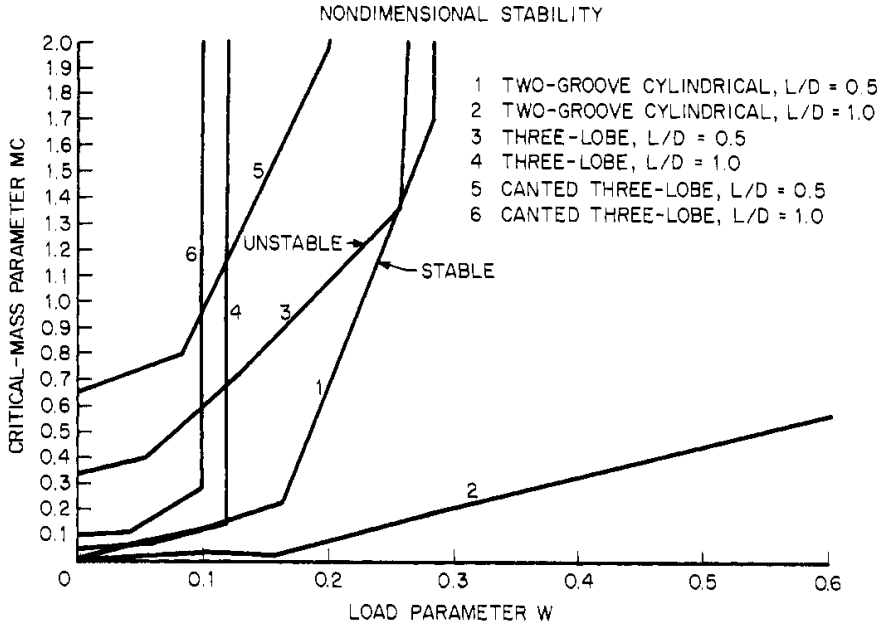


FIGURE 23 Critical mass versus load parameter for various bearing types

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where M = nondimensional critical mass
m = dimensional critical mass, lb · s²/in (kg · s²/mm)
ω = shaft speed, rad/s
R = shaft radius, in (mm)
c = bearing machined radial clearance, in (mm)
μ = lubricant viscosity, lb sec/in² (Pa · s)
L = bearing length, in (mm)

An examination of the curves in Figure 23 clearly indicates the superiority of the canted three-lobe bearing and the inferiority of the cylindrical configuration.

EXAMPLE Consider a high-speed bearing for which

N = 60,000 rpm

ω = 60,000π/30=6283 rad/s

D = 1 in (25.4 mm)

L = 0.5 in (12.7 mm)

μ = 1×10^{-6} lb · s/in⁵ (6.9×10^{-3} Pa · s)

C = 0.0005 in (0.013 mm)

w = 100 lb (45.35 kg)

$$W = \frac{wC^2}{6\mu\omega RL^3} = \frac{(100)(0.0005^2)}{(6)(1\times10^{-6})(6283)(0.5)(0.5^3)} = 0.0106$$

The value of M is obtained from Figure 22 and indicated in Table 4 for the various bearing types considered. Dimensional units are also given. Table 4 clearly indicates that the lobe bearings can permit significantly more attributable mass than the cylindrical bearing can. If a symmetric rotor is being supported by two bearings, the cylindrical bearings would be unstable if half the weight of the rotor exceeded 14.7 lb (6.67 kg). The half weights go to 258 lb (117 kg) and 501 lb (227.2 kg) for the symmetric three-lobe and canted three-lobe bearings, respectively. No mention has been made of the tilting-pad bearing because, for all practical purposes, these bearings are always stable.

OPERATING CONDITIONS THAT AFFECT BEARING STABILITY

Cavitation Figure 1 shows a pressure distribution in a journal bearing. Positive pressure is generated in the converging wedge because the journal is pumping fluid through a restriction. On the downstream side of the minimum film thickness, the journal is pumping fluid out of a diverging region. In this region, the pressure decreases. Either the pressure becomes negative, which is defined as pressure below the ambient pressure,

TABLE 4 Stability Comparison Chart

Bearing type	M	m, lb · s²/in (kg · s²/mm)	w', lb (kg)[a]
Two-groove cylindrical	0.02	0.038 (6.79×10^{-4})	14.7 (6.7)
Symmetric three-lobe	0.85	0.668 (11.9×10^{-3})	257.9 (117.1)
Canted three-lobe	0.68	1.299 ($23.2 > 10^{-3}$)	501.4 (227.6)
	$m = \frac{24M\mu h^5}{\omega Rc^3} = \frac{M(24)(10^{-6})(0.5^5)}{(6283)(0.5)(0.0005^3)} = 1.91M$		

[a]The w' represents the maximum mass in weight units that could act on the bearing.

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or the film cavitates and decreases to atmospheric pressure as the lubricant releases entrapped air.

With respect to stability, cavitation is a more desirable condition than the development of negative pressure. From an examination of Figure 1, it can be seen that the negative pressure pulls the journal in an orthogonal direction and increases the cross-coupling. The more eccentric the bearing, the larger the negative pressure or cavitated region.

In lightly loaded bearings that are pressure-fed, negative pressures can occur because pressures in the divergent region have not approached atmospheric pressure. Cavitation does not occur, and the bearing is prone to instability. Thus, the feed pressure and load on a bearing are two additional parameters that affect stability.

Lobe bearings are often used because of their excellent antiwhirl characteristics. Figure 9 earlier showed two types of lobe bearings: symmetric and canted. The symmetric lobe bearing is designed so that, in the concentric position, the minimum film thickness occurs at the center of each lobe. Note that this permits a region of converging film followed by a region of diverging film. Thus, depending upon the ambient pressure, it is possible to have negative pressures in a symmetric lobe bearing. Under very high ambient conditions, a symmetric lobe bearing can go unstable.

The canted lobe bearing is designed to have a completely converging wedge and positive pressure throughout its arc length. Its stability characteristics are superior to those of the symmetric lobe bearing. Its steady-state characteristics are also superior.

Hybrid Bearings At times, externally pressurized bearings are resorted to for stability improvements. The philosophy is that externally pressurized bearings are not subject to high attitude angles, as is the case with hydrodynamic journal bearings. Although this is generally true, hybrid hearings can still be subject to considerable cross-coupling. Figure 11 showed a schematic arrangement of a hybrid bearing. Oil is fed through restrictors from an external source into pocket recesses. From there, it exits into the clearance region between recesses. Lubricant is also pumped into and out of recesses by the rotating shaft by a viscous drag in the same manner as with a purely hydrodynamic bearing.

Consider recess 2 in Figure 11. Oil is pumped from the shaft via a converging wedge, and it augments the pressure in the recess provided by the external system. The net result is a higher pressure in the bearing domain covered by recess 2 than would occur without rotation. Now consider recess 3. Here the journal is pumping fluid out of the recess into a diverging film, so that the hydrodynamic action tends to reduce the pressure in this recess domain. By similar reasoning, it can be shown that recess 4 operates at a lower pressure than recess 1. The net result of these variations in pressure due to rotation is that cross-coupling forces are introduced and the hybrid bearing may not prevent instability.

BEARING MATERIALS AND FAILURE MODES

Materials The most common material used for oil-lubricated fluid film bearings is babbitt. Tin- and lead-based babbitts are relatively soft materials and offer the best insurance against shaft damage. They also enable embedded dirt and contaminants without significant damage.

Two types of babbitts are in common use. One has a tin base (86 to 88%), with about 3 to 8% copper and 4 to 14% antimony. The other has a lead base with a maximum of 20% tin and about 10 to 15% antimony. The remainder is principally lead. The physical properties of babbitt are shown in Table 5. The primary limitations of babbitt are operating temperature (300°F [140°C] max) and fatigue strength. The chemical composition of various babbitt alloys are indicated in Table 6.

Tin-based babbitts have better characteristics than lead-based babbitts; they have better corrosion resistance, are less likely to wipe under poor conditions of lubrication, and can be bonded more easily than lead-based materials. Because of cost considerations, however, lead-based babbitts are widely used. The more widely used is the SAE 15 alloy containing 1% arsenic (refer to Table 6).

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TABLE 5 Properties of Bearing Alloys

Bearing material	Brinell hardness at room temperature	Brinell hardness at 30°F (17°F)	Minimum Brinell hardness of shaft	Load carrying capacity, lb/in^2 (kg/m^2)	Max operating temperature, °F (°C)
Tin-based babbitt	20–30	6–12	150 or less	800–1500 (5.62×10^5 – 10.55×10^5)	300 (149)
Lead-based babbitt	15–20	6–12	150 or less	800–1200 (5.62×10^5 – 8.44×10^5)	300 (149)

SOURCE: Ref. 8

Tin-based babbitts do not experience as many corrosion problems as lead-based babbitts. SAE 11 babbitts (containing 8% antimony and 8% copper) are used extensively for industrial applications.

Babbitts are bonded to a backing shell of another material, such as steel or bronze, because they are not a good structural material. The thinner the babbitt layer, the greater the fatigue resistance. In automotive applications where resistance to fatigue is important, babbitt thickness is from 0.001 to 0.005 in (0.02 to 0.12 mm). For pump applications, the thickness varies from $\frac{1}{32}$ to $\frac{1}{8}$ in (0.8 to 3 mm). The thicker layers provide good conformity and embedability.

Failure Modes The failure modes most commonly found are fatigue, wiping, overheating, corrosion, and wear.

Fatigue occurs because of cyclic loads normal to the bearing surface. Figure 24 shows babbitt fatigue in a 7-in (178-mm) diameter journal bearing from a steam turbine.

Wiping results from surface-to-surface contact and smears the babbitt, as shown in Figure 25. The usual causes of wiping are a bearing overload, insufficient rotational speed to form a film, and loss of lubricant.

TABLE 6 Composition[a] Percentages of Babbitts with SAE Classifications of 11 to 15

Element	SAE No. (Similar ASTM Spec.)				
	11 (None)	12 (B23, alloy 2)	13 (None)	14 (B23, alloy 7)	15 (B23, alloy 15)
Tin (min)	86.0	88.2	5.0–7.0	9.2–10.8	0.9–1.2
Antimony	6.0–7.5	7.0–8.0	9.0–11.0	14.0–16.0	14.0–15.5
Lead	0.5	0.5	Remainder	Remainder	Remainder
Copper	5.0–6.5	3.0–4.0	0.5	0.5	0.5
Iron	0.08	0.08	—	—	—
Arsenic	0.1	0.1	0.25	0.6	0.8–1.2
Bismuth	0.08	0.08	—	—	—
Zinc	0.005	0.005	0.005	0.005	0.005
Aluminum	0.005	0.005	0.005	0.005	0.005
Cadmium	—	—	0.05	0.05	0.05
Others	0.2	0.2	0.2	0.2	0.2

[a]The percentage of minor constituents represents limiting values except as noted.

SOURCE: *SAE Handbook*, Society of Automotive Engineers. New York, 1960, p. 201.

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FIGURE 24 Babbitt fatigue in a 7-in (178-mm) diameter turbine bearing (Westinghouse Electric Corp. *Photo originally reproduced in* Ref. 8, p.18–19.)

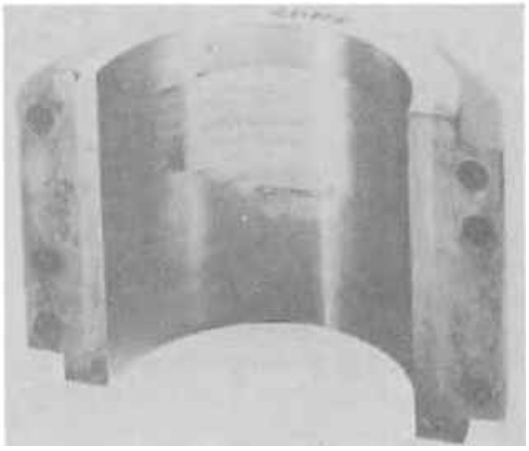


FIGURE 25 Bearing wipe on a 3-in (76-mm) diameter bearing due to temporary loss of lubricant (Westinghouse Electric Corp. *Photo originally reproduced in* Ref. 8, p.18–25)

Overheating is manifest by discoloration of the surface and cracking of the babbitt material. Corrosion is failure by a chemical action. It is more common with lead-based babbits, which react with acids in the lubricant. Figure 26 shows corrosion damage for a 5-in (127-mm) diameter, lead-based babbitt bearing.

Wear results from contaminants in the film and is evidenced by scoring marks that may be localized or persist around a large circumferential region of the bearing.

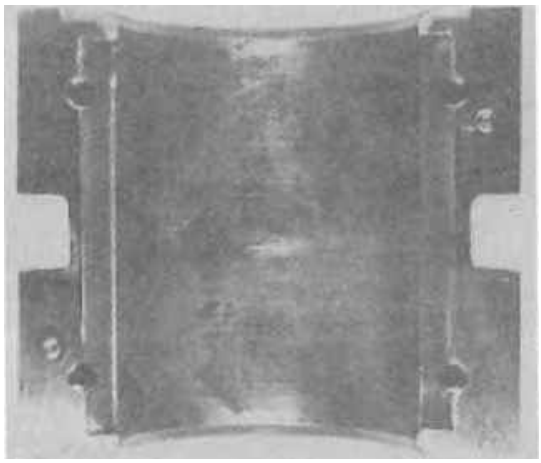


FIGURE 26 Corrosion on a 5-in (127-mm) diameter, lead-based babbitt bearing (Westinghouse Electric Corp. *Photo originally reproduced in* Ref. 8, p. 18–21)

REFERENCES

1. Fuller, D. D. *Theory and Practice of Lubrication for Engineers*. Wiley, New York, 1956.
2. Castelli, V. and W. Shapiro. "Improved Method of Numerical Solution of the General Incompressible Fluid-Film Lubrication Problem." *Trans. ASME, J. Lub. Technol.,* April, 1967, pp. 211–218.
3. Castelli, V.,and J. Pirvics. "Review of Methods in Gas-Bearing Film Analysis." *Trans. ASME, J. Lub. Technol.,* October 1968, pp. 777–792.
4. Pinkus, O. and B. Sternlicht. *Theory of Hydrodynamic Lubrication*. McGraw-Hill, New York, 1961.
5. Ng, C. W. and C. H. T. Pan. "A Linearized Turbulent Lubrication Theory." *Trans. ASME, J. Basic Eng.,* Series D, Vol. 87, 1965, p. 675.
6. Elrod, H. G. and C. W. Ng. "A Theory for Turbulent Films and Its Application to Bearings." *Trans. ASME, J. Lub. Technol.,* July 1967, p. 346.
7. Shapiro, W., and R. Colsher. "Dynamic Characteristics of Fluid Film Bearings." *Proc. Sixth Turbomachinery Symposium,* sponsored by the Gas Turbine Laboratories, Department of Mechanical Engineering, Texas A&M University, College Station, Texas, December 1977.
8. O'Connor, J. I., J Boyd, and E. A. Avallone. *Standard Handbook of Lubrication Engineering*. McGraw-Hill, New York, 1968.

SECTION 6.3
CENTRIFUGAL PUMP MAGNETIC BEARINGS

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LARRY HAWKINS
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Magnetic bearings maintain the rotor of a pump in suspension through the forces of attraction of a magnetic circuit. Thus, although they bear up the weight and hydraulic loads of the impellers and the shaft, they are not really bearings in the traditional sense of the rotating and stationary surfaces bearing on one another. The supporting magnet circuit for each bearing includes stationary magnets in a stator surrounding the shaft, a laminated rotor that fits on the shaft, and the shaft itself. The stator consists of electromagnets in the traditional heteropolar design, and if a homopolar design is employed, permanent magnets can be added. Sensors monitor the position of the shaft and signal a controller to adjust the magnetic loads to keep the shaft to within about 0.001 in (25 μm) of the desired position.

Magnetic bearings are found in small, high-speed turbomachinery such as high-speed, multistage, axial-flow turbomolecular vacuum pumps[1]. They were introduced into large turbomachinery in the early 1980s, mainly in gas compressors and turboexpanders. Their use and acceptance has grown slowly but steadily since then[2]. Pump applications of a significant size have appeared and have confirmed the general position that magnetic bearings can provide a technically sound bearing with maintenance and operating advantages, including zero wear. However, due to the technical complexity of magnetic bearing systems, the economies of scale associated with production quantities are required to make these systems affordable.

Two representative magnetic-bearing-equipped pumps are summarized in Table 1. One is a multistage boiler feedwater pump[3–6] and the other a single-stage double-suction hydrocarbon process pump[7]. The multistage pump was retrofitted with magnetic bearings (as shown in Figure 1) and is shown in Figure 2, together with another identical pump that still contains the oil-lubricated bearings—both installed in an electric generating station. The magnetic-bearing pump is not encumbered with the usual complexity of a bearing lubrication system.

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TABLE 1 Example of magnetic-bearing-equipped pumps

Parameter	Multistage Pump	Single-Stage Pump
Power, hp (MW)	610 (0.46)	800 (0.6)
Rated speed, rpm	3,580	1780
Shaft weight, lb (kg)	520 (236)	732 (332)
Radial bearing design load, lb (kN)	800 (3.6)	Thrust end: 930 (4.1) Drive end: 1,415 (6.3)
Thrust bearing design load, lb (kN)	4,000 (17.8)	4,000 (17.8)
Number of stages	8	1

MAGNETIC BEARING PRINCIPLES

How Magnetic Bearings Work In an active magnetic bearing system, a stator composed of an array of stationary magnets, or electromagnetic coils, interacts with a ferrous rotor (or a ferrous sleeve on a nonferrous rotor) so as to suspend the shaft in a magnetic field (see Figure 3).

The position of the shaft is maintained dynamically through a continuous feedback system which comprises a position sensor, a controller, and an amplifier system (see Figure 4). Typically there are two radial bearings and one thrust bearing for a complete system. This system is tuned to the required characteristics of the pump, through a digital or analog controller (Figure 5), with the capability of adjusting the bearing stiffness and damping as a function of pump speed. Alarms and trips can be set at any required rotor offset or bearing load to provide the operator with warnings or to trip the drive unit as necessary. Figure 6 illustrates a typical bearing transfer function, showing a statically stiff bearing, with a dynamic stiffness over the operating speed range designed to meet the rotor dynamics requirements of the unit. The stiffness is then rolled off above the operating range to avoid excitation of higher modes in the rotor or stator.

Controller redundancy can be provided with the control loops switching to backup units upon sensing a failure. Also, backup power supplies should be provided, either through alternate sources or a battery system. Typically the power required is only 1 or 2 kW or less.

A catcher (also known as auxiliary, backup, or touchdown) bearing (indicated in Figure 1) is required to protect the rotor stator interface during maintenance and in the event of loss of power or a severe transient beyond the force capability of the bearing. Typically, the catcher bearing is designed for 5 to 20 lifetime drops from full speed, and will be a readily replaceable rolling element or sleeve bearing. The pump of Figures 1 and 2 has rolling

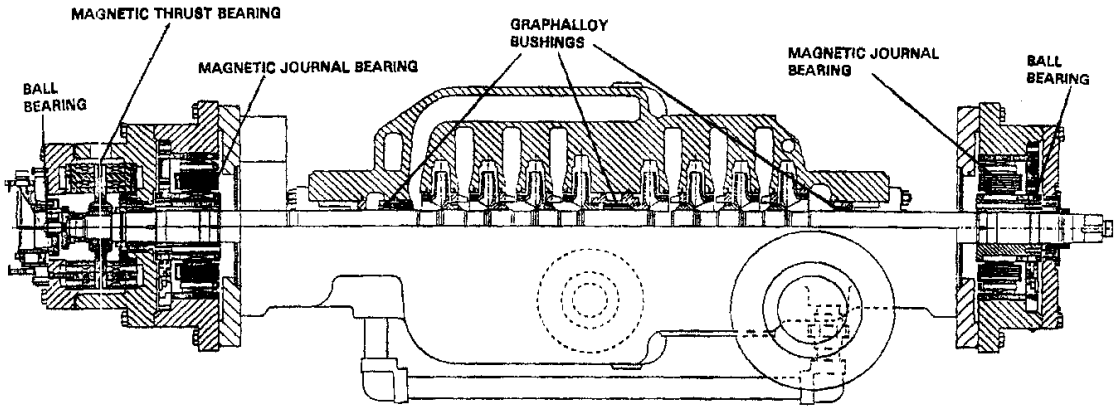


FIGURE 1 Magnetic bearing configuration in multistage pump[6]

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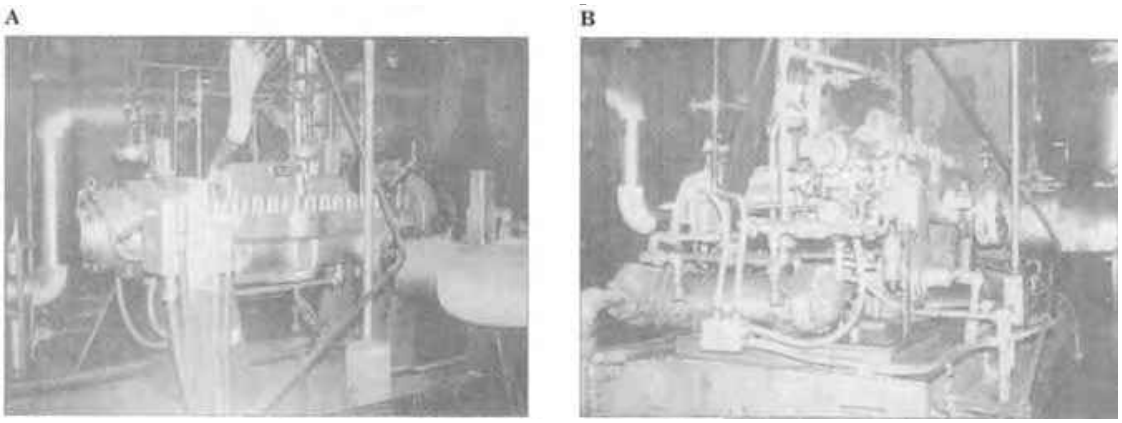


FIGURE 2 Multistage pumps installed in boiler feed service [610 hp (0.46 MW)]: a) pump with magnetic bearings; b) the same pump with oil-lubricated bearings. (Flowserve Corporation)

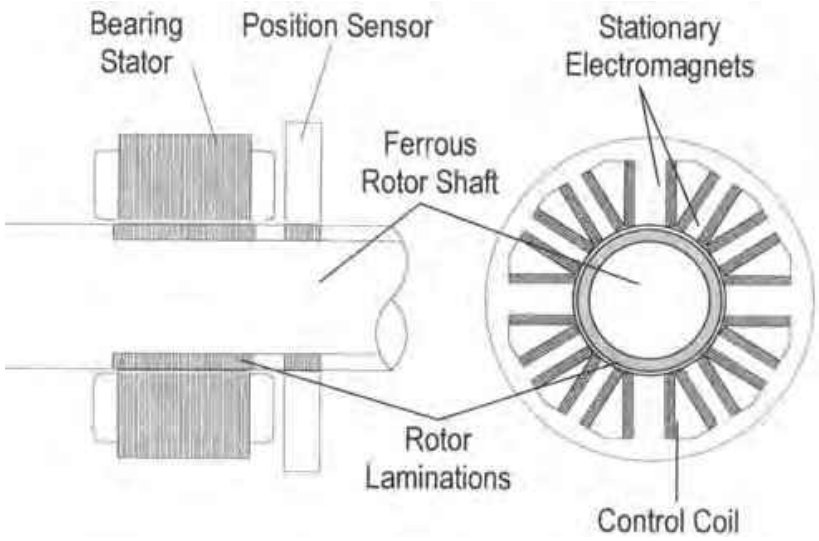


FIGURE 3 Typical radial electromagnetic bearing (Axial thrust bearings have the stator coils arranged in a disk configuration, a ferrous rotating disk being supported in the resulting magnetic circuit.)

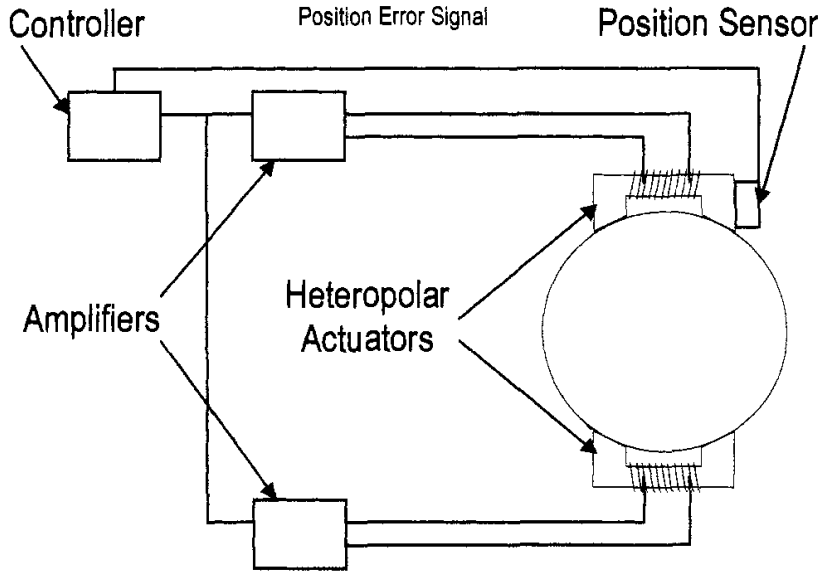


FIGURE 4 Typical system control loop

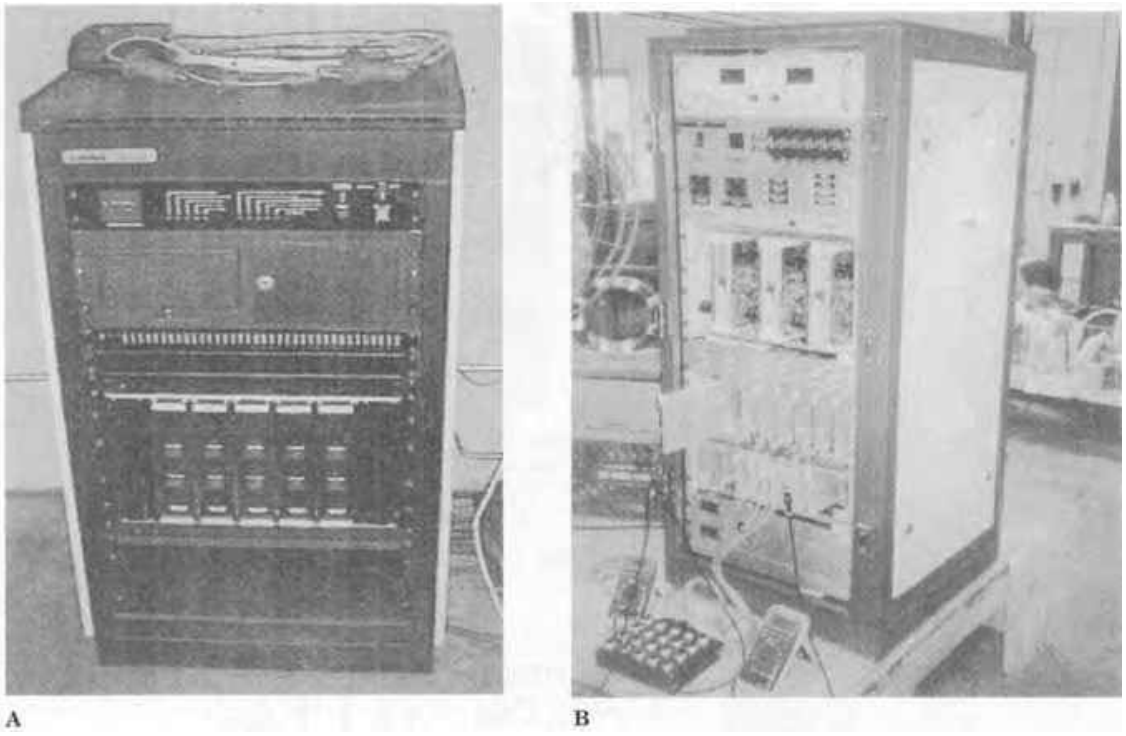


FIGURE 5 Controllers for magnetic bearings, containing rectifier and amplifiers: a) digital controller; b) analog controller

element catcher bearings. The radial clearances G1 (see Figure 7) between the magnetic bearing stator and rotor are of the order of 20 to 40 mil (0.5 to 1 mm), and those in the catcher bearings (G2) are about half of that value.

Reasons for Using Magnetic Bearings There are several reasons to use magnetic bearings in pumps. While any one of these reasons may not be sufficient justification on its own, together they can provide a strong justification.

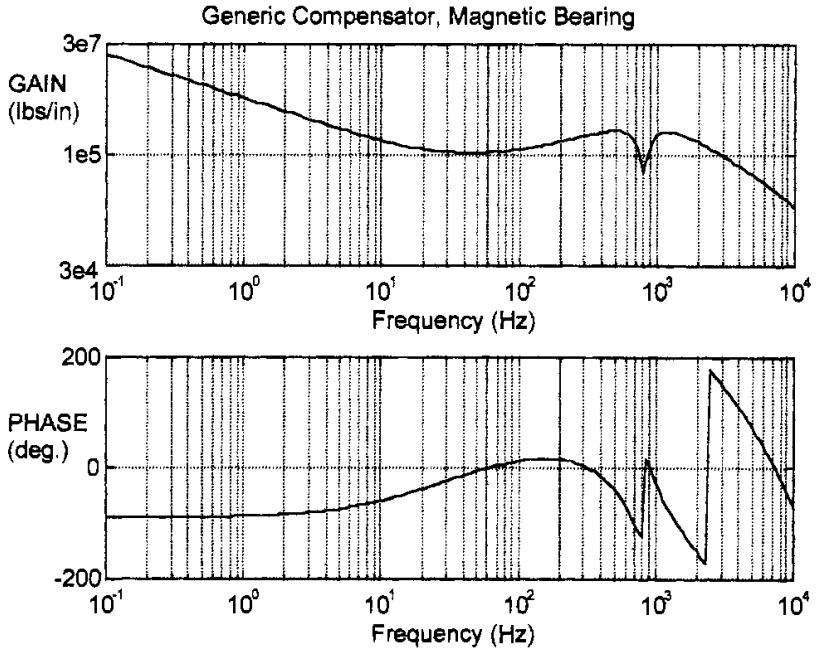


FIGURE 6 Bearing transfer function

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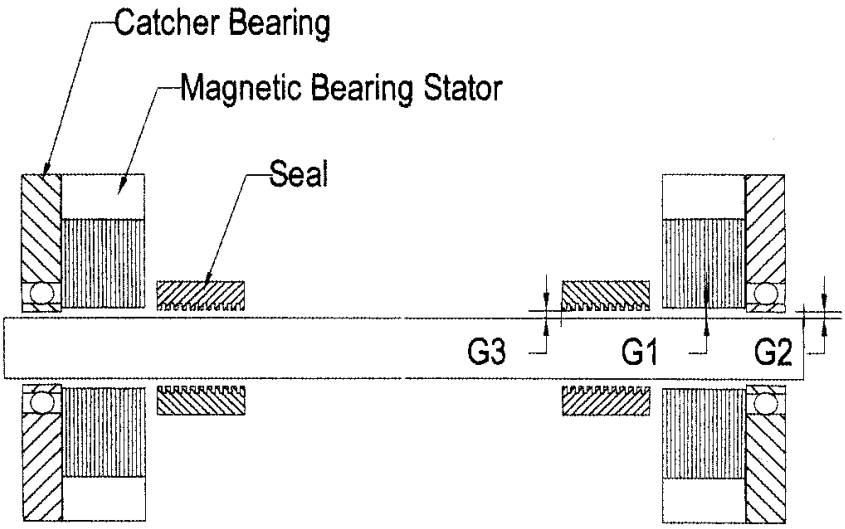


FIGURE 7 Clearance arrangement: Seal ring clearance G3 is greater than catcher-bearing clearance G2, but is less than magnetic-bearing clearance G1.

Reliability is a key incentive. The components of a magnetic bearing are essentially the same components as are found in an electric motor: laminations and coils. Because no wear is involved due to the lack of contact, these components will generally last the life of the equipment involved. Thus maintenance of a magnetic bearing system is transferred from mechanical components inside the pump to the external controller, which has plug-in card replacement maintenance. Pump reliability is therefore improved, whereas repair times and costs are reduced.

Reduced power consumption is a second advantage, with the elimination of all losses associated with fluid film bearings and oil pumping equipment. This is replaced by the smaller power requirements of the bearing controller. Further, if the lifting force is supplied by a permanent magnet, supplemented by an active control circuit, this power requirement can be even smaller.

The ability to submerge the bearing in the pump fluid is a major advantage that allows the outboard mechanical seal to be eliminated, thereby eliminating maintenance and replacement of this seal[8]. [This was not done for the pumps of Table 1 (and Figures 1 and 2), because in both cases magnetic bearings were retrofitted to existing machines.]

More indirect savings are also possible in two other areas. Rotor dynamics can be controlled through the ability to adjust stiffness and damping as a function of pump speed, allowing higher imbalance without the need for shutdown. The diagnostic output inherent in the information provided in the controller can be fed into the overall plant operating system and the short-term and long-term health of the pump and the system can be monitored. This is done by inferring seal wear, transient hydraulic loads, and so on.

The actual figures for the savings possible due to the previous advantages are very pump- and system-specific, and general numbers are not very useful. Reference 9 has developed methodology for considering the economic effect of the types of advantages given.

Main Types of Magnetic Bearings and Their Selection There are two main types of magnetic bearings: passive and active. Passive bearings rely only on permanent magnets in repulsion and provide low stiffness, low damping, and no ability to control either of these parameters. Passive bearings are not applicable to pumps for this reason.

Active bearings using the feedback system previously described are essential for pump applications. Within the active bearing systems are the options of heteropolar and homopolar and of electromagnetic and permanent magnet bias. The principles of the heteropolar and homopolar approaches are shown in the Figure 8.

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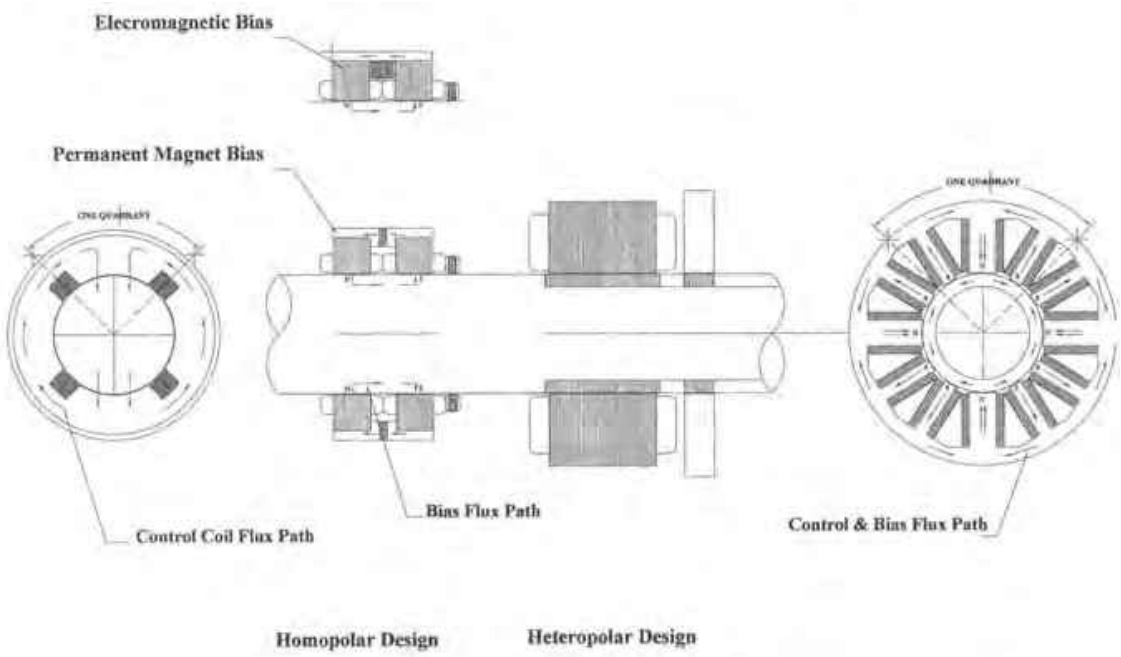


FIGURE 8 Heteropolar and homopolar bearings

The main difference between the two types is that in the heteropolar design, the bias and control flux flow in the same magnetic circuit radially through the rotor, whereas in the homopolar design the bias flux flows axially along the rotor and only the control flux flows radially through the rotor.

The homopolar design has two options for providing the bias flux for the bearing system[10]. One is to use an electromagnetic effect, and the other is to use a permanent magnet. The permanent magnet generation in the homopolar configuration results in a more linear relationship between force and distance. In a simple magnetic circuit, the attraction force of a magnet on a ferromagnetic target decreases as the square of distance (the target cuts $\frac{1}{4}$ as many flux lines at twice the distance). With the permanent magnet in a homopolar circuit, the effect of the air gap is therefore reduced.

DESIGN CONSIDERATIONS

Design Loads The specification of a magnetic bearing requires a different approach to that required for a conventional bearing system. This section identifies the key areas where these differences occur.

The rotor can move considerably within the magnetic bearing and catcher bearing clearances, typically up to 10 mil (0.25 mm), before any contact is involved. Thus the clearances G3 in the internal ring-seal system (see Figure 7) become the key controlling parameter in setting the bearing clearance design limits and in the degree of motion permitted during transients. Thus, very early in the design process, the magnetic bearing clearances, catcher bearing clearances, and sealing-ring clearances must be optimized with due consideration for manufacturing tolerances and assembly concerns.

Synchronous filters or open-loop control methods can be used to handle imbalance loads so that the degree of imbalance acceptable (based on allowable bearing loads and shaft motion within the catcher bearing clearances) can often be significantly greater than in conventional bearings.

There is an area where careful analysis is needed for each new pump configuration. A conventional bearing, if overloaded, will accept the load with a higher wear rate, but a magnetic bearing has a sharp cut-off at the point where the flux saturation level is

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reached. At that point, any additional load will be transferred to the catcher bearing. Thus it is important to include all loads in the design specification, with the appropriate margins. All applications to date have generally shown a) that there are loads which were mistakenly considered to be insignificant, or b) that the values of the loads were underestimated due to lack of knowledge.

Magnetic bearings also must be carefully designed to handle transient loads, some of which may occur only once in a lifetime. Two approaches can be taken. One is to overdesign the bearing to handle the load without contacting the catcher bearing; the other is to allow momentary contact with the catcher bearing. Examples of these types of loads are hydraulic loads, seal touchdown loads, system loads such as water hammer, valve operation, pump switchover, pump to driver alignment loads, and seismic loading.

Rotordynamics Considerations Magnetic bearings have the capability to control the rotor dynamics of a pump very effectively. If the pump is running below its first flexible mode, this is usually straightforward. If the pump has to traverse a flexible mode, the position of the bearings and the position sensor must be such that the modes can be recognized by the position sensor, and the bearings can exert a positive restoring force to control the mode. That is, the position sensors and bearings must not be positioned at or close to a node and certainly not positioned on opposite sides of a mode. Thus, rotordynamics considerations should be taken care of very early in the configuration of the system.

The two typical rigid body modes are shown in Figure 9. The location of the bearing and position sensor is not usually an issue for these modes, but the flexible modes require that the position sensor and bearing be positioned in such a way that the rotor deflection can be measured and the bearing can exert the necessary restoring force and damping to control the mode.

Magnetic bearing controllers are programmed with a transfer function designed specifically for the pump. This transfer function, or control algorithm, provides the necessary stiffness and damping at all operating speeds to control the rotor as previously described.

The key requirements of the transfer function are that it

- Provides correct damping and stiffness to handle the rotor rigid body modes
- Provides sufficient force with the appropriate control bandwidth to handle the rotor flexible modes below the maximum operating speed

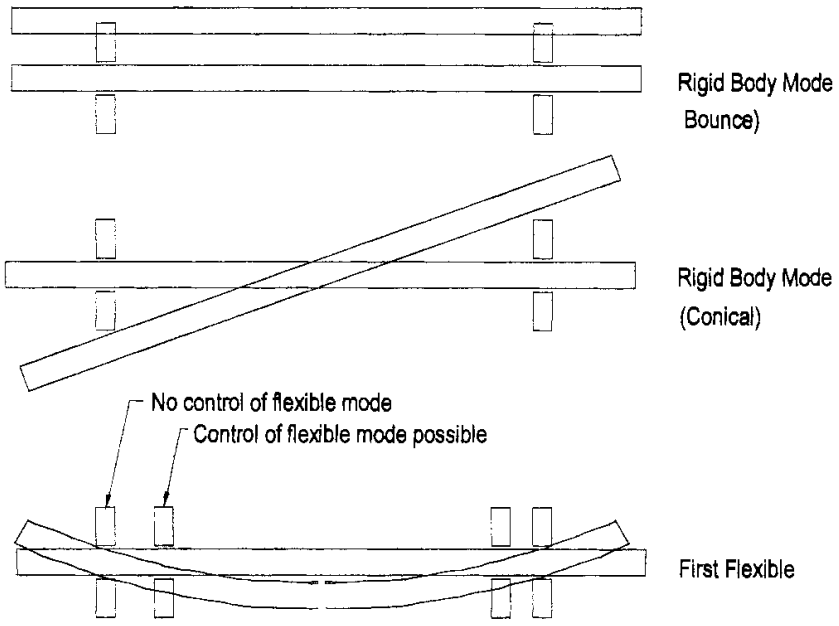


FIGURE 9 Flexible mode design considerations

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- Does not excite any rotor modes above the maximum operating speed
- Does not excite any stator vibration modes at any frequency
- Takes into account the stiffness and damping contribution from wear rings and interstage annular seals

Auxiliary or Catcher Bearings However reliable magnetic bearings become, a landing surface for maintenance will be required. Further, designing a bearing that will take all transient loads without any possibility of overload will usually result in over-sized and costly bearings. Hence catcher, bearings are expected to be required. The design limits for catcher bearings are usually established by calculating the forces that will result from a drop at operating speed.

For applications running at significant speeds, a nonlinear analysis is required to determine the motion and loading on the bearing during such a drop. The key design considerations are impact loads, heat generation during the rundown, and the response to imbalance if, when running on the catcher bearings, any critical speeds have to be traversed.

Rolling element bearings have typically been used as catcher bearings; however, sleeve bearings and bushings have been used in several applications, and are better suited to a submerged application.

MAGNETIC BEARING SIZING

The fundamental magnetic bearing sizing problem is to define the pole area at the bearing air gap that is necessary to achieve the desired force capacity without saturating the selected pole materials. Given the pole area, the minimum stator outside diameter and maximum rotor inside diameter can be determined using simple algorithms to ensure that no other part of the magnetic circuit saturates. The stator geometry must also include sufficient volume for the control and bias coils.

Approximate size and load capability Table 2 contains rough approximations for the dimensions of magnetic bearings. (Refer to Figure 1.) These are based on the experience with the pumps of Table 1. Also included are the unit load capabilities, which are given a) for radial bearings in terms of radial load F_r divided by the projected area DL of the active polar area of the bearing at the air gap, and b) for axial thrust bearings in terms of the axial load F_z divided by the active pole area of the runner disk between the inner diameter D_i and the outer diameter D_o. These approximations provide the designer and user with an idea of the design configuration of a magnetic-bearing-equipped pump. The

TABLE 2 Approximate magnetic bearing sizing relationships

Sizing Parameter	Heteropolar	Homopolar
Radial Bearings		
a) Dimensions, in multiples of the shaft diameter		
Air gap diameter, D	2	1.5
Stator outer diameter	4	3
Overall axial length	2	1.5
Active axial length of poles at air gap, L	0.9	0.8
b) Unit load capability, F_r/DL, lb/in^2 (MPa)	40 (0.28)*	60 (0.41)
Axial Thrust Bearings		
a) Diameter ratio D_i/D_o of active pole area	0.5	0.5
b) Unit load capability, $F_z / [\pi(D_o^2 - D_i^2)]$, lb/in^2 (MPa)	50 (0.34)*	70 (0.48)

*Higher with special lamination material

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low unit loads of magnetic bearings result in more space being needed for them in comparison to conventional bearings.

Theory of One-Dimensional Sizing To perform more accurate, in-depth sizing, the theory of both heteropolar and homopolar magnetic bearings is applied. Magnetic bearing sizing and geometry programs normally use simple one-dimensional magnetic circuit theory to obtain initial sizing and perform design iterations. This initial sizing is then followed up with design analysis using 2D or 3D magnetic FEA analysis to verify the design. The basis of the classic one-dimensional sizing for a magnetic bearing is discussed next, first for the heteropolar bearing and then for the homopolar bearing.

HETEROPOLAR BEARING

a. *Magnetic circuit*—The basic magnetic circuit equation, derived from Ampere's Loop Law, is

$$MMF = \Phi \mathrm{R} \tag{1}$$

where MMF = magnemotive force
Φ = magnetic flux
R = path reluctance

A sketch of one quadrant (one electromagnet) of a heteropolar bearing is shown in Figure 10.

Assuming the air gap area and path areas are equal, Eq. 1 becomes

$$2Ni = BA\left(\frac{g}{\mu_o A} + \frac{g}{\mu_o A} + \frac{l_{stat}}{\mu_o \mu_{r,\,stat} A} + \frac{l_{rot}}{\mu_o \mu_{r,\,rot} \mathrm{A}}\right) \tag{2}$$

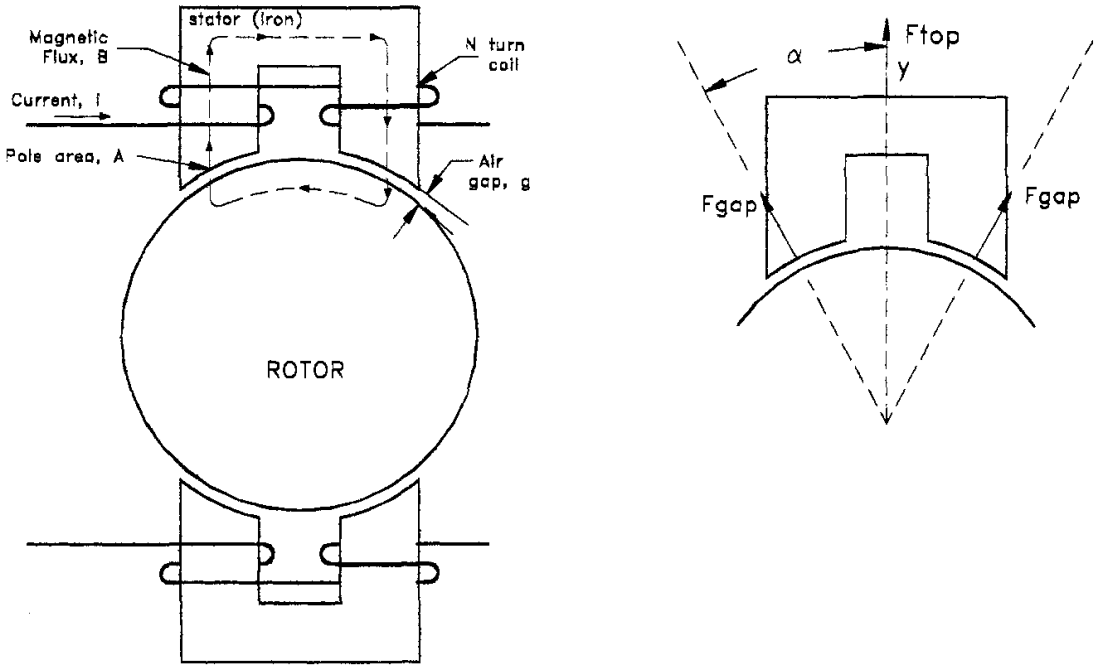


FIGURE 10 Simple electromagnetic circuit for one quadrant of a heteropolar magnetic bearing

Where N = number of turns per pole
i = coil current
B = magnetic flux density
A = pole area
g = air gap
μ_o = permeability of free space
μ_r = relative permeability
l = iron path length

If the iron permeability is high relative to the air gap ($\mu_{r,rot}, \mu_{r,stat} >> \mu_o$), the iron reluctance terms are insignificant and the following equation can be obtained for the flux:

$$B = \frac{Ni\mu_o}{g} \tag{3}$$

b. *Force calculation*—The basic force equation for an air gap is

$$F_{gap} = \frac{B^2A}{2\mu_o} \tag{4}$$

This equation assumes negligible leakage and fringing and that the flux density is uniform in the air gap. The combined vector force along the center of the bearing for the two air gaps of the top magnet is

$$F_{top} = 2F_{gap}\cos\alpha \tag{5}$$

Substituting from Eq. 4 gives

$$F_{top} = \cos\alpha\,\frac{B^2A}{\mu_o} \tag{6}$$

If the saturation flux density, B_{sat}, of the iron material is used for B, Eq. 6 defines the load capacity as a function of pole area for a heteropolar magnetic bearing.

c. *Linearization of the force/current characteristic*—Substituting from Eq. 3 gives:

$$F_{top} = \cos\alpha\,\frac{A}{\mu_o}\left(\frac{Ni\mu_o}{g}\right)^2 = \cos\alpha\,AN^2\mu_o\frac{i^2}{g^2} = k_1\frac{i^2}{g^2} \tag{7}$$

$$k_1 = \cos\alpha\,AN^2\mu_o$$

Thus, the force in a given magnet is proportional to the square of the current, a result that makes the bearing more difficult to control. Additionally, a single electromagnet can only apply a force in one direction (an attractive force). For these two reasons, opposed electromagnets are used together with a bias current (or flux) in each coil.

The current relationship is

$$i = i_{bias} + i_{con}$$

to increase the force,

$$i = i_{bias} - i_{con}$$

to decrease the force.

The rotor may also be off-center in the air gap, described by

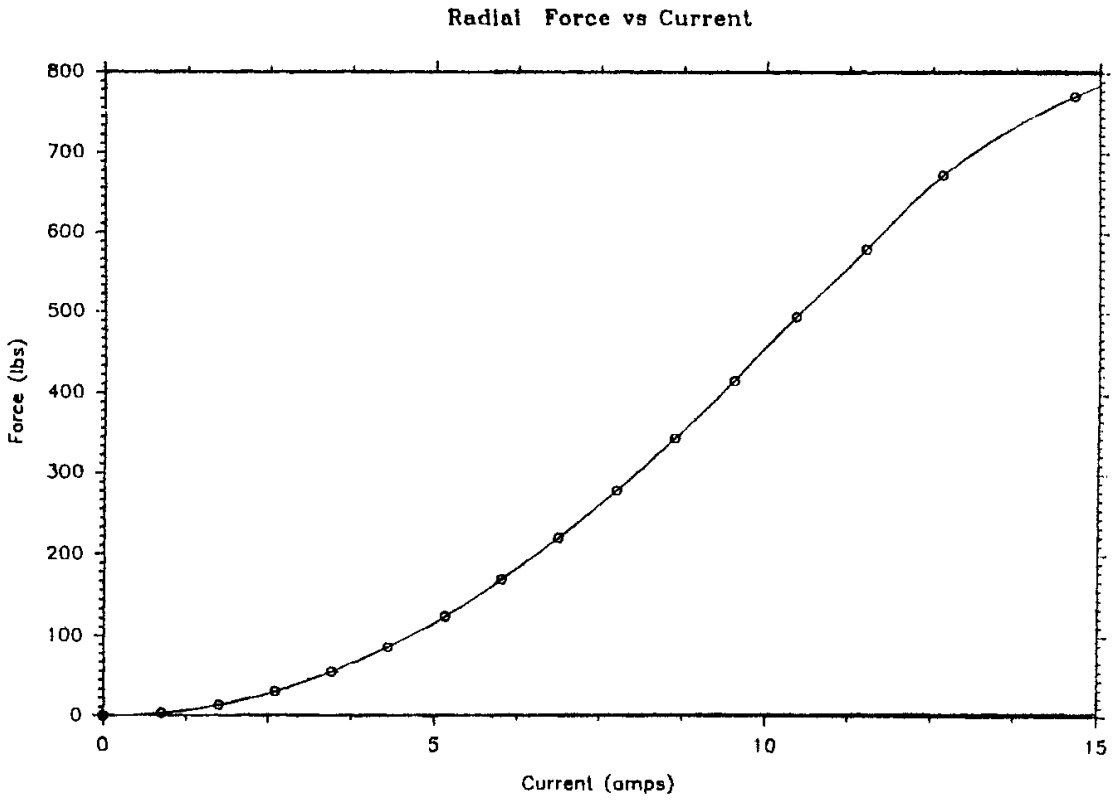


FIGURE 11 Force versus current for radial bearing of multistage pump (lb × 4.448 = N)

$$g = g_0 - y \qquad \text{top air gap}$$

$$g = g_0 + y \qquad \text{bottom air gap}$$

Applying Eq. 7 to both top and bottom electromagnets yields

$$F_y = F_{top} - F_{bot} = k_1\left[\left(\frac{i_{bias} + i_{con}}{g_0 - y}\right)^2_{top} - \left(\frac{i_{bias} - i_{con}}{g_0 + y}\right)^2_{bot}\right] \tag{8}$$

With the rotor centered ($y = 0$), this can be reduced to

$$F_y = 4\frac{k_1}{g_0^2} i_{bias} i_{con} \tag{9}$$

Thus the net bearing force is proportional to the control current. Figures 11 and 12 are examples of measured force versus current for radial and axial thrust bearings respectively. These measurements were made on the magnetic bearing of the multistage pump in Table 1 (refer to Figures 1 and 2).

d. *Force constant and negative stiffness*—Eq. 8 can be also be linearized for small motion about the center ($y \ll g_0$) by differentiating with respect to i_{con} and y, the two quantities in Eq. 8 that can change in normal operation of the bearing. The result is

$$F_y = -k_f i_{con} - k_n y \tag{10}$$

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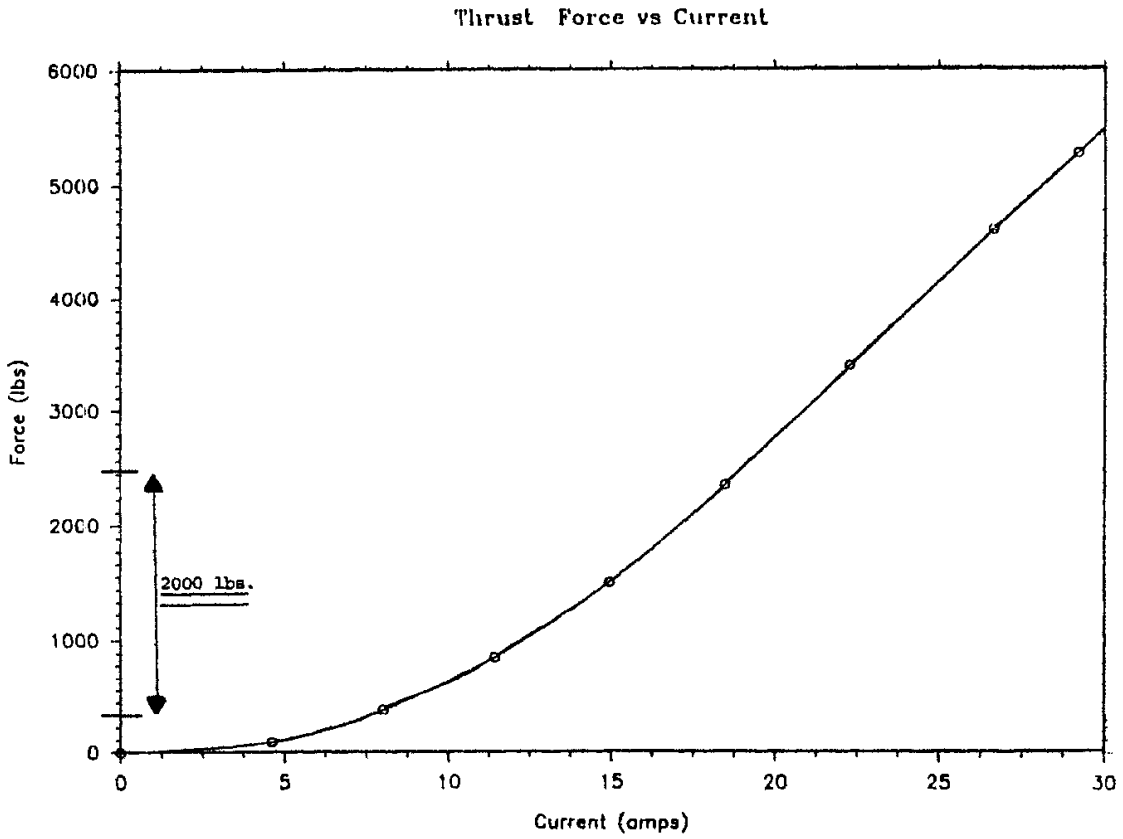


FIGURE 12 Force versus current for axial thrust bearing of multistage pump (lb × 4.448 = N)

where

$$k_f = \frac{-4k_1 i_{bias}}{g_0^2} = \text{force constant or current stiffness} \tag{11}$$

$$k_n = \frac{-4k_1 i_{bias}^2}{g_0^3} = \text{negative stiffness or position stiffness} \tag{12}$$

The position stiffness is the passive stiffness of the bearing with a bias field but with no control current. The position stiffness is always negative, indicating that if the rotor is displaced from center, it will be pulled further from its equilibrium position if no control current is applied. The force constant defines the relationship of the control force to current (lb/amp or N/amp) with the bearing centered. It is also negative, indicating that applying a control current pulls the rotor from its centered position. Many practitioners use positive values for the force constant and the position stiffness as a matter of convenience. In this case, the minus signs in Eq. 10 become plus signs.

The control current is determined by the measured displacement, y, and the characteristics of an adjustable sensor/compensator/amplifier transfer function:

$$G_{con}(s) = \frac{i_{con}}{y} \tag{13}$$

Substituting Eq. 11 into Eq. 10 gives

$$Fy = -k_F G_{con}(s)y - k_n y \tag{14}$$

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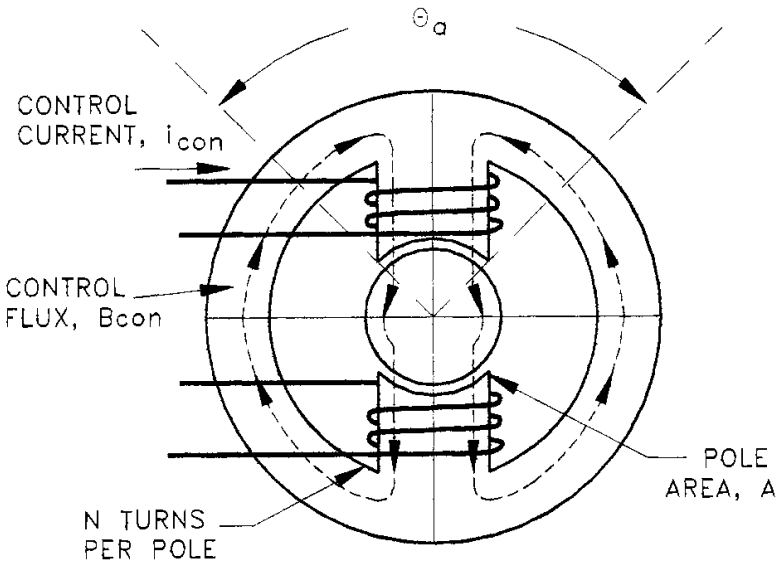


FIGURE 13 Control magnetic circuit for one axis of a homopolar magnetic bearing

The characteristics of the control transfer function $G_{con}(s)$ are determined in order to stabilize the rotor/bearing system.

HOMOPOLAR BEARING In the homopolar bearing, a bias flux is used for linearization just as in the heteropolar bearing; however, the bias flux and control flux follow different paths. Additionally, the bias flux can be generated by either a permanent magnet (most common) or an electromagnetic coil. The use of a permanent magnet for bias reduces power consumption and makes the bearing more linear at large position offsets.

a. *Magnetic control circuit*—The control circuit for one half of the homopolar bearing, shown in Figure 13, is across the top air gap, through the rotor, out the bottom air gap, and around the back iron.

When Eq. 1 is applied to this circuit, the result is similar to Eq. 2:

$$2Ni_{con} = B_{con}A\left(\frac{g}{\mu_o A} + \frac{g}{\mu_o A} + \frac{l_{stat}}{\mu_o \mu_{r,stat} A} + \frac{l_{rot}}{\mu_o \mu_{r,rot} A}\right) \tag{15}$$

Again, if the iron permeability is high relative to the air gap, this can be reduced to the following:

$$B_{con} = \frac{Ni_{con}\mu_o}{g} \tag{16}$$

b. *Magnetic bias circuit*—The magnetic circuit equation for the bias circuit, shown in Figure 14, is

$$\frac{B_r l_m}{\mu_o \mu_{r,\,mag}} = B_{bias}A\left(\frac{g}{\mu_o A} + \frac{g}{\mu_o A} + \frac{l_m}{\mu_o \mu_{r,\,mag} A_m}\right) \tag{17}$$

Where B_r = residual induction of the magnet
l_m = axial length of the magnet
$\mu_{r,\,mag}$ = relative permeability of the magnet (≅1.0–1.05)
A_m = magnet cross-sectional area per quadrant

With the magnet permeability assumed to be 1.0, this can be reduced to

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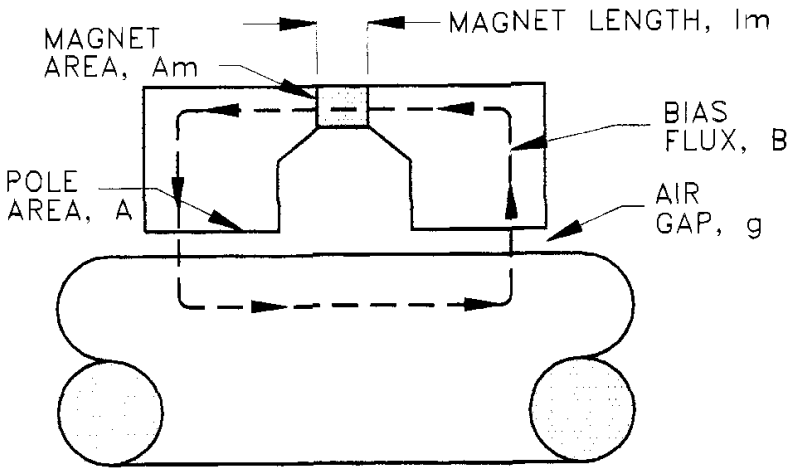


FIGURE 14 Bias magnetic circuit for one quadrant of a homopolar magnetic bearing

$$B_{bias} = \frac{B_r}{\frac{2g}{l_m} + \frac{A}{A_m}} \tag{18}$$

c. *Force calculation*—The basic force equation for an air gap is given by Eq. 4. In the homopolar bearing, the pole face covers a broader arc of the rotor than in the heteropolar bearing; therefore, the air gap force must be integrated over the surface to obtain the desired vector force, F_{top}. Both the front and back control stacks must also be included. The result is

$$F_{top} = 2C_g F_{gap}$$

Where:

$$C_g = \frac{2\sin(\theta_{a/2})}{\theta_a} \tag{19}$$

This result assumes negligible leakage and fringing and that the flux density is uniform in the air gap. Substituting from Eq. 4 into Eq. 19 gives

$$F_{top} = C_g \frac{B^2 A}{\mu_o} \tag{20}$$

If the saturation flux density, B_{sat}, of the iron material is used for B, Eq. 20 defines the load capacity as a function of pole area for a magnetic bearing.

d. *Linearization of the force/current characteristic*—The air gap flux density in the homopolar bearing is the superposition of the bias and control flux. The control coils and permanent magnet polarity are arranged such that when the control flux adds to the bias in the top air gaps, the control subtracts from the bias in the bottom air gaps. Thus the net vertical force is

$$F_y = F_{top} - F_{bot} = \frac{C_g A}{\mu_o}\left[(B_{bias} + B_{con})^2_{top} - (B_{bias} - B_{con})^2_{bot}\right] \tag{21}$$

This can be reduced in a similar manner as before to produce

$$F_y = \frac{4C_g A}{\mu_o} B_{bias} B_{con} \tag{22}$$

Substituting from Eq. 16

$$F_y = \frac{4C_g AN}{g} B_{bias}\, i_{con} \tag{23}$$

Thus the control force is proportional to the control current as desired.

e. *Force constant and negative stiffness*—The expression given in Eq. 10 for the heteropolar bearing also applies to the homopolar bearing:

$$Fy = -k_f\, i_{con} - k_n\, y \tag{24}$$

The definition of the force constant is

$$k_f = \frac{-4C_g ANB_{bias}}{g_0} = \text{force constant or current stiffness} \tag{25}$$

The expression for negative stiffness is not easily reducible to analytical form due to the complexity of the bias circuit. However, the existence of the permanent magnet in the bias flux path as a fixed and large reluctance improves the linearity of this bearing for off-center operation. Eqs. 13 and 14 for the heteropolar bearing apply to the homopolar bearing as well.

INSTALLATION AND TUNING

Mechanical Installation Magnetic bearings have the advantage that they can be set, by adjusting offsets in the controller, to center the rotor on the magnetic bearing stator, the catcher bearing, the seals, or any other mechanical reference in the pump. The rotor can even be offset vertically to cancel the effect of gravity, thus reducing static power requirements to near zero. Careful consideration is needed during design to decide which is the best approach.

The position sensors in the bearings can also be used to measure the clearances between the rotor and any physical stops such as a sealing ring, without disassembly.

Tuning The rotor dynamics analysis performed during the early design stage will be used to determine the initial controller compensation, which will have a transfer function matched to the pump requirements. During initial testing, this transfer function will have to be adjusted to match the as-built dynamics of the rotor and support structure.

Normally the rotor will be accurately modeled and little change will be needed. If there are shrink fits or bolted joints, there may be some stiffness variation from the theoretical model. This may require on-site controller compensation adjustment. However, the stator is often a complex structure, and adjustments may be needed to avoid the excitation of stator modes.

DIAGNOSTICS AND USER INTERFACE

Diagnostic Capabilities The controller, in order to function, must analyze a continuous stream of information on the shaft location in each of the five control axes, two for each radial bearing and one for the thrust bearing. This information can be accessed for external diagnostic use, as can the corresponding information on bearing current, from which can be inferred the bearing load. This diagnostic information can be a very useful source of information on the health of the pump, its mechanical components, and on the system it is operating in.

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Interface Requirements The magnetic bearing system controller can also be interfaced with the plant control system with the following type of logic:

- No drive unit start without levitation
- No delevitation at speed
- Rotor offset and bearing load alarms
- Rotor offset and bearing load driver trips, with possible time delay

RELIABILITY AND MAINTENANCE

Bearing Cartridges As explained earlier, the reliability of the bearing stator and rotor components should be such as to provide lifetime service.

Controller The main life-limiting component in the controller is likely to be the amplifier. In a redundant system, online replacement is possible without loss of levitation. In a nonredundant system, a preventative maintenance approach should be used for this component.

OPERATING EXPERIENCE

Multistage Boiler Feed Pump The multistage pump of Table 1 (refer to Figure 1) was installed as one of three otherwise identical pumps (two in parallel, one standby) in an electric utility generating plant (refer to Figure 2). The objective was to show that magnetic bearings would work in a typical field application of a pump of significant power level. This project took the first step of replacing the conventional bearings in this 610 hp (0.46 MW) eight-stage centrifugal pump, which were outboard of the pump itself, and replacing them with heteropolar active magnetic bearings without any major design changes[6]. This was seen as the first of two steps, the second being a project where the bearings would be submerged in the operating fluid, allowing one seal system to be replaced[8].

As with many magnetic bearing projects, the main lesson learned was that the transient bearing loads could not always be predicted ahead of time, and that the magnetic bearings gave very precise and important feedback of this information. Table 3 contains the design and field data. In this case, the 2500 lb (11 kN) transient load occurred when the plant underwent a suction pressure transient in which the available NPSH became so low that the first stage of this horizontally-opposed staging configuration (refer to Figure 1) apparently lost pressure rise completely[6]. The axial thrust of this stage was accordingly lost, destroying the intended axial hydrodynamic thrust balance of the pump. (See Sections 2.1.1 and 2.2.1.) Nevertheless, the conservative design of the thrust bearing enabled it to accommodate this load.

TABLE 3 Operating experience with multistage pump

	Expected Load, lb (kN)		Design Load, lb (kN)	Actual Load, lb (kN)	
Bearing	Steady	Transient		Steady	Transient
Radial	280 (1.2)	560 (2.5)	800 (3.6)	280 (1.2)	580 (2.6)
Axial	1,000 (4.4)	2,000 (8.9)	4,000 (17.8)	1,100 (4.9)	2,500 (11.1)

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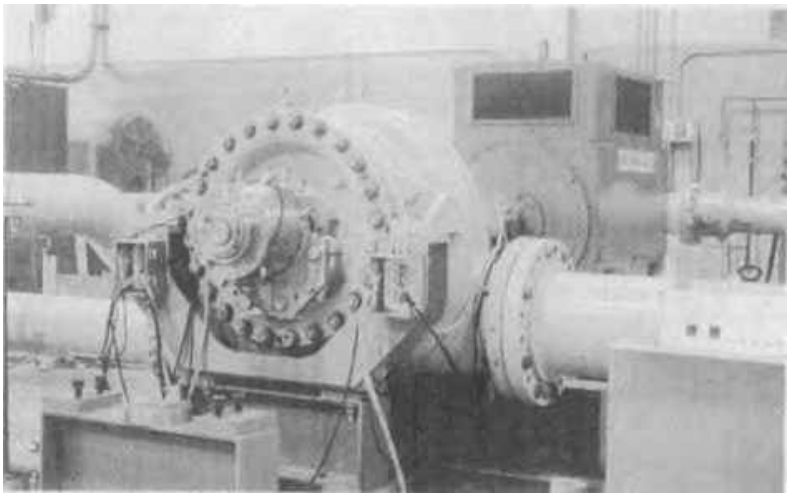


FIGURE 15 Single-stage double-suction 10 × 26 size process pump [800 hp (0.6 MW)] with magnetic bearing (Reference 7)

Single-Stage Process Pump The 800 hp (0.6 MW) single-stage double-suction process pump of size 8 × 26 (200 mm × 660 mm) of Table 1 was retrofitted with homopolar bearings[7].

Closed-loop testing was conducted in the pump manufacturer's facility (see Figure 15). As indicated in Table 4, the results showed that a substantial operating margin exists for the axial thrust bearing. Greater design capacity would provide the same margin with respect to radial loads.

The margins evident in Table 3 for the multistage pump may appear excessive, but until more operating knowledge about such pumps is acquired, it would appear that this degree of conservatism in designing magnetic bearings for pumps is merited.

COSTS

Backed by the vision of complete magnetic suspension and the attendant benefits, magnetic bearing technology has been proven and demonstrated in pumping machinery. However, the major deterrent to further application of this technology is cost. Retrofitting the magnetic bearings to the multistage pump described above cost at least twice the price of the pump itself, and this included the analog controller, which accounted for about a third of the retrofit cost. Digital controllers, a later development, are one-third the size and cost of analog controllers; this has significantly reduced the overall cost. Much of this overall cost is in the engineering of the magnetic bearings, which includes matching this system to the rotordynamic characteristics of the pump. Use of the same system in quantity production would reduce the cost by up to 80%.

TABLE 4 Design and experimental loads for the single-stage pump

Bearing	Expected Load, lb, (kN)	Design Load, lb (kN)	Actual Load, lb (kN)
Radial	953 (4.2)	1,430 (6.4)	1,450 (6.4)
Axial	1,000 (4.4)	4,000 (17.8)	2,100 (9.3)

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REFERENCES

1. *Marks Standard Handbook for Mechanical Engineers.* 9th ed. E. A. Avallone and T. Baumeister, eds., McGraw-Hill, 1987, pp. 3–56.
2. McCloskey, T., and Jones, G. "Electric Utility Applications for Active Magnetic Bearings." *Proceedings of MAG '92*, Magnetic Bearings, Magnetic Drives and Dry Gas Seals Conference and Exhibition, University of Virginia, July 1992, pp. 3–18.
3. Cooper, P., McGinnis, G., Janik, G., Jones, G., and Shultz, R. "Application of Magnetic Bearings in a Multistage Boiler Feed Pump." *Proceedings of the Second International Symposium on Magnetic Bearings*, July 1990.
4. McGinnis, G., Cooper, P., Janik, G., Jones, G., and Shultz, R. "A Boiler Feed Pump Employing Active Magnetic Bearings." Presented at International Joint Power Generation Conference, Boston, MA, ASME, October 1990, ASME.
5. Cooper, P., and Jones, G. "Operating Experience, Including Transient Response, of a Magnetic-Bearing-Equipped Boiler Feed Pump." *Proceedings of MAG '92*, Magnetic Bearings, Magnetic Drives and Dry Gas Seals Conference and Exhibition, University of Virginia, July 1992, pp. 19–28.
6. Jones, G., and Penfield, S. Jr. "Magnetic Bearing Boiler Feed Pump Demonstration", Research Report EP89-39, Empire State Electric Energy Research Corporation, October 1992.
7. Brown, E., Thorp, J. M., Hawkins, L., and Sloteman, D. "Development and Application of a Homopolar, Permanent-Magnet-Bias Magnetic Bearing System for an API 610 Pump." *Saudi Aramco Journal of Technology,* Spring 1997, pp. 2–13.
8. Hanson, L., and Imlach, J. "Development of a Magnetic Bearing API Process Pump with a Canned Motor." *Proceedings of the Ninth International Pump Users Symposium,* Texas A&M University, March 1992, pp. 3–8.
9. "Guidelines for the Use of Magnetic Bearings in Turbomachinery." Prepared by Technology Insights, San Diego, California and published by the Electric Power Research Institute, February 1996.
10. Meeks, C. R., DiRusso, E., and Brown, G. V. "Development of a Compact, Lightweight Magnetic Bearing." *Proceedings of the 26th Joint Propulsion Conference.* AIAA/SAE/ASME/ASEE, Orlando, FL, 1990.

C • H • A • P • T • E • R • 7

JET PUMPS

SECTION 7.1
JET PUMP THEORY

RICHARD G. CUNNINGHAM

INTRODUCTION

The jet pump transfers energy from a liquid or gas primary fluid to a secondary fluid. The latter may be a liquid, a gas, a two-phase gas-in-liquid mixture, or solid particles transported in a gas or a liquid. Examples of all these combinations have been reported in the technical literature. Reference 1, the major bibliography in this field, contains over 400 abstracts. Although the terms "ejector" and "eductor" are also applied, the term "jet pump" will be used here. The jet pump offers significant advantages over mechanical pumps: no moving parts for improved reliability, adaptability to installation in remote or hazardous environments, simplicity, and low cost. The primary drawback is efficiency: both frictional losses and unavoidable mixing losses are incurred. Nevertheless, careful design can produce pumps with efficiencies on the order of 30–40%. The jet pump in Figure 1 is typical of liquid-jet pumps and low Mach-number gas-jet/gas pumps. Compressible-flow pumps, for example, steam-jet ejectors, employ converging-diverging nozzles for full expansion of the jet.

NOMENCLATURE

A = area, ft^2 (m^2)
A_w = throat wall area, ft^2 (m^2)
C = velocity of sound, ft/sec (m/s)
CR = cavitation resistance
D = diameter, ft (m)

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E = energy rate, ft lb/sec (joule/s)
K = friction loss coefficient
°K = absolute temperature, Kelvin
LJL = liquid-jet liquid pump
LJG = liquid-jet gas compressor
LJGL = liquid-jet gas and liquid pump
M = liquid/liquid flow ratio, Q_2/Q_1
MN = Mach number
N = pressure ratio, LJL jet pump
NPSH = net positive suction head
P, P = pressure: static, total psia (kPa abs.)
P_v = vapor pressure, psia (kPa abs.)
Q = volumetric rate, ft³/sec (m³/s)
R = gas constant, ft lbs/slug °R (joules/kg °K)
°R = absolute temperature, Rankine
S = density ratio, ρ_2/ρ_1
T = temperature °R (°K)
V = velocity, ft/sec (m/sec)
W = work rate, ft lbs/sec (joules/s)
Z = jet dynamic pressure, psi (kPa)
a = diffuser area ratio, A_t/A_d
b = jet pump area ratio, A_n/A_t
$c = A_{2G0}/A_n = (A_t - A_n)/A_n = (1 - b)/b$
m = mass flow rate, slugs/s (kg/s)
s = seconds
psi = pounds per square inch
psia = pounds per square inch, absolute
r_v = gas/liquid vol. flow rate ratio Q_G/Q_2
r_{v0} = gas/liquid vol. flow rate ratio at 0: Q_{G0}/Q_2
r_m = gas/liquid mass flow-rate ratio
sp = nozzle-to-throat spacing, ft(m)
sp/D_{th} = spacing, throat diameters
η = efficiency
γ = gas density ratio at s, ρ_{Gs}/ρ_1
$\gamma\phi_s = m_{Gs}/m_1$
ρ = density, slugs/ft³ (kg/m³)
τ = shear stress, psi (kPa)
ϕ = gas flow ratio Q_G/Q_1
ϕ_s = gas flow ratio Q_{Gs}/Q_1 at s

Subscripts

1 = liquid primary flow
2 = liquid secondary flow

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G = gas secondary flow
$2G$ = bubbly secondary flow
$2G0$ = bubbly sec. flow at 0
mep = maximum efficiency point
op = operating point
L = limit for cavitating flow
3 = combined fluids 1, 2, and G
i, s, n = locations (see Figure 1)
$0, t, d$ = locations (see Figure 1)
$c0$ = flow-ratio cut off
f = friction loss
n = nozzle
en = throat entry
th = mixing throat
di = diffuser
td = throat and diffuser

Subscript Examples

Q_2 = secondary liquid vol. flow rate
Q_{G0} = secondary gas flow rate, at 0
Q_{2G0} = flow rate of bubbly mixture of gas in the secondary liquid, at 0

NOTE: The convention for pressure and stress in this section is to use psi (and psia) for pounds per square inch (and pounds per square inch absolute) instead of the conventional lb/in^2 used throughout the rest of the handbook.

LIQUID-JET PUMP THEORY FOR THREE SECONDARY-FLOW TYPES

The liquid-jet pump model is based on conservation equations for energy, momentum, and mass. Real-fluid losses are accounted for by friction-loss coefficients (K). The primary or motive fluid is a liquid of density ρ_1. In the following derivation, the secondary/pumped fluid

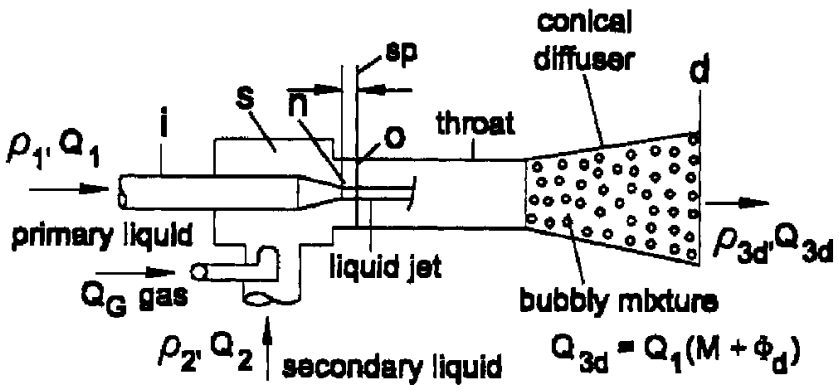


FIGURE 1 Liquid-jet gas and liquid pump and nomenclature

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can be a second liquid of density ρ_1 or ρ_2, or a gas-in-liquid bubbly mixture, or a gas. These three jet pump flow regimes are referred to as liquid-jet liquid (LJL), liquid-jet gas liquid (LJGL), and liquid-jet gas (LJG). Equations (1), (3), (5), and (7) below apply to all three.

Assumptions:

a. The primary and secondary streams enter the mixing throat with uniform velocity distributions, and the mixed flows leave the throat and diffuser with a uniform velocity profile.

b. The gas phase—if present—undergoes isothermal compression in the throat and diffuser.

c. All two-phase flows at the throat entry and exit consist of homogeneous bubble mixtures of a gas in a continuous liquid.

d. Heat transfer from the gas to the liquid is negligible—the liquid temperature remains constant.

e. Change in solubility of the gas in the liquid from pressure P_s to P_d is negligible.

f. Vapor evolution from and condensation to the liquid are negligibly small.

NOZZLE EQUATION With reference to Figure 1

$$P_i + \rho_1 \frac{V_i^2}{2} = P_o + \rho_1 \frac{V_n^2}{2} + K_n \rho_1 \frac{V_n^2}{2} \tag{1}$$

For

$$P_i = \overline{P_i}$$

the nozzle equation is

$$P_i - P_O = Z(1 + K_n)$$

THROAT-ENTRY EQUATION The two-phase secondary flow is described by

$$\frac{dP}{\rho} = VdV + d\left(K_{en}\frac{V^2}{2}\right) = 0 \tag{2}$$

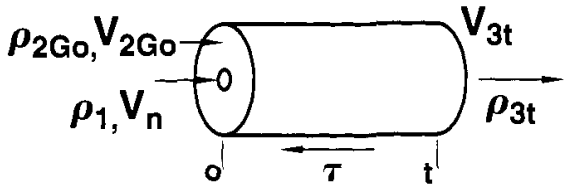
Density of the secondary fluid as a function of static pressure and flow ratios M and ϕ is

$$\rho_{2G} = \frac{m_2 + m_G}{Q_2 + Q_G} = \frac{m_1\left[\frac{m_2}{m_1} + \frac{m_G}{m_1}\right]}{Q_1(M + \phi)} = \rho_1\left[\frac{SM + \gamma\phi_s}{m + \phi}\right]$$

Integration of Eq. (2) using this density relation and continuity results in the throat-entry equation:

$$M(P_s - P_o) + P_s\phi_s \ln\frac{P_s}{P_o} = Z\frac{SM + \gamma\phi_s}{c^2}(1 + K_{en})(M + \phi_o)^2 \tag{3}$$

MIXING THROAT MOMENTUM EQUATION Equating control volume forces and fluid momentum changes:



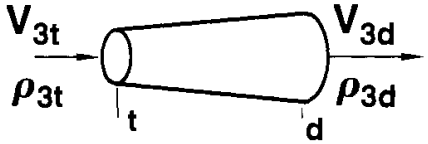
$$(P_0 - P_t)A_{th} - \tau A_w = (m_1 + m_2 + m_G)\, V_{3t} m_1 V_n - (m_2 + m_G)\, V_{2G0} \tag{4}$$

where $$\frac{\tau A_w}{A_{th}} = \frac{K_{th}\rho_{3t}V_{3t}^2}{2}$$

Substituting the density and continuity relations in Eq. (4) and diving A_{th} produces a quadratic equation in P_t. The throat momentum equation becomes

$$\begin{aligned} P_t^2 - Z\Big[& 2b - b^2(2 + K_{th})(1 + SM + \gamma\phi_s)(1 + M) \\ & + 2(SM + \gamma\phi_s)(M + \phi_0)\frac{b^2}{(1 - b)} + \frac{P_0}{Z}\Big]P_t \\ & + Z[b^2(2 + K_{th})(1 + SM + \gamma\phi_s)P_s\phi_s] = 0 \end{aligned} \tag{5}$$

DIFFUSER EQUATION The mixed primary and secondary fluids flow from t to d:



The t-d flow is described by Eq. (6):

$$\int_t^d \frac{dP}{\rho} + \int_t^d V dV + \int_t^d \frac{\Delta P_f}{\rho_{3t}} = 0 \tag{6}$$

Integrating Eq. (6) and substituting the density and continuity relations, the diffuser equation becomes

$$\begin{aligned} (P_d - P_t) + \frac{P_s\phi_s}{1 + M}\ln\frac{P_d}{P_t} = Zb^2\left[\frac{1 + SM + \gamma\phi_s}{1 + M}\right] \\ \times [(1 + M + \phi_t)^2 - a^2(1 + M + \phi_d)^2 - K_{di}(1 + M + \phi_t)(1 + M)] \end{aligned} \tag{7}$$

LIQUID-JET PUMP EQUATIONS The liquid jet pump is described in terms of the four flow processes by Equations (1), (3), (5), and (7). The term $\gamma\phi_s = \rho_{Gs} Q_{Gs}/\rho_1 Q_1$. For isothermal flow, $\gamma\phi_s = 144P_s\phi_s/\mathbf{R}T_s\rho_1$. For air, with $\mathbf{R}$ = 1716 ft lb/slug °R, and water, ρ_1 = 1.94 slugs/ft^3 and $\gamma\phi_s = .0432P_s\phi_s/T_s$. In SI, $\mathbf{R}$ = 286.92 mN/kg °K, ρ_1 = 1000 kg/m^3, and $\gamma\phi_s = .00348P_s\phi_s/T_s$, with P_s in kN/m2, and T_s in °K. The $\mathbf{R}$ and/or ρ_1 values must of course be replaced for fluids other than water and air.

In the LJGL pump, the secondary flow is a bubbly mixture of a gas in a liquid. Compressible flow phenomena must be considered because the velocity of sound is quite low in bubbly fluids. For example, the velocity of sound in a 50/50 uniform mixture of air in water is about 70 ft/s (21.3 m/s), far below sonic velocities in air (1100 ft/s or 335 m/s) or water (5000 ft/s or 1524 m/s).

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The Mach number of a bubbly secondary flow at the throat entrance is (see Reference 5):

$$\mathbf{MN}_{2G0} = \frac{V_{2G0}}{C_{2G0}} = \frac{\phi_0}{c}\sqrt{\frac{2Z}{P_s\phi_s}(SM + \gamma\phi_s)} \tag{8}$$

When this Mach number is 1.0, the LJGL pump has reached limiting flow; that is, a reduction in back pressure P_d no longer causes an increase in the bubbly secondary-flow rate.

Pump Efficiencies The LJGL pump produces two useful work results:

- Static-pressure increase of the liquid component of the secondary flow stream.
- If a gas is entrained in this liquid stream, isothermal compression of the gas component.

With W as the work rate, ft-lb/s, (power) $W_L = Q_2 (P_d - P_s)$ is the work rate on the liquid component, and $W_G = \rho_{Gs} Q_{Gs} \mathbf{R} T ln(P_d - P_s)$ is the work rate on the gas component. The energy rate input is $E_{\text{in}} = Q_1(P_i - P_s)$. The LJGL pump mechanical efficiency is the total work rate divided by the energy rate in

$$\eta = \frac{M(P_d - P_s) + P_s\phi_s \ln \dfrac{P_d}{P_s}}{\mathrm{P_i} - \mathrm{P_d}} = \eta_L + \eta_G \tag{9}$$

The Jet Loss Jet pumps in practical applications have nozzle-to-throat spacings sp/*Dth* of one or more mixing-throat diameters. The power jet traverses from a static pressure at or near P_s down to P_o, with no useful work recognized in the one-dimensional theory. Thus a "jet loss" occurs, which is in addition to the frictional and mixing losses (see Reference 9).

In the LJL and LJGL (but not the LJG) pumps, throat-inlet pressure drops—and hence jet losses—are significant (Reference 6).

Pump Efficiency, Incorporating Jet Loss In Eq. (9), $(P_i - P_d)$ is expanded: $(P_i - P_d) = (P_i - P_s) - (P_d - P_s)$, and $(P_i - P_s) = (P_i - P_o) - (P_s - P_o) = Z(1 + K_n) - j(P_s - P_o)$, where $j = 1$ for a fully inserted nozzle, *no jet loss:* and $j = 0$ for the usual case of retracted nozzle, which produces *full jet loss.* Eq. 9 now becomes

$$\eta = \frac{M(P_d - P_s) + P_s\phi_s \ln \dfrac{P_d}{P_s}}{Z(1 + K_n) - j(P_s - P_o) - (P_d - P_s)} = \eta_L + \eta_G \tag{10}$$

Eq. (10) is recommended for predicting liquid-jet pump efficiencies as follows: Use $j = 0$ for pumps with normally-retracted nozzles (full jet loss); use $j = 1$ for no-jet-loss pumps (thin-walled nozzle tip fully inserted so sp = 0). The pressure in Eq. (10) should be calculated from the one-dimensional theory using Eqs (1), (3), (5), and (7). (See below for the LJL jet pump.)

Computer Programs for LJGL and LJG Models Solutions for the compressible flow cases are generated using computer spreadsheet or Fortran programs. Values for Z, b, P_s, Ts, $\mathbf{R}$, ρ_1, S and the four K coefficients are fixed/assumed for each pump and operating conditions. Eqs. (3), (5), and (7) are then solved for each step increase in flow-ratio M, with ϕ_s held constant. Alternatively, M may be held constant and the equations solved for step increase in ϕ_s. Eqs. (3), (5), and (7) are interdependent: solution of Eq. (5) requires P_o values from Eq. (3) and solution of Eq. (7) requires P_t values from Eq. (5). The program outputs at each flow-ratio step are static pressure P_o, $P_t P_d$, and the three pump efficiencies defined by Eq. 10.

LJGL FLOW CUT-OFF Compressible-flow choking of the secondary stream at the throat entrance will occur at $\mathbf{MN}_o = 1$. The flow ratio at which this will occur can be predicted from critical-flow theory. For further details, see Reference 5.

Performance of the gas compressor (LJG) can be calculated from Eqs. (1), (3), (5), and (7) by setting $M = 0$. Although simplified, the equations are still coupled as in the LJGL case. The one-dimensional theory predicts actual performance quite well (see References 6 and 7) provided the mixing is completed within the length of the mixing throat. Theory-experiment agreement fails and the gas compressor efficiency declines—mixing is allowed to extend into the diffuser.

THE LIQUID-JET LIQUID (LJL) PUMP

Equations for the LJL pump are much simpler than the corresponding LJGL and LJG equations because of the elimination of all ϕ terms. In this case, Eqs. (1), (3), (5), and (7) reduce to the following set:

Nozzle $$P_i - P_o = Z(1 + K_n) \tag{1}$$

Throat Entry $$P_s - P_o = ZS(1 + K_{en})M^2/c^2 \tag{11}$$

Throat $$P_t - Po = Z[2b + 2SM^2b^2/(1 - b) - b^2(2 + K_{th})(1 + SM)(1 + M)] \tag{12}$$

Diffuser $$P_d - P_t = Z_b^2(1 + SM)(1 + M)(1 - K_{di} - a^2) \tag{13}$$

Pump efficiency η is defined as the ratio of useful work rate on the secondary fluid Q_2 to the energy extracted from the primary liquid:

$$\eta = Q_2(P_d - P_s)/Q_1(P_i - P_d) = MN \tag{14}$$

Two other definitions of efficiency are found in the literature, as follows:

$$\eta' = Q_2(P_d - P_s)/Q_1(P_i - P_s) = \eta/N + 1$$

$$\eta'' = (M + 1)(P_d - P_s)/(P_i - P_s)$$

These two other definitions assume that the primary/power stream pressure falls to P_s, not P_d, as shown in Figure 2. Efficiency conversions are possible only if all three pressures and two flow rates involved are given. Comparisons of efficiencies reported in the literature should be made with caution.

Combining Eqs. 1, 11–14, the theoretical pressure characteristic N for the LJL pump is

$$N = \frac{2b + \dfrac{2SM^2b^2}{1 - b} - b^2(1 + K_{td} + a^2)(1 + M)(1 + SM) - \left(\dfrac{SM^2}{c^2}\right)(1 + K_{en})}{1 + K_n - 2b - \dfrac{2SM^2b^2}{1 - b} + b^2(1 + K_{td} + a^2)(1 + M)(1 + SM) + (1 - j)\left(\dfrac{SM^2}{c^2}\right)(1 + K_{en})} \tag{15}$$

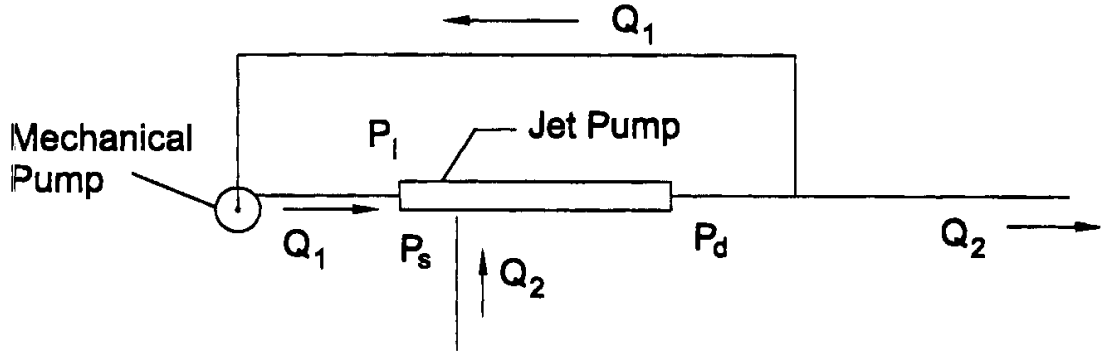


FIGURE 2 LJL jet pump and installation

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where $K_{td} = K_{th} + K_{di}$. The term $j = 0$ for the normal (jet loss) case, and $J = 1$ for no jet loss. Jet pumps normally are designed with a finite nozzle-throat spacing, typically $sp/D_{th} = 1$; jet loss is experienced, and thus $j = 0$ for this (usual) case. Eq. (15) becomes

$$N = \frac{2b + \dfrac{2SM^2b^2}{1-b} - b^2(1 + K_{td} + a^2)(1 + M)(1 + SM) - \left(\dfrac{SM^2}{c^2}\right)(1 + K_{en})}{1 + K_n - \text{numerator}} \tag{16}$$

In terms of pressure,

$$N = (P_d - P_s)/(P_i - P_d)$$

Two simplifications of Eq. (16) are often appropriate: 1) The area ratio term $a^2 = 0$ for the usual 5°–8° included-angle diffuser ($a = A_t/A_d$ is small). 2) The density ratio $S = 1$ for similar primary and secondary liquids, for example, a water primary jet pumping water as the secondary fluid. With these simplifications, Eq. (16) for the normal (jet loss) case becomes

$$N = \frac{2b + \dfrac{2SM^2b^2}{1-b} - b^2(1 + K_{td})(1 + M)^2 - \left(\dfrac{M^2}{c^2}\right)(1 + K_{en})}{1 + K_n - \text{numerator}} \tag{17}$$

And pump efficiency is

$$\eta = \eta_L = MN \tag{18}$$

For the LJL jet pump, note also that

$$(P_d - P_s)/(P_i - P_s) = N/(N + 1) \tag{19}$$

Computer Programs for LJL Models It is convenient to use Eqs. (17) and (18) in spreadsheet form to generate tables of "$N(b, M, K_n, K_{td}, K_{en})$;" that is, the LJL jet pump pressure characteristic N as a function of the bracketed variables. Typically the Ks and the area ratio b are held constant and a table is generated using step increases in M to show resultant N values. Table 1 shows $N(M)$ for $b = .25$; and $K_n = .05$, $K_{td} = .2$, and $K_{en} = 0$. The performance of this pump is shown in Figure 3 as $N(M)$ and $\eta(M)$.

TABLE 1 Performance versus M, for $b = .25$, $K_n = .05$, $K_{td} = .2$, $K_{en} = 0$

M	$(P_d - P_s)/Z$	$(P_i - P_d)/Z$	N	$\eta\%$
0.000	0.4250	0.6250	0.6800	0.00
0.200	0.3942	0.6558	0.6012	12.02
0.400	0.3619	0.6881	0.5259	21.04
0.600	0.3280	0.7220	0.4543	27.26
0.800	0.2926	0.7574	0.3862	30.90
1.000	0.2556	0.7944	0.3217	32.17
1.200	0.2170	0.8330	0.2605	31.26
1.400	0.1769	0.8731	0.2026	28.36
1.600	0.1352	0.9148	0.1478	23.65
1.800	0.0920	0.9580	0.0960	17.29
2.000	0.0472	1.0028	0.0471	9.42
2.200	0.0009	1.0491	0.0008	0.19
2.204	0.0000	1.0500	0.0000	0.00
0.676	0.3147	0.7353	0.4280	28.93
1.014	0.2529	0.7971	0.3173	32.17

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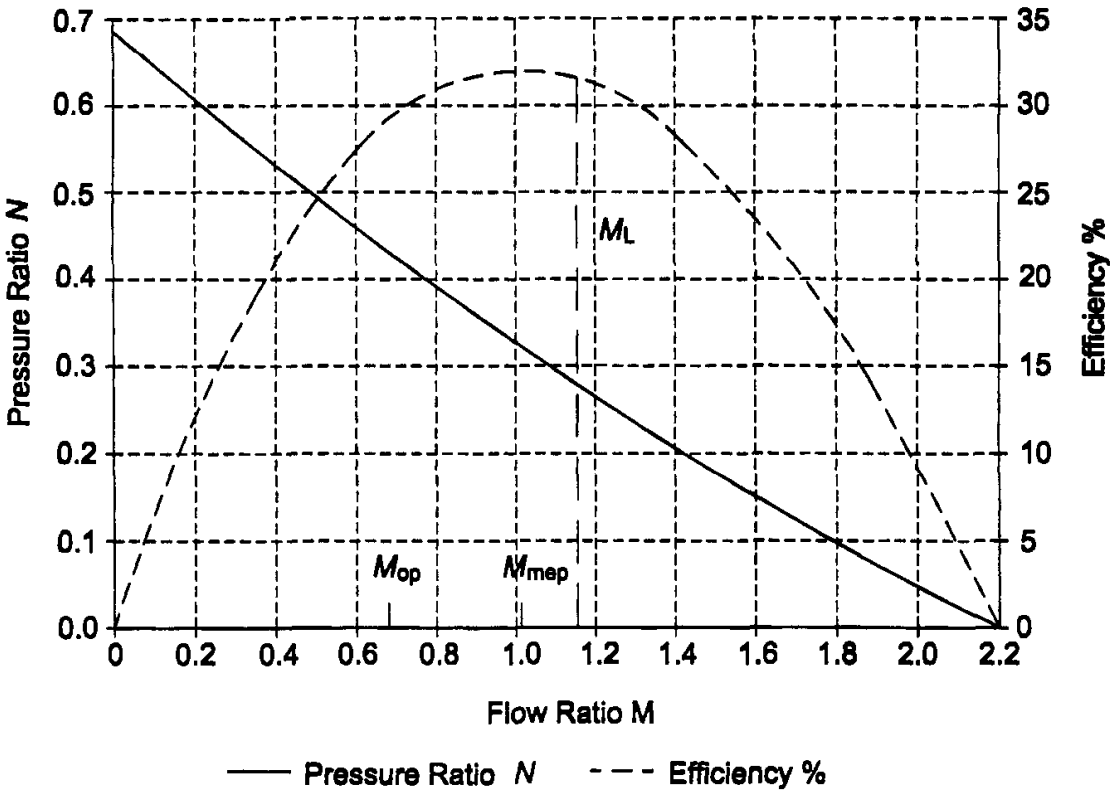


FIGURE 3 $K_n = .05, K_{td} = .20, K_{en} = 0$

It has been shown (see References 2, 3, and 4) that one-dimensional analyses successfully predict actual LJL pump performance. But theory-experiment agreement obtains only if the test-pump flow conforms with the model assumptions. Two conditions that will cause departure of measured N and η data from the theoretical curves are

- Cavitation, which occurs in the mixing throat
- Extension of the throat mixing process out into the diffuser

Operating Flow Ratio M_{op} In Figure 3, the flow ratio M_{op} is indicated on the left slope of the "parabolic" efficiency curve, which peaks at $M = M_{mep}$. The recommended location of the design operating point is to set $M_{op} = \frac{2}{3} M_{mep}$. Higher operating M ratios would provide slightly higher efficiencies, but at a greater risk of cavitation. (Cavitation as part of the design process is included in the following examples given.) Finding M_{mep} can be readily accomplished from Table 1 type data using spreadsheet successive approximations. Alternatively, Eqs. (17)–(18) can be differentiated and set equal to zero to find the peak efficiency M_{mep} value.

Cavitation LJL pumps may encounter cavitation, which occurs in the mixing throat. With reference to Figure 3, the LJL pump normally responds to a reduction in back pressure P_d (Q_1 and P_s constant) by producing a larger Q_2 secondary flow, and hence a larger M. Measured pressure ratios (N) and efficiencies (η) track along these theory-based characteristic curves as shown in Figure 3. But after the throat-inlet pressure (P_o) is reduced to the vapor pressure (P_v) of the secondary liquid, any further drop in the back pressure has no effect on the flow ratio, which stabilizes at $M = M_L$, the *cavitation-limited flow ratio.* Note the vertical dashed line in Figure 3: measured N and η values fall on this vertical line, under M_L operating conditions. In this manner, cavitating-pump performance departs radically from predicted/normal behavior.

Published studies (see References 1 and 8) have shown that NPSH-type correlations adequately explain and predict cavitation-limited flow phenomena. **Comparing the predicted M_L with the intended M_{op} is an essential step in designing a jet pump**

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installation. If $M_{op} < M_L$, the LJL pump can be expected to perform "on design;" that is, to follow the Eq. (17) $N(M)$ relation, with no cavitation. M_L can be predicted from the operating conditions as follows (see Reference 8):

$$M_L = c\sqrt{\frac{P_s - P_v}{\sigma Z}} \tag{20}$$

where σ is a cavitation coefficient.

Experiments, primarily with water and lubricating oils, have shown a σ range of 0.8–1.4 (see Reference 1). One investigator (see Reference 10) found that improving the nozzle exterior profile and related throat-inlet internal profile reduced the measured σ from 1.4 to 1.0, indicating a significant improvement in cavitation resistance. For design use, a conservative value of $\sigma = 1.35$ is recommended (see Reference 8).

Area Ratio b The nozzle-to-throat area ratio b is the only geometric parameter in these liquid-jet pump models, and including the LJL case, Eq. (17). Area ratio b is all-important, affecting pump efficiency, flow capacity, cavitation, and pressure characteristic $N(M)$. Comparisons of b-value series of pumps (see References 1–4) show that peak efficiency is highest for $b = .2$ to .3. This optimum occurs because efficiency of the jet pump reflects the two unavoidable losses: friction and the mixing loss. The latter (a maximum at zero M) *decreases* with rising M, whereas frictional losses *increase* with M. The sum of these losses is a minimum for pumps in the $b = .2 - .3$ range, each operating at or near the respective M_{mep} (see Reference 9). As indicated in the numerical Examples 1 and 2 further on, it is recommended that jet pump designs start—in the absence of other restraints—with $b = .25$. This initial area ratio can then later be adjusted, for example, to handle a cavitation problem. In specific cases the jet pump placement, or a desired flow ratio, may determine N or M, and hence b. Example 3 illustrates the situation.

Friction-Loss Coefficients Solutions of Eqs. (11)–(18) require use of appropriate K loss coefficients. The following sources are suggested:

1. Adopt published results from jet pump studies that have included measurements of K values (see References 3, 10, and 11).
2. Use published loss coefficients for the (four) components of the LJL pump; that is, converging nozzle, tube entry, short tube, and diffuser.
3. Measure Ks directly by running flow test of jet pumps or component parts. Full evaluation of the four Ks requires measurement of two flow rates and five static pressures, P_i, P_s, P_o, P_t, and P_d. Most studies, however, have omitted P_o and P_t pressure taps, and measured only three of these pressures: P_i, P_s, and P_d. This omission limits K evaluations based on pump tests to K_n and K_{td}. Fortunately, the theory works quite well under this limitation, largely because the throat and diffuser Ks are additive ($K_{td} = K_{th} + K_{di}$) in the $N(M)$ equations.

In regard to Item 1 above, Table 2 shows the range and recommended values for the Ks which appear in the theoretical model equations.

Table 2 recommendations assume that only P_i, P_s, and P_d are utilized; if this is the case, it is necessary to assume that $K_{en} = 0$, as indicated. A finite value of K_{en} in Eqs. (15)–(17)

TABLE 2 Recommended values for K friction-loss coefficients

	Range	Rec. Values	References
K_n	.04–1.0	.05	3, 10, 11
K_{en}	—	0	ditto
K_{td}	.17–.40	.20	ditto

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TABLE 3 Values of $M_{op} = \frac{2}{3}M_{mep}$, and N_{op}, for Figure 4

b	M_{mep}	M_{op}	N_{op}
0.05	4.8520	3.2347	0.0739
0.10	2.7120	1.8080	0.1509
0.15	1.8294	1.2196	0.2344
0.20	1.3330	0.8887	0.3264
0.25	1.0142	0.6760	0.4280
0.30	0.7913	0.5275	0.5406
0.40	0.5020	0.3347	0.8039
0.50	0.3270	0.2180	1.1210
0.60	0.2117	0.1411	1.4913

increases the slope of the theoretical $N(M)$ curve at low M values. At operating M values, however, this effect is negligible.

Finding b as a Function of N_{op} or M_{op} Figure 4 presents $N_{op}(b)$ and $M_{op}(b)$ for the recommended K friction coefficients of .05, 0, and .20. The Figure 4 curves are cross-plots of M_{op} values ($=\frac{2}{3}M_{mep}$) and associated N_{op} values. (To find M_{mep} value at b, Eqs. (17)–(18) spreadsheets of $\eta(M)$ and $N(M)$ were prepared for b values covering the range .05–.6; these numerical values are provided in Table 3.) Graphical representation of the LJL pump characteristics as shown in Figure 4 is useful in estimating b values when the jet pump application fixes N or M, and hence b. Example 3 illustrates the utility of Figure 4.

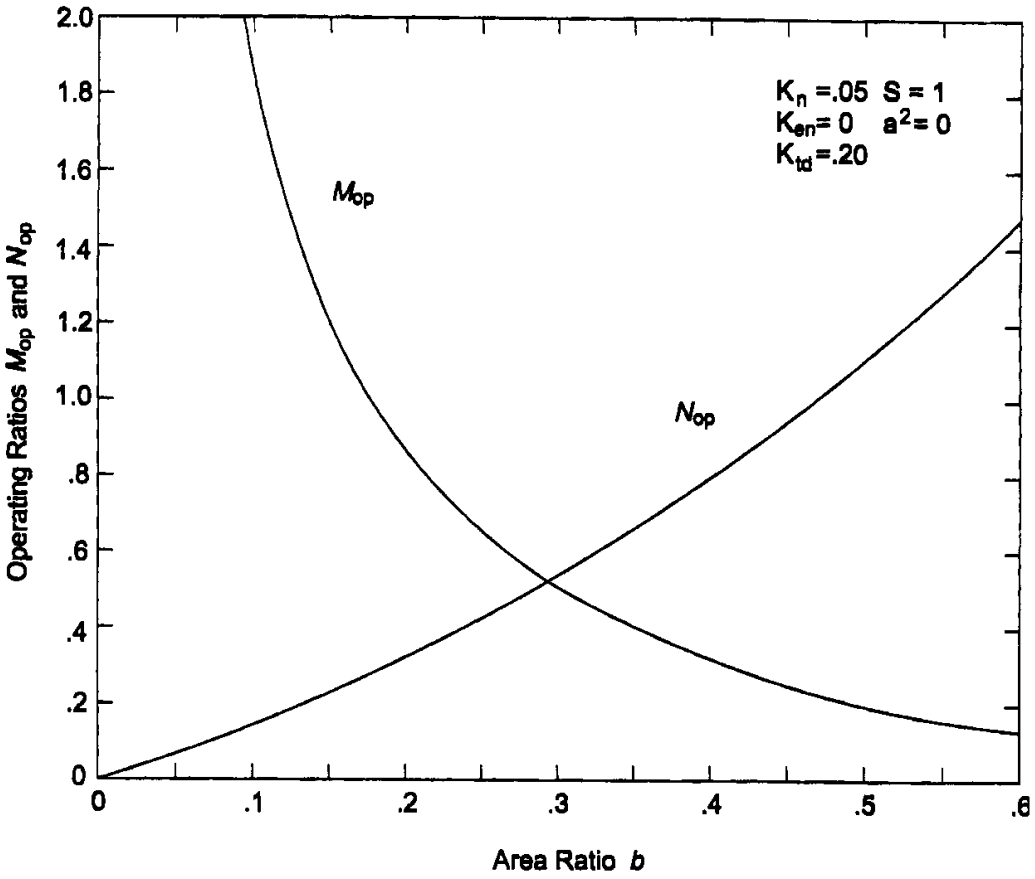


FIGURE 4 Response of LJL operating pressure ratio N_{op} and operating flow ratio M_{op} to jet pump design area ratio b

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Straight-Line N(M) Approximation Early investigators (see Reference 2) noted a nearly linear behavior of N versus M. A "straight-line" approximation for $N(M)$ based on the axis intercepts N_o and M_o would, of course, be useful. A true parabolic efficiency curve would result so $M_{mep} = \frac{1}{2} M_o$, and M_o is easily found. Unfortunately, the degree of this linearity varies with area ratio b. At about $b = .12$, the N curve is in fact a straight line. But as b is increased, the $N(M)$ curve is increasingly concave down so the $\eta = MN$ "parabolic" curve is increasingly skewed to the left. The converse is true at very small b values. Accordingly, the straight-line approximation is not recommended for use in designing LJL jet pumps. Instead, the Eqs. (17) and (18) theoretical model should be solved and charted for each b-value pump under consideration. Again, the recommended operating flow ratio $M_{op} = \frac{2}{3} M_{mep}$.

Longitudinal Dimensions of the LJL Jet Pump One-dimensional theory includes dimensions perpendicular to flow, but provides no guidance for longitudinal shapes/profiles. Two hardware dimensions of key importance must be taken from the literature or determined experimentally: nozzle-to-throat spacing "sp" and mixing throat length "L." Both are expressed in terms of throat diameters; sp/D_{th} and L/D_{th}. Recommended values are given as follows.

NOZZLE-THROAT SPACING Many experimental searches for optimum sp have been reported. Sanger and Vogel both found (see References 10 and 12) that maximum efficiency (slightly above 40% for Reference 10) is obtained with $sp/D_{th} = 0$, using nozzles with an external concave tapering, leading to a thin lip at the outlet. This configuration matches the theoretical model: The jet discharges to pressure P_o at the throat inlet, and thus the normal jet loss was eliminated. But a zero spacing promotes cavitation, even with a thin-lipped nozzle tip, and certainly with the rounded-nose exterior profile used in Reference 4. Sanger found that retracting nozzles a distance of about one diameter provided good cavitation resistance and at only a small loss in efficiency. Other investigators have found that performance is insensitive in the range of $sp/D_{th} = 0.5$–2 diameters, and that small spacings do promote cavitation. (see Reference 1). *It is recommended that LJL jet pumps be designed with* $sp/D_{th} = 1$.

MIXING-THROAT LENGTH The parallel-walled throat should be long enough to allow complete mixing, but throat lengths should be as short as possible to minimize frictional losses. L/D_{th} values as well as length (a venturi shape) ranging 1.0 to 10 have been reported for LJL pumps. Several factors affect optimum throat length:

- **a.** When sp/D_{th} is finite—true for most jet pumps—mixing takes place in the distance (sp plus part or all of L, the mixing-throat length); thus, sp and L are interrelated. Sanger (see Reference 10) found that optimum sp/D_{th} increased from 0 to 2.3, for pumps with $L/D_{th} = 7.5$ and 3.5 respectively.
- **b.** The primary-flow nozzle affects required L: Long tapered nozzles promote boundary-layer build-up producing jets that delay mixing, increasing required L. Multi-hole nozzles and swirl-inducing nozzles promote mixing and reduce required length L, but pump efficiency suffers because of increased nozzle-flow losses. (See "Primary-Flow Nozzle Design.")
- **c.** Pump area-ratio b can affect optimum throat length. Small-b pumps operate with high flow ratios and throat lengths of $L/D_{th} = 8$ were required. For pumps with larger b values, throat lengths of four diameters sufficed (see Reference 13).
- **d.** The gas compressor jet pump (LJG) requires longer throats, as high as 10–30 throat diameters (see Reference 7).

$L/D_{th} = 6$ *is recommended for general LJL design use.* Efficiency of the proposed pump may subsequently be improved by optimizing L (experimentally) for the given pump and duty.

PRIMARY-FLOW NOZZLE DESIGN A short-entry internally-convex profile similar to the ASME metering flow nozzle, is recommended for the LJL pumps. Avoid long conical nozzles.

Liquid-jet flow from a sharp-edged orifice mixes readily and is recommended for the LJG gas compressor (see Reference 7). The annular-nozzle liquid-jet pump has been investigated. In this configuration the secondary flow is axial, surrounded by the primary flow at the throat entrance. This arrangement is advantageous in pumping sticky secondary fluids because it prevents wall contact of the sticky fluid at the throat entry (see Reference 1). An obvious disadvantage is the increased nozzle frictional loss caused by flow of the primary fluid over the comparatively large surface area of the annular nozzle.

THROAT-INLET CONTOUR Many jet pumps reported in the early literature had long (small-convergence-angle) conical sections connecting the suction chamber to the throat, and usually including a sudden wall angle change at the throat entry. Later developments led to the short (large-convergence-angle) entry, well rounded at the throat. The long (small-angle) conical entry is wrong because its proximity to the nozzle exterior throttles the secondary flow and because it promotes cavitation. Secondly, a long approach section increases wall friction (reflected in a high K_{en} if it is measured). A short entry to the throat and a well-rounded profile connecting the suction chamber and throat is recommended.

Laboratory Flow Tests Performance testing of LJL pumps requires a facility with appropriate instrumentation, pumps, flow meters, and control valves. The flow rates Q_1 and Q_2, and at least three static pressures, P_i, P_s, and P_d, must be recorded at each test point. In addition, it is recommended that the throat section(s) contain static pressure taps for measurement of P_o and P_t. Jet pump test data will then permit measurement of K_{en}, K_{th}, and K_{di}. For on-design operation, these Ks vary little, if at all, with change in flow ratio M. Departures of $K(M)$ from nominal levels serve to reveal otherwise hidden problems. One example is that K_{th} will change if an increase in flow ratio causes mixing to extend/persist from the throat section into the diffuser (inadequate throat length). Another example is that a sudden rise in K_{di} may indicate diffuser-wall separation.

LIQUID-JET LIQUID (LJL) PUMP DESIGN EXAMPLES

Equations (1), (3), (5), and (7) will model jet pumps for LJL, LJG, and LJGL configurations. Equations (1), (11–19) apply specifically to the LJL jet pump. The three numerical examples provided here are for this widely used pump only. LJG and LJGL liquid-jet pumps are less common: please see References 5, 6, and 7.

Example 1 Design a jet pump to handle 50 gpm (11.36 m³/h) of water at 80°F (26.7°C) from a suction at 14.7 psia atmospheric pressure (101.325 kPa) to discharge at 40 psi (275.8kPa). Determine the required primary flow rate, jet nozzle pressure, and dimensions of the jet pump.

Solution: For best efficiency, select $b = .25$. From Table 2, adopt $K_n = 0.5$, $K_{td} = .20$, and $K_{en} = 0$. Eq. (17) then produces $N(M)$. A computer spreadsheet table for Eqs. (17) and (18) showing output values based on increments of M is recommended, for example, see Table 1. As shown by the bottom line in Table 1, the $M_{mep} = 1.104$ (found by successive approximations using the spreadsheet program). The operating flow ratio is $M_{op} = \frac{2}{3}M_{mep} = .676$. The spreadsheet program at this M_{op} produces the N_{op} (.428) and η(28.934%) values shown in Table 1, second line from the bottom. From Eq. 19 $(P_d - P_s)/P_i - P_s) = N/(N + 1)$, so and $P_i - P_s = (1.428/.4828)40 = 1.3346$ psi (920.212 kPa). Thus $P_i = 1.3346 + 14.7 = 148.16$ psia (1021.56kPa). Liquid jet pumps—with very few exceptions—operate with the nozzle tip withdrawn from the throat entry by one throat-diameter or more. The nozzle tip experiences a discharge pressure close to P_s, not P_o, and "jet loss" thus occurs. For this jet loss condition, Eq. (1) changes to $(P_i - P_s) = Z(1 + K_n)$. The jet velocity head $Z = 1.3346/1.05 = 1.2710$ psi (876.35kPa).

CAVITATION-LIMITED FLOW RATIO M_L The flow ratio M_L is now evaluated using Eq. (20) to answer the question: will $M_{op} = .676$ avoid pump cavitation? The area ratio $c = (1 - .25)/$

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.25 = 3. For the secondary fluid (water at 80°F, 26.7°C), P_v = .506 psia (3.49 kPa). The conservative value of σ = 1.35 is used in Eq. (20) as follows:

$$M_L = 3\sqrt{\frac{14.7 - .506}{1.35 \times 1.2710}} = .863$$

Because M_{op} (.676) < M_L (.863), this pump will not cavitate at the specified operating condition.

JET PUMP DIMENSIONS $Q_1 = Q_2/M_{op}$ = 50/.676 − 73.96 gpm (16.8 m³/h) = 73.96/(7.481 × 60) = .165 ft³/s. Longitudinal dimensions: sp/D_{th} = 1.0, thus sp = .937 in (23.81 mm). For L/D_{th} = 6, L = 6 × .937 = 5.62 in (142.75 mm). A diffuser with an included angle of 5° (conservative) and area ratio a = .224 would have length of approximately 12 in (304.8 mm). A shorter diffuser may be desirable, but kinetic-energy leaving losses will be higher.

EXAMPLE 1 USING OTHER AREA RATIOS An infinite number of different jet pumps can be designed to handle the Example 1 duty of pumping 50 gpm at 40 psi discharge pressure. The previous numerical example (b = .25) was repeated using b = .1, .4, and .6. Table 4 compares the results of these four b-ratio pumps. In each case the design is based on $M_{op} = \frac{2}{3}M_{mep}$; the assumed K values and calculation procedure are similar for the four pumps.

The expression CR% in Table 4 indicates the $(M_L - M_{op})/M_{op}$ % separation of operating flow ratio M_{op} and the limiting flow ratio M_L; the larger this number, the better. For this constant-duty example (Q_2 = 50 gpm and P_d = 40 psi), Table 4 shows that cavitation resistance can be improved by using a larger b ratio, i.e., the $(M_L - M_{op})/M_{op}$ % figure increases with b. Note that at the smallest b value (.1), the cavitation value (sixth column) is CR = −5.3%: this indicates that $M_{op} > M_L$, and this pump *would* encounter cavitation: b = .1 should not be used at these M_{op} and σ values. A redesign of the pump suction chamber (nozzle-external profile and throat-inlet profile) leading to an improvement in cavitation coefficient σ could render the b = .1 pump usable (the value σ = 1.35 used previously is conservative). For example, an improvement to σ = 1.0 would raise M_L to 2.0. With this change, a b = .1 pump could be used, cavitation-free.

Table 4 shows two other important facts: (1) The b = .25 pump provides the highest efficiency, even though all four are designed to operate at $\frac{2}{3}M_{mep}$, i.e., each at a similar position on the efficiency curve specific to that pump. (2) Small b-value pumps operate at large nozzle pressure-drops ($(P_i - P_s)$) and with small Q_1 primary-values. Conversely, high b-value pumps operate with small nozzle pressure-drops, but use a large flow rate Q_1. The pumps have in common similar expenditures of energy (Q_1 × pressure change, and allowing for efficiency differences) to handle the duty, which is the same in all four cases. In some design problems the nozzle-pressure—or possibly the primary-fluid flow volume—might outweigh the importance of mechanical efficiency. Table 4 shows how an adjustment in b might be used to achieve improved cavitation, albeit with efficiency sacrifices.

REDUCING THE FLOW RATIO M_{OP} TO COPE WITH CAVITATION For a given b-value LJL jet pump, a reduction in M_{op} offers a way to improve cavitation resistance. The b = .1 pump used

TABLE 4 Four b-value jet pumps for duty of 50 gpm (11.35 m³/h) at P_d = 40 lb/in² (275 kPa)

b	M_{op}	N_{op}	η%	M_L	CR%	$(P_i - P_s)$, lb/in²	Q_1, gpm	D_n, in
0.10	1.8080	0.1509	27.28	1.7122	−5.30	305.08	27.65	0.233
0.25	0.6760	0.4280	28.93	0.8628	27.65	133.46	73.96	0.469
0.40	0.3347	0.8039	26.91	0.5261	57.19	89.76	149.39	0.736
0.60	0.1411	1.4913	21.04	0.2711	92.13	66.82	354.36	1.219

[Note: for SI conversion: in × 25.4 = mm]

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TABLE 5 Reducing M_{op} improves CR% cavitation resistance (b = .1)

M_{op}	N_{op}	η%	M_L	CR%	$(P_i - P_s)$, lb/in²	Q_1, gpm	D_n, in
1.808	0.1509	27.28	1.7122	−5.30	305.08	27.65	0.233
1.500	0.1631	24.46	1.7704	18.01	285.08	33.33	0.260
1.000	0.1823	18.23	1.8566	85.67	259.45	50.00	0.326
0.500	0.2006	10.03	1.9326	286.56	239.36	100.00	0.471

[For SI conversions: lb/in² × 6.89 = kPa; gpm × 0.277 = m³/h; in × 25.4 = mm]

in this example is examined for the effects of M_{op} reduction, in Table 5. The conditions for all four pumps in Table 5 are b = .1, and again, K_n = .05, K_{td} = .2, K_{en} = 0. The Table 5 top line is M_{op} = 1.808, followed by three .5 decrements of M_{op}. The corresponding N_{op} and η_{op} values are found from successive approximations using the spreadsheet based on Eqs. (17) and (18). The M_L values are calculated from Eq. (20). The CR% cavitation merit figure (5th column) is the same as used in Table 4. Table 5 shows that while a b = .1 pump at M_{op} = 1.808 flow ratio will cavitate, the three lower M_{op} cases all exhibit increased CR% cavitation resistance. In Table 5, M_L (4th column) changes little with M_{op}; the cavitation improvements largely result from the decreases in M_{op}. The primary flow rate Q_1 increases markedly, and the nozzle pressure decreases somewhat as M_{op} is reduced. Compared with adjustment in b, cavitation avoidance by reducing M_{op} has obvious disadvantages: Efficiency is severely reduced, and the pump size (see D_n, 8th column) must be increased to handle the larger flow Q_1.

In summary, every jet pump design should be evaluated for cavitation resistance. The CR% index used here ($(M_L - M_{op})/M_{op}$ %) is suggested if several options are to be compared. Otherwise, the simpler test $M_{op} < M_L$, is adequate. Cavitation resistance for a given pump duty can be improved by increasing the area ratio b, or by reducing M_{op} of a particular pump, or by reducing the cavitation coefficient σ by improved internal design.

Example 2 Design a jet pump to lift 80°F (27.67°C) water to 20 ft (6.1m) to discharge at an atmospheric pressure of 14.7 psia or 34 ft of water (10.3 m). A mechanical pump provides the primary flow stream; fluid power available at the jet pump is 10 kW. Find the secondary flow Q_2, and pump dimensions.

Solution: Assume the same K values used in Example 1. Again select b = .25, for maximum efficiency. Eq. 17 (via spreadsheet program) produces M_{mep} = 1.014 and $M_{op} = \frac{2}{3}(1.014)$ = 6.76, where N_{op} = .428 and η = 28.93%. P_s = (34 − 20) = 14 ft water abs. (4.261 m abs.) at the suction chamber. And P_s = 14 × 1.934 × 32.174/144 = 6.05 psia (41.72 kPa). $(P_d - P_s)$ = (14.7 − 6.05) = 8.65 psi (59.64 kPa). From Eq. 19, $(P_i - P_s) = (P_d - P_s)(N + 1)/N$, so that $(P_i - P_s)$ = (14.7 − 6.05)(1.428/.428) = 28.86 psi (199 kPa). Pi = 28.86 + 6.05 = 34.91 psia (40.7 kPa). $Z = (P_i - P_s)/(1 + K_n)$ = 28.86/1.05 = 27.486 psi (189.52 kPa).

Cavitation-Limited Flow Ratio M_L Limiting flow ratio (Eq. 20)

$$M_L = 3\sqrt{\frac{6.05 - .506}{1.35 \times 27.486}} = 1.16$$

Because M_{op} (.676) < M_L (1.16), the proposed LJL jet pump will not cavitate.

JET PUMP DIMENSIONS Power in = $Q_1(P_i - P_d)$; Q_1 = power in/$(P_i - P_d)$, where $(P_i - P_d)$ = (34.91 − 14.7) = 20.21 psi. With 10 kW treated here as the net "fluid" power available at the nozzle. (In Figure 2 the jet and mechanical pumps are at the same elevation)

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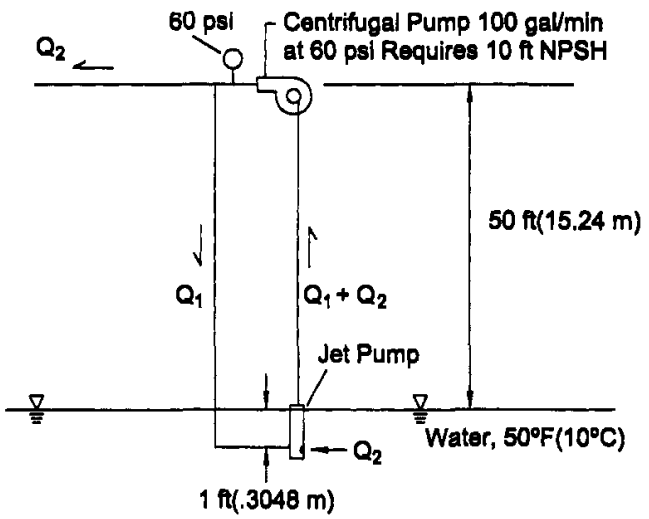


FIGURE 5 Jet and centrifugal pump system, for Example 3

$$Q_1 = \frac{10 \times 7.481 \times 60}{1.356 \times 10^{-3} \times 144 \times 20.21} = 1137.42 \text{ gal/min } (258.6 \text{ m}^3/\text{h})$$

(conversion factors 7.481 gal/ft^3 and 1.356 watts/ft-lb/s were used previously)

Secondary flow $Q_2 = 1137.42 \times .676 = 768.9$ gpm (174.6 m^3/h).

Nozzle area:

$$A_n = \frac{Q_1}{V_n}, \text{ where } V_n = \sqrt{2 \times 27.476 \times \frac{144}{1.934}} = 64 \text{ ft/s } (19.51 \text{ m/s})$$

$$A_n = (768.9 \times 144)/(64 \times 7.481 \times 60) = 3.85 \text{ in}^2 (24.84 \text{ cm}^2)$$

Nozzle diameter:

$$D_n = \sqrt{3.85 \times \frac{4}{3.1416}} = 2.22 \text{ in } (56.39 \text{ mm})$$

Example 3 As shown in Figure 5, a surface-mounted centrifugal pump requires a NPSH of 10 ft (3 m) of water, and the centrifugal-pump discharge pressure is 60 psi (414 kPa) at 100 gpm (22.7 m^3/h). Water at 50°F (10°C) is to be lifted 50 ft (15.2 m). Design the jet pump including dimensions, and find the primary and secondary flow rates.

Solution: The jet pump's placement and centrifugal-pump discharge and inlet pressures determine all three jet-pump operating pressures, and hence the pressure characteristic N.

$$P_i = 60\,psi + (50 + 1) \times 1.94 \times 32.17/144 = 82.1\,psi\ (566.1\,kPa)$$

$$P_s = 1 \times 1.94 \times 32.17/144 = .433 \text{ psi } (299\text{kPa}).\ P_d = (50 + 10) \times$$

$$1.94 \times 32.17/144 = 26 \text{ psi } (179.27 \text{ kPa}).\ N_{op} = (P_d - P_s)/(P_i - P_d) = .456.$$

To estimate the b level which will operate at this N, enter Figure 4 with $N_{op} = .46$: the corresponding $b = .27$, and $M_{op} = .62$. Estimate b values should be confirmed—and adjusted if necessary—using a spreadsheet solution based on Eqs (17) and (18). In this case the

spreadsheet results are $N_{op} = .461$, $b = .265$, $M_{op} = .626$, and $\eta = 28.8\%$. The cavitation check and calculation of dimensions follow next. *Note:* This solution is a first estimate because piping frictional losses have not been included in the calculations. Second and third—if necessary—calculations must include the pressure losses in the piping connecting the two pumps.

CAVITATION LIMITED FLOW RATIO M_L Jet velocity head $Z = (P_i - P_s)/(1 + K_n) = (82.1 - .433)/(1 + .05) = 77.78$ psi (536.29kPa).

For water at 50°F (10°C), $P_v = .178$ psia (1.23 kPa)

$$M_L = c\sqrt{\frac{P_s - P_v}{\sigma} \times Z} = 2.774\sqrt{\frac{15.133 - .178}{1.35} \times 77.78} = 1.047$$

Because M_{op} (.626) < M_L (1.047), the jet pump will not cavitate at the operating point.

JET PUMP DIMENSIONS $Q_1 + Q_2 = 100$ gpm (19.71 m³/h), and $Q_2/Q_1 = .626$, so $Q_1 = 61.5$ and $Q_2 = 38.5$ gpm (12.12 and 7.59 m³/h). $A_n = Q_1/V_n$.

Jet velocity:

$$V_n = \sqrt{\frac{2Z}{\rho_1}} = \sqrt{\frac{2 \times 77.8}{1.94}} = 107.46 \text{ ft/s (32.75 m/s).}$$

$$Q_1 = 61.5/7.481 \times 60 = .137 \text{ ft}^3\text{/s(13.97 m}^3\text{/h)}$$

$$\textit{Nozzle area } A_n = (.137/107.46)144 = 1.87 \text{ in}^2\text{(1.187 cm}^2\text{).}$$

Nozzle diameter:

$$D_n = \sqrt{\frac{4 \times A_n}{\pi}} = \sqrt{\frac{4(.184)}{3.1416}} = .483 \text{ in (12.27 mm).}$$

Throat diameter:

$$D_{th} = \frac{D_n}{\sqrt{b}} = \frac{.483}{.515} = .938 \text{ in (23.83 mm)}$$

Nozzle-throat spacing recommended: $sp/D_{th} = 1$, hence sp = .938 in (23.83 mm)

Mixing-throat length recommended: $L/D_{th} = 6$, hence $L = 6$ (.938) = 5.63 in (143 mm)

Centrifugal-Jet-Pump Interactions Oil- and water-well pumping are major applications of LJL pumps. Compact jet pumps, lowered down the bore hole near to or below the liquid surface, are powered by primary flow supplied from a mechanical pump located above, at the earth's surface. The jet pump pressurizes the surface pump's suction preventing cavitation. The design of a combination pump system normally requires designing for operation at the intersection of the two pumps' operating curves (References 17, 18, and 19) and of course inclusion of frictional losses (neglected in Example 3) in the associated piping. One operating point (100 gpm at 60 psi) in Example 3 represented the centrifugal pump. The *60* psi discharge pressure of the centrifugal pump and the vertical placement fixed the jet pump's N_{op} operating point. This in turn determined the jet pump's b value. Consider now a change in the centrifugal pump to one providing 100 gpm at 100 psi (22.7 m³/h at 6.89 kPa): the jet pump's N_{op} (and b) would be reduced, and M_{op} raised. Details are omitted here, but the resultant area ratio would be about $b = .167$, which would provide $M_{op} = 1.09$, a considerable increase over $M_{op} = .626$, as found in Example 3.

SIGNIFICANCE OF ON-DESIGN OPERATING CONDITIONS

With reference to Figure 3, assume that the pump is operating on-design at $M_{op} = \frac{2}{3}M_{mep}$; that is, on the left branch of the efficiency "parabola." Primary flow rate Q_1 and suction pressure P_s are constant. If the back pressure P_d is then *reduced,* the secondary flow rate Q_2 (and hence M_{op}) will rise and pressure ratio N_{op} will fall. The new operating points will move to the right, along the characteristic curves for N and η. This increased-flow response to lowered P_d will end when cavitation-limited flow ratio M_L is reached, assuming the NPSH is low enough to allow cavitation. Any further reduction of P_d will cause the operating points to leave the N and η theoretical curves on paths vertically downward as shown by the dashed line in Figure 3. Consider next the same initial condition (on design: $M_{op} = \frac{2}{3}M_{mep}$) followed by an *increase* in jet-pump discharge pressure P_d. In response, the secondary flow Q_2 (and hence M) will decrease, the N_{op} and η_{op} operating points moving left along the pump's N and η characteristic curves. Ultimately, increasing the back-pressure will result in reaching the $M = 0$ point, where $N = N_o$. (See Table 1, showing $N = N_o = .680$ at $M = 0$.)

Mechanical efficiency is important in most jet pump installations. Two low-efficiency situations that should obviously be avoided are

- Throttling the pump discharge pressure (to raise the P_d to the design point) wastes energy. Design P_d should match the pressure of the system into which the pump is to discharge.
- Operating the LJL pump with a discharge pressure P_d less than the design P_d (same Q_1 and P_s) will yield more Q_2 flow ($M > M_{op}$) and slightly higher efficiency. But such an increase in M may reach the cavitation point, that is, where $M = M_L$, causing a marked departure of actual/measured operating points downward from the N and η characteristic curves.

OTHER JET PUMP APPLICATIONS

Virtually any fluid can be pumped by the liquid-jet pump. Suspensions of solids in a liquid can be handled with the LJL analysis given by incorporating the "S" density-ratio term. In extreme cases, slurry pumping may also be affected by slip velocity and solids concentration (see References 1 and 15).

Steam jet pumps are found in the earliest literature (see Reference 1). Analyses have usually combined one-dimensional treatment with shock-wave compressibility phenomena. Theory has not contributed to the longitudinal dimensions problem, as is the case with liquid jet pumps. Advances in applying two- and three-dimensional flow theory have contributed to understanding low Mach-number mixing length requirements. Air/air jet pump theory and performance have been compared (see Reference 16). Analysis of high-velocity compressible-flow gas-jet pumps has been less successful.

Design of a two-pump combination system (illustrated by Examples 2 and 3) required system operation at an intersection of the mechanical- and jet-pump characteristic curves, selected so both pumps operate at or near their maximum efficiency points (see References 17, 18, and 19). And the system design must of course include consideration of the pressure losses that will occur in the piping connecting the two pumps.

REFERENCES

1. Bonnington, S. T., and King, A. L. "Jet Pumps and Ejectors: A State of the Art Review and Bibliography." Published by BHRA Fluid Engineering, Cranfield, Bedfordshire MK43 OAJ, United Kingdom, 1976.
2. Gosline, J. E., and O'Brien, M. P. "The Water Jet Pump." *University of California Publications in Engineering*, v. 3, pp. 167–190, 1934.

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3. Cunningham, R. G. "Jet Pump Theory and Performance with Fluids of High Viscosity." *Trans. ASME,* v. 79, pp. 1807–1820, 1957.
4. Mueller, N. H. G. "Water Jet Pump." *Proceedings ASCE, Journal of the Hydraulics Division,* v. 90, pp. 83–113, 1964.
5. Cunningham, R. G. "Liquid Jet Pumps for Two-Phase Flows." *Trans. ASME, Journal of Fluids Engineering,* v. 117, pp. 309–316, 1995.
6. Cunningham, R. G. "Gas Compression with the Liquid Jet Pump." *Trans. ASME, Journal of Fluids Engineering*, v. 6, pp. 203–315, 1974.
7. Cunningham, R. G., and Dopkin, R. J. "Jet Breakup and Mixing Throat Lengths for the Liquid Jet Gas Pump." *Trans. ASME, Journal of Fluids Engineering*, v. 94, pp. 216–226, 1974.
8. Cunningham, R. G., Hansen, A. G., and Na, T. Y. "Jet Pump Cavitation." *Trans. ASME, Journal of Basic Engineering*, v. 92, pp. 483–494, 1970.
9. Cunningham, R. G. "Liquid Jet Pump Modeling: Effects of Axial Dimensions on Theory-Experiment Agreement." *Proceedings: Second Symposium on Jet Pumps and Ejectors,* BHRA Fluid Engineering, Cranfield, Bedfordshire MK43 OAJ, United Kingdom, 1975.
10. Sanger, N. L. "An Experimental Investigation of Several Low-Area-Ratio Water Jet Pumps." *Trans. ASME, Journal of Basic Engineering*, v. 92, pp. 11–20, 1970.
11. Na, T. Y. "Performance of Liquid Jet Pumps at Elevated Temperatures." *Proceedings: Symposium on Jet Pumps and Ejectors,* BHRA Fluid Engineering, Cranfield, Bedfordshire MK43 OAJ, United Kingdom, 1972.
12. Vogel, R. "Theoretical and Experimental Investigations on Jet Devices." *Maschinenbautechnik,* v. 5, pp. 619–637, 1956.
13. Schulz, F., and Fasol, K. H. *Wasserstrahlpumpen zur Forderung von Flussigkeiten.* Published by Springer-Verlag, Vienna, 73 pages, 1958.
14. Hoggarth, M. L. "The Design and Performance of High Pressure Injectors as Gas Jet Boosters." *Proceedings: I. Mech. E.,* v. 185, pp. 755–766, 1970–71.
15. Bonnington, S. T. *Jet Pumps.* (Handling solid-liquid mixtures), SP 529, BHRA Fluid Engineering, Cranfield, Bedfordshire MK43 OAJ, United Kingdom, 1956.
16. Razinsky, E., and Brighton, J. A. "A Theoretical Model for Nonseparated Mixing of A Confined Jet." *Trans. ASME Series D,* v. 94, pp. 551–558, 1972.
17. Hansen, A. G., and Na, T. Y. "Optimization Jet Pump Systems." ASME Paper 66-FE 4, April 1966.
18. Radha Kirishna, H. C., and Kumaraswamy, S. "Some Investigations on the Combination Performance of Jet-Centrifugal Pump." *Proceedings: Second Symposium on Jet Pumps and Ejectors and Gas Lift Techniques,* BHRA Fluid Engineering, Cranfield, Bedfordshire MK43 OAJ, United Kingdom. Paper B-1, March 1975.
19. Radha Kirishna, H. C., and Kumaraswamy, S. "Matching the Performance of Jet and Centrifugal Pumps." *Proceedings: Second Symposium on Jet Pumps and Ejectors and Gas Lift Techniques,* BHRA Fluid Engineering, Cambridge, UK. Paper B-3, March 1975.

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SECTION 7.2
JET PUMP APPLICATIONS

ALEX M. JUMPETER

This section contains extensive design and application experience for a variety of jet pump configurations. Although this section concentrates on eductors (termed LJL jet pumps in Section 7.1), experience with other motive (primary) and secondary fluids is also included. The theoretical developments of Section 7.1 are the basis for what is presented here, the dimensional design ratios being generally within the ranges mentioned therein. Therefore, the only theory in this section is the empiricism that is utilized in the examples and applications presented. Refer to Section 7.1 for further explanation.

DEFINITION OF TERMS

A definition of standard ejector terminology is as follows:

Ejector	General name used to describe all types of jet pumps that discharge at a pressure intermediate between motive and suction pressures.
Eductor	A liquid jet pump using a liquid as motive fluid.
Injector	A particular type of jet pump that uses a condensable gas to entrain a liquid and discharge against a pressure higher than either motive or suction pressure; principally, a boiler injector.
Jet Compressor	A gas jet pump used to boost pressure of gases.
Siphon	A liquid jet pump utilizing a condensable vapor, normally steam, as the motive fluid.

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EDUCTORS

Design The elements of an eductor design are shown in Figure 1. Quantities involved are defined as follows:

P_1 = static pressure upstream, lb/ft^2 (N/m^2)
P_s = static pressure at suction (nozzle tip), lb/ft^2 (N/m^2)
V = velocity, ft/s (m/s)
γ_1 = specific weight (force) of motive fluid, lb/ft^3 (N/m^3)

$$\frac{P_1 - P_s}{\gamma_1} = \text{operating head, ft (m)}$$

P_s = static pressure at suction, lb/ft^2 (N/m^2)
P_2 = static pressure at discharge, lb/ft^2 (N/m^2)
γ_2 = specific weight (force) of mixed fluids, lb/ft^3 (N/m^3)

$$\frac{P_2 - P_s}{\gamma_2} = \text{discharge head, ft (m)}$$

The head ratio R_H is defined as the ratio of the operating head to the discharge head:

$$R_H = \frac{(P_1 - P_s)/\gamma_1}{(P_2 - P_s)/\gamma_2} = \frac{(P_1 - P_s)\gamma_2}{(P_2 - P_s)\gamma_1} \tag{1}$$

Because ratios are involved, it is convenient to replace specific weight with specific gravity:

$$R_H = \frac{(P_1 - P_s)(\text{sp. gr.}_2)}{(P_2 - P_s)(\text{sp. gr.}_1)} \tag{2}$$

When the suction and motive fluids are the same, no gravity correction is required and Eq. 2 becomes:

$$R_H = \frac{H_1 - H_s}{H_2 - H_s} \tag{3}$$

where $H_1 - H_s$ = operating head, ft (m)
$H_2 - H_s$ = discharge head, ft (m)

Entrainment relates the mass (flow rate) of motive fluid and suction fluid:

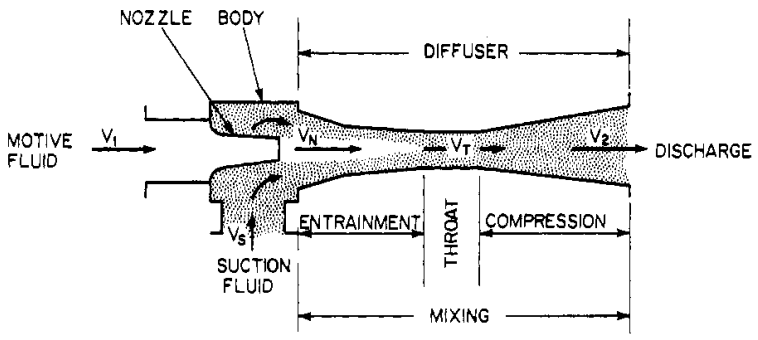


FIGURE 1 Eductor elements and terminology

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M_1 = mass of motive (primary) fluid, slugs (kg)
M_s = mass of suction (secondary) fluid, slugs (kg)

$$R_w = \frac{M_s}{M_1} = \text{the weight operating ratio} \tag{4}$$

The volume ratio R_q is then simply

$$\frac{Q_s}{Q_1} = R_w \frac{\text{sp. gr.}_1}{\text{sp. gr.}_2} \tag{5}$$

where Q_s = suction (secondary) flow in volumetric units
Q_1 = motive (primary) flow in volumetric units

The performance of eductors is expressed here in terms of R_H and R_w, utilizing an empirical relationship that involves an efficiency factor ϵ as follows:

$$R_w = \epsilon\sqrt{R_H} - 1 \tag{6}$$

This equation is used to calculate the motive quantity or pressure from the operating parameters. This nozzle and diffuser diameters are calculated from the equation $Q = wAV$, using suitable nozzle and diffuser entrance coefficients. The principal problems in design concern the size and proportions of the mixing chamber, the distance between nozzle and diffuser, and the length of the diffuser. Eductor designs are based on theory and empirical constants for length and shape. The most efficient units are developed from calculated designs that are then further modified by prototype testing.

Figure 2 shows this factor plotted against NPSH (net positive suction head) for a single-nozzle and annular-nozzle eductor. In an annular-nozzle eductor, the motive fluid is introduced around the periphery of the suction fluid, either by a ring of nozzles (Figure 15) or by an annulus created between the inner wall of the diffuser and the outer wall of the suction nozzle (Figure 14). The NPSH is the head available at the centerline of the eductor to move and accelerate suction fluid entering the eductor mixing chamber. NPSH is the total head in feet (meters) of fluid flowing and is defined as atmospheric pressure minus suction pressure minus vapor pressure of suction or motive fluid, whichever is higher.

Increased viscosity of motive or suction fluid increases the frictional and momentum losses and therefore reduces the efficiency factor of Figure 2. Below 20 cP, the effect is minimal (approximately 5% lowering of ϵ). Above this value, the loss of performance is more noticeable and empirical data or pilot testing is used to determine sizing parameters.

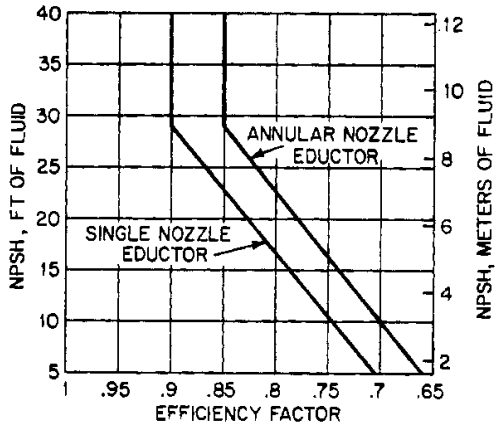


FIGURE 2 NPSH versus efficiency factor (Schutte and Koerting)

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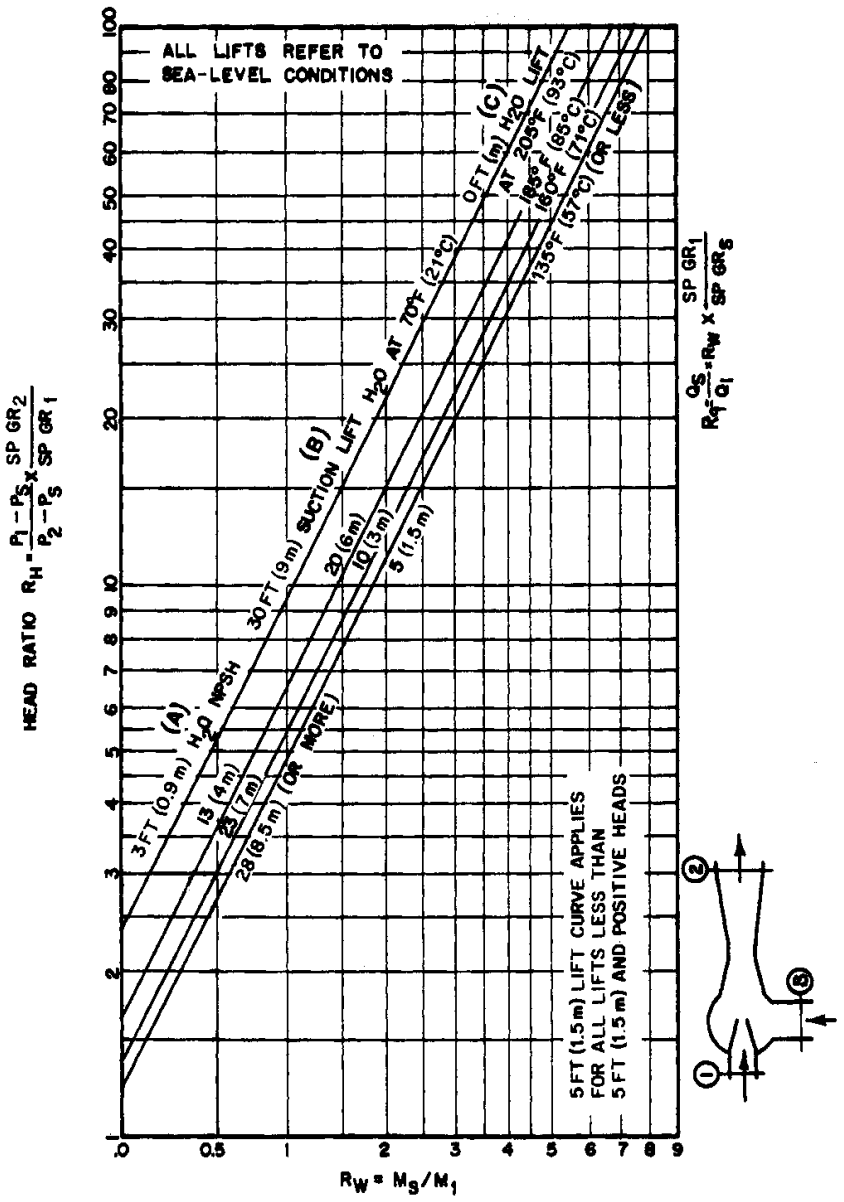


FIGURE 3 Estimating operating ratios for liquid jet eductors. An eductor can be designed for only one head point (Schutte and Koerting)

Figure 3 shows the operating ratio R_w versus the head ratio R_H for various lift conditions. The efficiency factor has been incorporated into this curve.

The final size of the eductor is determined by the discharge line and is based on normal pipeline velocities, which are usually 3 to 10 ft/s (0.9 to 3 m/s). Figure 4a and 4b are used for estimating eductor size. To illustrate the use of Figures 3 and 4, consider the following example.

EXAMPLE 1 It is desired to remove 100 gpm (22.7 m³/h) of water at 100°F (38°C) from a pit 20 ft (6.1 m) deep. Discharge pressure is 10 lb/in² (0.69 bar*) gage. Motive water is available at 60 lb/in² (4.1 bar) gage and 80°F (26.6°C). The eductor is to be located above the pit. Find the eductor size and motive water quantity required.

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

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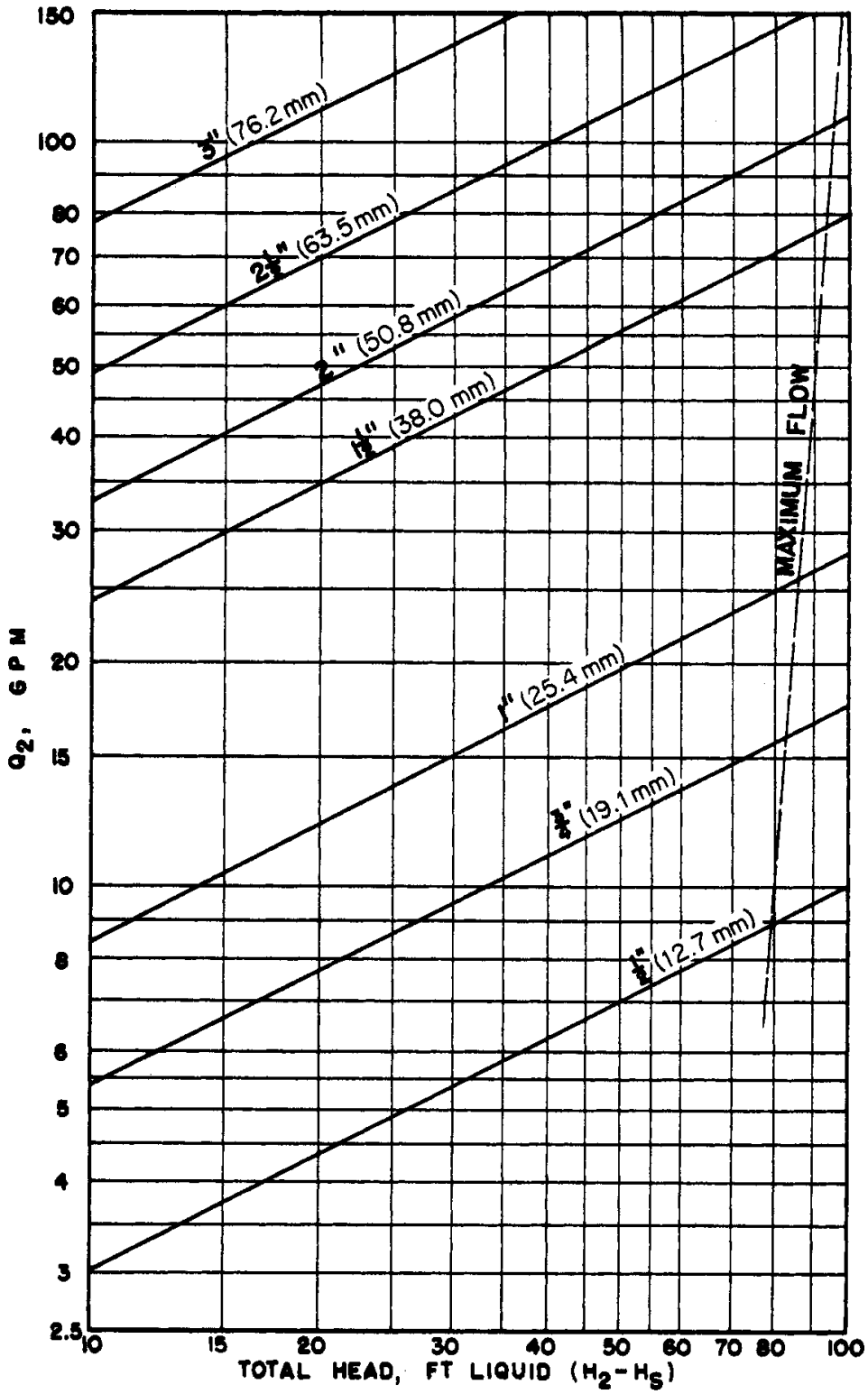
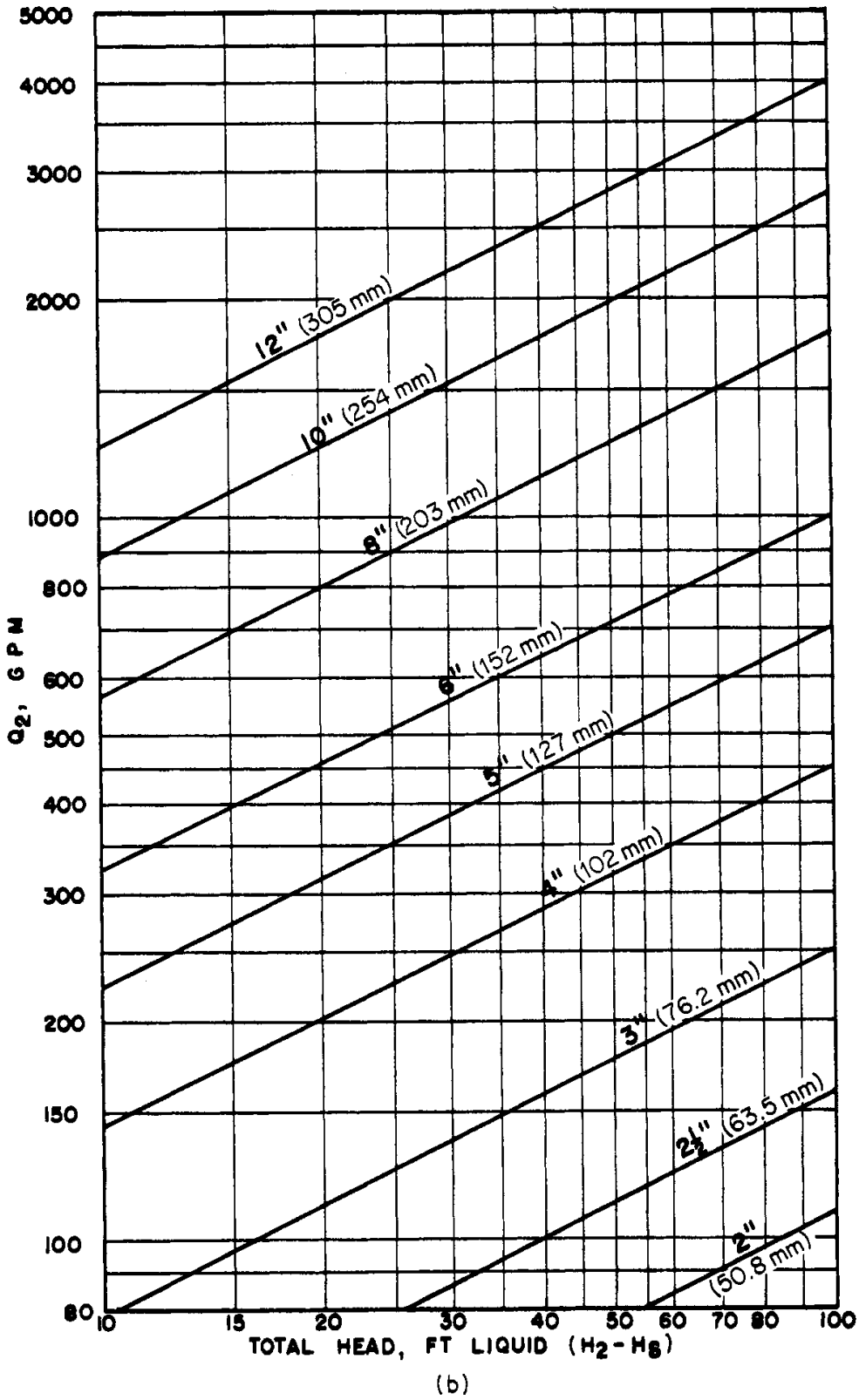


FIGURE 4A Sizing curve (gpm × 0.227 = m³/h; ft × 0.3048 = m)

Solution To use Figure 3, it is necessary to determine the NPSH and the head ratio R_H. The centerline of the eductor is chosen as the datum plane, and NPSH is taken to be atmospheric pressure minus suction lift minus vapor pressure at 100°F (38°C):

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(b)

FIGURE 4B Sizing curve (continued)

in USCS units $\text{NPSH} = 34 \text{ ft} - 20 \text{ ft} - 1.933 \text{ inHg} \left(\frac{13.6}{12}\right) = 11.81 \text{ ft}$

in SI units $\text{NPSH} = 10.36 \text{ m} - 6.1 \text{ m} - 49 \text{ mmHg} \left(\frac{13.6}{1000}\right) = 3.59 \text{ m}$

Because motive and suction are the same fluid, it is convenient to work in feet (meters) rather than pounds per square inch (bar), and

$P_1 = 60 \text{ lb/in}^2 \text{ gage} = 138.6 \text{ ft } H_2O \text{ (4.1 bar = 42 m)}$

$P_2 = 10 \text{ lb/in}^2 \text{ gage} = 23.1 \text{ ft } H_2O \text{ (0.69 bar = 7 m)}$

$P_s = -20 \text{ ft } (-6.1 \text{ m})$

Then

in USCS units $R_H = \dfrac{138.6 - (-20)}{23.1 - (-20)} = \dfrac{158.6}{43.1} = 3.68$

in SI units $R_H = \dfrac{42 - (-6.1)}{7 - (-6.1)} = 3.68$

Enter Figure 3 at $R_H = 3.68$ and NPSH = 11.81 (3.59 m); read $R_w = 0.48$. Because there is no gravity correction,

in USCS units $R_w = R_q = \dfrac{0.48 \text{ gal suction}}{\text{gal motive}}$

in SI units $R_w = R_q = \dfrac{0.48 \text{ m}^3 \text{ suction}}{\text{m}^3 \text{ motive}}$

The same result can be obtained by using the efficiency factor from Figure 2. Then R_w is $0.77\sqrt{R_H - 1} = 0.48$ and the required motive fluid is

in USCS units $\dfrac{100 \text{ gpm suction}}{0.48} = 208 \text{ gpm at } 60 \text{ lb/in}^2 \text{ gage}$

in SI units $\dfrac{22.7 \text{ m}^3/\text{h}}{0.48} = 47.3 \text{ m}^3/\text{h at } 4.1 \text{ bar gage}$

Discharge flow is

in USCS units $208 + 100 = 308 \text{ gpm}$

in SI units $47.3 + 22.7 = 70.0 \text{ m}^3/\text{h}$

The size is obtained from Figure 4. Enter Figure 4b at $Q_2 = 308$ gpm (70 m³/h) and discharge head $(H_2 - H_3) = 23.1 - (-20) = 43.1$ ft [7 − (−6.1) = 13.1 m]; read eductor size of 4 in (102 mm) based on the discharge connection.

NOTE: If there were any appreciable length of run on the discharge line, it would be necessary to calculate the pressure drop in this line and recalculate the eductor size after adding the line loss to the discharge head required. Frictional losses on the suction side must also be included. In the example chosen, however, 100 gpm (22.7 m³/h) in a 4-in (102-mm) suction line 20 ft (6.1 m) long will have negligible frictional loss, less than 0.25 ft (0.08 m) H_2O.

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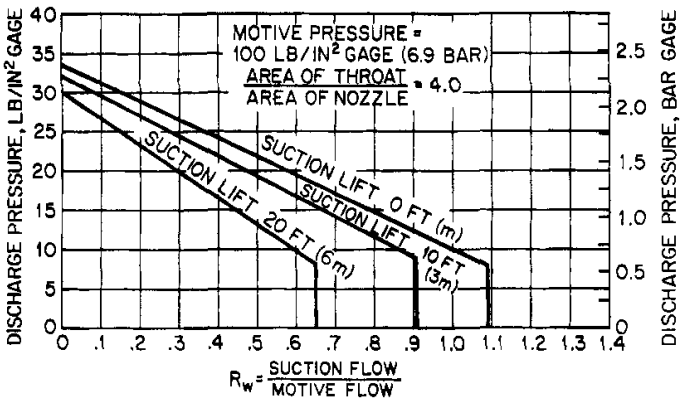


FIGURE 5 Characteristic performance of an eductor

Performance Characteristics Figure 5 illustrates the performance characteristics of eductors. Note the sharp break in flow rate below the design point. For this reason, all eductors are not designed for a peak efficiency. It is often advantageous to have a wide span of performance with lower efficiencies rather than a peak performance with very limited range. (See Section 7.1; where, also, the validity of assuming a straight-line head-vs.-suction flow characteristic is discussed.)

Applications Beside the obvious advantages of being self-priming, having no moving parts, and requiring no lubrication, eductors can be made from any machinable material in addition to special materials, such as stoneware, Teflon,®* heat-resistant glass, and fiberglass. The applications throughout industry are too numerous to mention, but some of the more common will be discussed here. The type of eductor is determined by the service intended.

GENERAL PURPOSE EDUCTORS Table 1 is a capacity (flow rate) table for a general purpose eductor used for pumping and blending. This type of eductor, illustrated in Figure 6, has a broad performance span rather than a high peak efficiency point. Standard construction materials for this type of eductor are cast iron, bronze, stainless steel, and PVC. Typical uses include cesspool pumping, deep-well pumping, bilge pumping aboard ship, and condensate removal.

The following problem illustrates the use of Table 1.

EXAMPLE 2 Pump 30 gpm (6.81 m^3/h) of water from a sump 5 ft (0.61 m) below ground. Discharge to drain at atmospheric pressure. Motive water available is 40 lb/in^2 (2.8 bar) gage.

Solution Enter left side of Table 1 at 5 ft (1.5 m) suction lift and 0 lb/in^2 (bar) gage discharge pressure. Read horizontally across to 40 lb/in^2 (2.8 bar) gage operating water pressure. Read 9.6 gpm (2.18 m^3/h) suction and 7.3 gpm (1.66 m^3/h) operating fluid. These values are obtained in a 1-in (25.4-mm) eductor with a capacity ratio of 1.0.

To determine the capacity ratio of the required unit, divide the required suction by the quantity handled in 1-in (25.4-mm) eductor:

in USCS units $\quad \text{Capacity ratio} = \dfrac{30}{9.6} = 3.13$

in SI units $\quad \text{Capacity ratio} = \dfrac{6.81}{2.18} = 3.13$

*Teflon is a registered trademark of E. I. DuPont de Nemours and Co., Inc.

TABLE 1 Capacity (flow rate) table of standard 1-in (25.4-mm) water-jet eductors, gpm[a]

Suction lift, ft(m)	Discharge pressure, lb/in² (bar) gage	Function	Operating water pressure, lb/in² (bar) gage							
			10 (0.69)	20 (1.4)	30 (2.1)	40 (2.8)	50 (3.4)	60 (4.1)	80 (5.5)	100 (6.9)
	0 (0)	Suction	5.85	8.1	9.5	10.0	12.0	12.0	12.0	12.0
		Operating	3.55	5.0	6.1	7.1	7.9	8.7	10.0	11.0
	5 (0.34)	Suction	...	1.4	4.1	6.0	8.0	10.0	11.0	12.0
		Operating	...	4.9	6.1	7.0	7.9	8.6	10.0	11.0
	10 (0.69)	Suction	...	...	0.28	2.3	4.8	6.4	8.8	11.0
		Operating	...	...	5.9	6.8	7.8	8.5	9.8	11.0
0 (0)	15 (1.0)	Suction	...	...	...	...	1.2	3.4	5.9	8.6
		Operating	...	...	...	...	7.7	8.4	9.8	11.0
	20 (1.4)	Suction	...	...	...	...	...	0.3	3.5	5.9
		Operating	...	...	...	...	...	8.2	9.7	11.0
	25 (1.7)	Suction	...	...	...	...	...	...	0.83	3.9
		Operating	...	...	...	...	...	...	9.6	11.0
	30 (2.1)	Suction	...	...	...	...	...	...	...	1.7
		Operating	...	...	...	...	...	...	...	11.0
	0 (0)	Suction	4.4	6.8	8.6	9.6	11.0	11.0	12.0	12.0
		Operating	3.9	5.3	6.4	7.3	8.1	8.8	10.0	11.0
	5 (0.34)	Suction	...	1.5	3.2	5.0	7.0	9.0	11.0	11.0
		Operating	...	5.2	6.3	7.2	8.0	8.7	10.0	11.0
	10 (0.69)	Suction	...	...	...	1.9	3.6	5.6	8.6	10.0
		Operating	...	...	...	7.1	7.9	8.6	10.0	11.0
5 (1.5)	15 (1.0)	Suction	...	...	...	...	1.1	2.6	5.8	8.3
		Operating	...	...	...	...	7.8	8.6	9.9	11.0
	20 (1.4)	Suction	...	...	...	...	...	...	3.3	5.6
		Operating	...	...	...	...	...	...	9.8	11.0
	25 (1.7)	Suction	...	...	...	...	...	...	0.47	3.6
		Operating	...	...	...	...	...	...	9.8	11.0
	30 (2.1)	Suction	...	...	...	...	...	...	...	1.5
		Operating	...	...	...	...	...	...	...	11.0
	0 (0)	Suction	2.0	4.6	6.7	8.3	9.0	10.0	10.0	10.0
		Operating	4.2	5.5	6.6	7.4	8.2	9.0	10.0	11.0
	5 (0.34)	Suction	...	...	2.0	4.3	5.9	7.7	9.9	10.0
		Operating	...	...	6.5	7.4	8.2	8.9	10.0	11.0
	10 (0.69)	Suction	...	...	...	1.1	3.0	4.5	8.1	9.6
		Operating	...	...	...	7.3	8.1	8.8	10.0	11.0
10 (3.0)	15 (1.0)	Suction	...	...	...	...	1.1	2.1	5.6	7.3
		Operating	...	...	...	...	8.0	8.7	10.0	11.0
	20 (1.4)	Suction	...	...	...	...	...	...	2.8	5.3
		Operating	...	...	...	...	...	...	9.9	11.0
	25 (1.7)	Suction	...	...	...	...	...	...	...	2.8
		Operating	...	...	...	...	...	...	...	11.0
	30 (2.1)	Suction	...	...	...	...	...	...	...	1.1
		Operating	...	...	...	...	...	...	...	11.0
	0 (0)	Suction	...	3.3	5.3	7.9	8.4	8.9	8.9	9.1
		Operating	...	5.7	6.8	7.6	8.4	9.1	10.0	12.0
	5 (0.34)	Suction	...	...	...	4.0	4.9	7.3	8.6	9.1
		Operating	...	...	...	7.6	8.3	9.0	10.0	11.0
	10 (0.69)	Suction	...	...	...	...	2.4	4.0	6.4	8.6
		Operating	...	...	...	...	8.2	9.0	10.0	11.0
15 (4.6)	15 (1.0)	Suction	...	...	...	...	...	...	4.2	6.8

(*Continued*)

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TABLE 1 Capacity (flow rate) table of standard 1-in (25.4-mm) water-jet eductors, gpm[a] (*Continued*)

Suction lift, ft(m)	Discharge pressure, lb/in² (bar) gage	Function	Operating water pressure, lb/in² (bar) gage							
			10 (0.69)	20 (1.4)	30 (2.1)	40 (2.8)	50 (3.4)	60 (4.1)	80 (5.5)	100 (6.9)
		Operating	...	...	...	...	...	...	10.0	11.0
	20 (1.4)	Suction	...	...	...	...	...	...	2.1	4.5
		Operating	...	...	...	...	...	...	10.0	11.0
	25 (1.7)	Suction	...	...	...	...	...	...	...	1.9
		Operating	...	...	...	...	...	...	...	11.0
	0 (0)	Suction	...	2.0	4.0	6.4	7.8	7.8	7.8	7.8
		Operating	...	6.0	7.0	7.8	8.6	9.3	11.0	12.0
	5 (0.34)	Suction	...	...	...	2.8	3.9	6.3	7.8	7.8
		Operating	...	...	...	7.7	8.5	9.2	10.0	12.0
	10 (0.69)	Suction	...	...	...	...	1.2	3.1	5.7	7.1
		Operating	...	...	...	...	8.3	9.1	10.0	12.0
20 (6.1)	15 (1.0)	Suction	...	...	...	...	...	...	3.6	5.4
		Operating	...	...	...	...	...	...	10.0	11.0
	20 (1.4)	Suction	...	...	...	...	...	...	1.4	3.8
		Operating	...	...	...	...	...	...	10.0	11.0
	25 (1.7)	Suction	...	...	...	...	...	...	...	1.5
		Operating	...	...	...	...	...	...	...	11.0

Relative capacities of standard sizes

Size eductor, in (mm)	½ (12.7)	¾ (19.1)	1 (25.4)	1½ (38.1)	2 (50.8)	2½ (63.5)	3 (76.2)	4 (102)	6 (152)
Capacity ratio	0.36	0.64	1.00	2.89	4.00	6.25	9.00	16.00	36.00

[a]gpm × 0.227 = m³/h.
Source: Schutte and Koerting.

Referring to the bottom of Table 1, a 2-in (50.8-mm) eductor with a capacity ratio of 4.0 is obtained. The required motive flow is then

in USCS units $4(7.3) = 29.2$ gpm

in SI units $4(1.66) = 6.64$ m³/h

and the suction capacity is

in USCS units $4(9.6) = 38.4$ gpm

in SI units $4(2.18) = 8.72$ m³/h

A 1½-in (38-mm) unit can handle 2.89 times the values in Table 1, or 27.7 gpm (6.3 m³/h) suction when using 21 gpm (4.8 m³/h) motive water at 40 lb/in² (2.8 bar) gage. If suction flow rate is not critical, some capacity can be sacrificed in order to use a smaller and therefore lower-cost eductor. If optimum performance is desired, it is necessary to size a special eductor using Figures 3 and 4.

Figure 7 illustrates more streamlined versions for higher suction lifts or applications involving the handling of slurries. This type of eductor is often used to remove condensate from vessels under vacuum. The advantage is that eductors require only 2 ft (0.61 m) NPSH and, being smaller than mechanical pumps, save considerable space. Further, a partial vapor load is much less likely to vapor-lock a jet pump because the venturi tube

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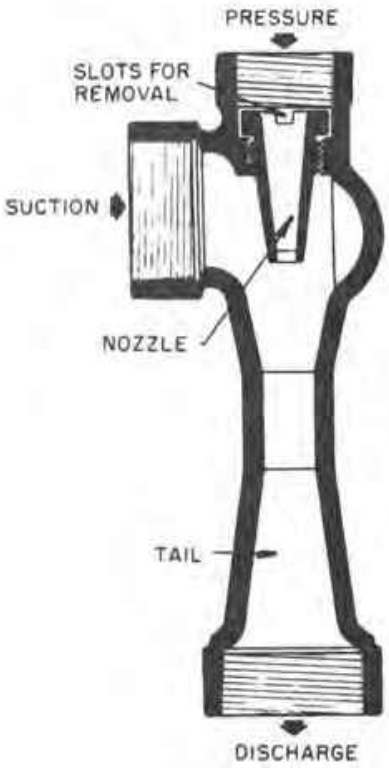


FIGURE 6 General purpose eductor (Schutte and Koerting)

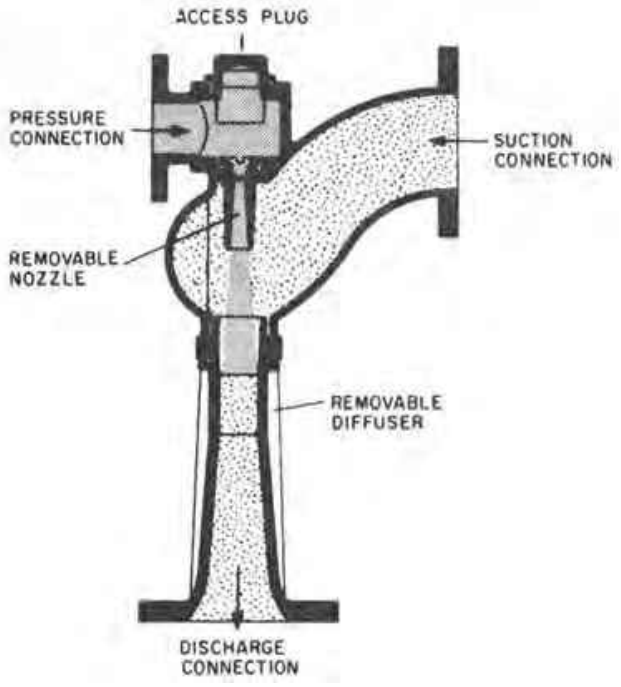


FIGURE 7 Streamlined eductor (Schutte and Koerting)

minimizes the expansion effect of flashing vapor. Sizing is done in the manner illustrated in Example 1, using Figures 3 and 4.

MIXING EDUCTORS Although any eductor is inherently a mixing device, some are specifically designed as mixers. They are used to replace mechanical agitators and are located inside the tank containing the fluid to be agitated. Figure 8 illustrates the simplest type of eductor, the *Sparger nozzle.* These units entrain a volume of suction fluid that is approximately three times the volume of motive fluid. A 20-lb/in² (1.4-bar) drop across the nozzle is recommended for proper mixing. Figure 9 shows the motive flow rates for this type

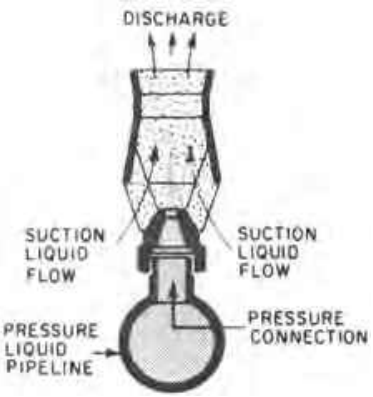


FIGURE 8 Sparger nozzle (Schutte and Koerting)

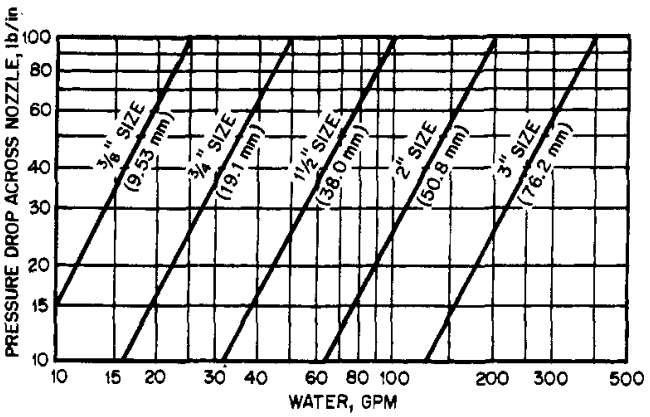


FIGURE 9 Motive flow rate of Sparger nozzles (gpm × 0.227 = m³/h; lb/in² × 0.0689 = bar) (Schutte and Koerting)

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TABLE 2 Motive flow rates of tank mixing eductors, gpm[a]

Size, in (mm)	Pressure difference, inlet to tank, lb/in² (bar) gage							
	10 (0.69)	20 (1.4)	30 (2.1)	40 (2.8)	50 (3.4)	60 (4.1)	80 (5.5)	100 (6.9)
½ (12.7)	3.5	5.0	6.0	7.0	8.0	8.5	10.0	11.0
¾ (19.1)	10.0	14.5	17.5	20.0	23.0	24.5	29.0	32.0
1 (25.4)	14.2	20.0	25.0	28.0	30.0	34.5	40.0	44.5
1¼ (31.8)	22.0	31.0	37.5	44.0	50.0	53.0	62.5	69.0
1½ (38.1)	31.5	45.0	54.0	63.0	72.0	76.5	90.0	99.0
2 (50.8)	56.0	80.0	96.0	112.0	128.0	136.0	160.0	176.0
3 (76.2)	126.0	180.0	216.0	252.0	288.0	306.0	360.0	396.0
4 (102)	224.0	320.0	384.0	448.0	512.0	544.0	640.0	704.0
5 (127)	350.0	500.0	600.0	700.0	800.0	850.0	1000.0	1100.0
6 (152)	494.0	720.0	864.0	1008.0	1152.0	1224.0	1440.0	1584.0

[a]gpm × 0.227 = m³/h
Source: Schutte and Koerting.

of eductor. Sparger nozzles are normally used for shallow tanks, whereas the following tank mixer described is preferred for deeper vessels.

Figure 10 illustrates a type of eductor called a *tank mixer.* It is installed under the tank containing the fluid to be agitated. Motive capacities are shown in Table 2. The units are usually custom-designed for a specific entrainment ratio, the required capacity being determined by the quantity of tank fluid, the ratio of mixture desired, and the depth of the tank being agitated.

EXAMPLE 3 It is desired to blend recycled tank fluid into a tank 20 ft (6.1 m) deep in a volume ratio of 1 motive to 1.5 suction. The tank contains 7500 gal (28.4 m³), and it is desired to turn over the tank in 30 min. The motive pump will deliver 60 lb/in² (4.14 bar) gage at the eductor nozzle. What size mixing eductor is needed?

Solution The 500 gal (28.4 m³) turned over in 30 minutes is equivalent to 250 gpm (56.8 m³/h). Because the motive fluid in this case is recycled from the tank, both motive and suction fluid contribute to the tank turnover. In the ratio of 1.5 suction to 1 motive fluid, the motive quantity required to attain a circulation rate of 250 gpm (56.8 m³/h) is 100 gpm (22.7 m³/h). To select the size, it is necessary to obtain the differential pressure across the nozzle orifice of the eductor. Because the eductor is below the tank, the net driving head is 60 lb/in² gage − 20/2.31 = 51.35 lb/in² (4.14 − 6.1/10.2 = 3.54 bar) gage across the nozzle. Enter Table 2 and interpolate between 50 and 60 lb/in² (3.4 and 4.1 bar) gage. A 1½-in (38-mm) eductor will pass only 73 gpm (16.6 m³/h), whereas a 2-in (51-mm) eductor will pass 129 gpm (29.3 m³/h). The selection would then be a 2-in (51-mm) mixing eductor.

SPINDLE PROPORTIONING EDUCTORS Another type of mixing eductor is illustrated in Figure 11. Typical applications of this type include mixing hydrocarbons with caustic, oxygen, or copper chloride slurries; producing emulsions; and proportioning liquids in chemical process industries. In critical applications, the regulating spindle is sometimes fitted with a diaphragm operator to achieve close control. Table 3 shows operating pressures and flow rates on several typical applications for units of this type.

SAND AND MUD EDUCTORS Figure 12 illustrates a sand and mud eductor used for pumping out wells, pits, tanks, sumps, and similar containers where there is an accumulation of

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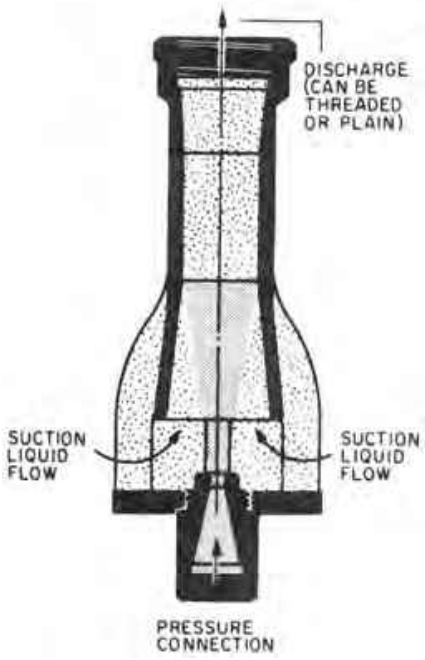


FIGURE 10 Tank mixing eductor (Schutte and Koerting)

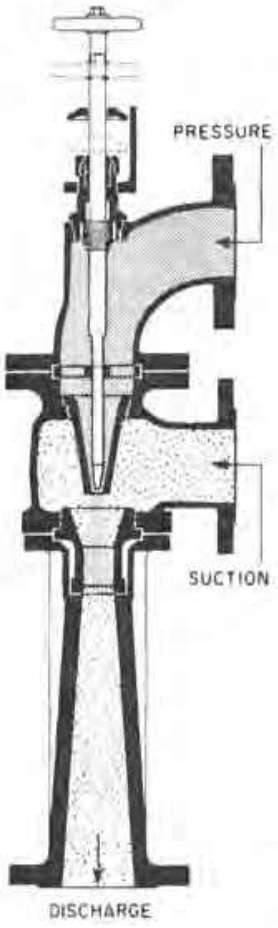


FIGURE 11 Proportioning eductor (Schutte and Koerting)

TABLE 3 Operating pressures and flow rates—proportioning eductor

Motive liquid/ suction fluid	Naphtha/ copper chloride slurry	Hydrocarbon/ hydrocarbon	Gasoline/ slurry	Gasoline/ water	Sour kerosene/ kerosene slurry
Pressure, lb/in² (bar) gage					
Motive	165 (11.4)	295 (20.3)	170 (11.7)	75 (5.2)	146 (10.1)
Suction	40 (2.8)	5 (0.3)	75 (5.2)	50 (3.4)	60 (4.1)
Discharge	75 (5.2)	10 (0.7)	100 (6.9)	50 (3.4)	70 (4.8)
Flow, gpm (m^3/h)					
Motive	30 (6.8)	10 (2.3)	90 (20.4)	170 (38.6)	482 (109.5)
Suction	20 (4.5)	58 (13.2)	74 (16.8)	42 (9.5)	700 (159.0)
Discharge	50 (11.4)	68 (15.4)	164 (37.2)	212 (48.1)	1182 (268.5)
Eductor size, in (mm)	$1\frac{1}{2}$ (38.1)	3 (76.2)	4 (102)	4 (102)	6 (152)

Source: Schutte & Koerting.

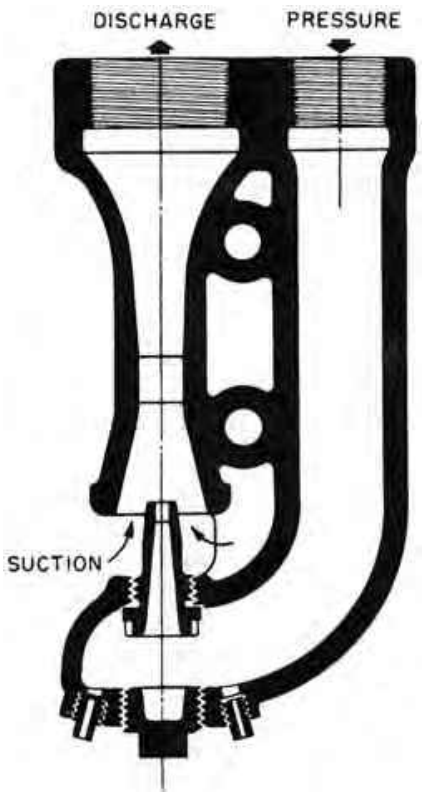


FIGURE 12 Sand and mud eductor (Schutte and Koerting)

sand, mud, slime, or other material not easily handled by other eductors. With this type of eductor, the bottom of the pressure chamber is fitted with a ring of agitating nozzles that stirs the material in which the jet is submerged to allow maximum entrainment. Relative capacities for this type of eductor are shown in Table 4, which is used in the same manner as Table 1. The required suction flow is divided by the suction capacity selected from Table 4 under the appropriate motive pressure. This value is the capacity ratio. From the table, select the eductor by choosing the next highest capacity ratio. Actual flow rates are then determined by multiplying the values in the table by the capacity ratio of the eductor selected. Maximum discharge head is read from the table.

SOLIDS-HANDLING EDUCTORS Figure 13 illustrates a specific type of eductor called a *hopper eductor,* made for handling slurries or dry solids in granular form and used for

TABLE 4 Relative capacities (flow rates) of sand and mud eductors

Capacity of standard 3-in (76.2-mm) eductor			
Operating water pressure, lb/in² (bar) gage	40.0 (2.8)	50.0 (3.4)	60.0 (4.1)
Total motive fluid, gpm (m³/h)	69.5 (15.8)	77.5 (17.6)	85.0 (19.3)
Net suction fluid, gpm (m³/h)	30.0 (6.8)	34.5 (7.8)	38.5 (8.7)
Maximum discharge head, ft (m)	22.0 (6.7)	26.0 (7.9)	32.0 (9.8)

Relative capacities of standard sizes						
Size eductor, in (mm)	$1\frac{1}{2}$ (38.1)	$2\frac{1}{2}$ (63.5)	3 (76.2)	4 (102)	5 (127)	6 (152)
Capacity ratio	0.29	0.62	1.00	1.85	2.80	3.80

Source: Schutte and Koerting.

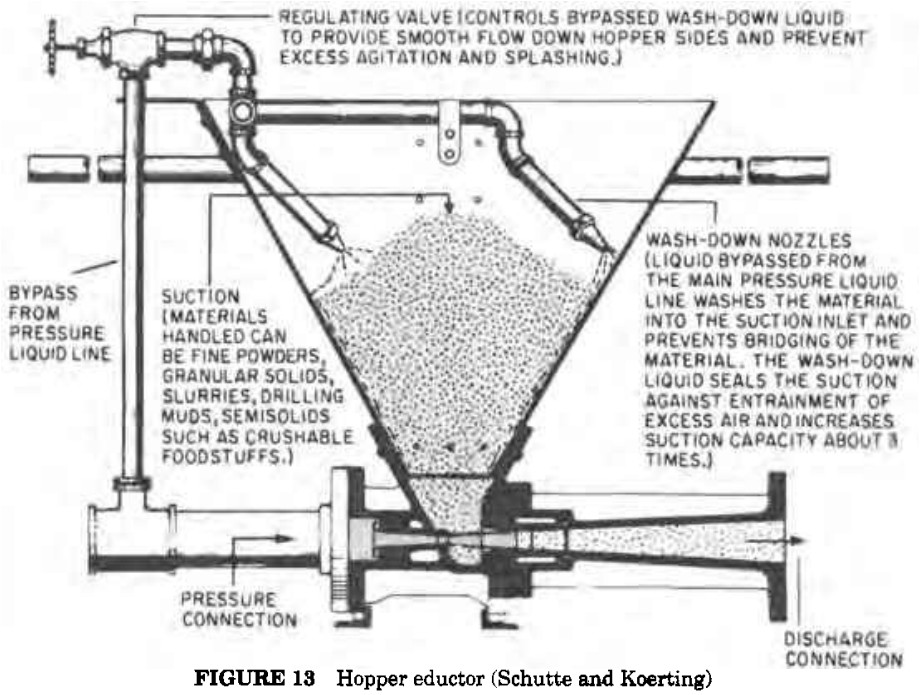


FIGURE 13 Hopper eductor (Schutte and Koerting)

ejecting sludges from tank bottoms, pumping sand from filter beds, and washing or conveying granular materials. Typical construction is cast iron with hardened steel nozzle and throat bushings. In operation, the washdown nozzles are adjusted to provide smooth flow down the hopper sides, thus preventing bridging of the material being handled and also sealing the eductor suction against excess quantities of air. Without this seal, the capacities shown in Table 5 should be divided by approximately 3. Table 6 shows typical materials handled by this eductor and their bulk density. Use of the capacity table for hopper eductors is similar to use of Tables 1 and 4, except the suction quantities required are expressed in cubic feet (cubic meters). Capacity ratio is determined by dividing the value in the table into the required suction flow, and the next largest size eductor is selected.

Another type of solids-handling eductor is illustrated in Figure 14. This *annular-orifice eductor* is used where the material being handled tends to agglomerate and gum up when wetted and has been used successfully for handling and mixing hard-to-wet solids. In this

TABLE 5 Relative capacities (flow rates) of hopper eductors

Capacity of standard $1\frac{1}{2}$-in (38-mm) eductor				
Operating water pressure, lb/in² (bar) gage	30 (2.1)	40 (2.8)	50 (3.4)	60 (4.1)
Suction capacity, ft³/h (m³/h)	13 (0.37)	36 (1.0)	72 (2.0)	90 (2.5)
Maximum discharge pressure, lb/in² (bar) gage	14 (1.0)	17 (1.1)	18 (1.2)	20 (1.4)
Motive water consumption, gpm (m³/h)[a]	35 (7.9)	40 (9.1)	45 (10.2)	50 (11.4)

Relative capacities of standard sizes					
Size, in (mm)	$1\frac{1}{2}$ (38.1)	2 (50.8)	3 (76.2)	4 (102)	6 (152)
Capacity ratio	1.00	1.60	3.50	6.00	18.00

[a]Based on using approximately 10% motive water through washdown nozzles.
Source: Schutte and Koerting.

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TABLE 6 Typical materials handled by hopper eductors

Material	Approx. bulk density, lb/ft³ (kg/m³)
Borax	50–55 (800–880)
Charcoal	18–28 (290–450)
Diatomaceous earth	10–20 (160–320)
Lime, pebble	56 (900)
Lime, powdered	32–40 (510–640)
Fly ash	35–40 (560–640)
Mash	60–65 (960–1040)
Rosin	67 (1070)
Salt, granulated	45–51 (720–820)
Salt, rock	70–80 (1120–1280)
Sand, damp	75–85 (1200–1360)
Sand, dry	90–100 (1440–1600)
Sawdust, dry	13 (210)
Soda ash, light	20–35 (320–560)
Sodium nitrate, dry	80 (1280)
Sulfur, powdered	50–60 (800–960)
Wheat	48 (770)
Zinc oxide, powdered, dry	10–35 (160–560)

Source: Schutte and Koerting.

unit, intimate mixing occurs in the throat, and the device is virtually clogproof. Normally this unit is installed directly over the tank into which the mixture is discharged. Table 7 shows capacities for this type of unit.

Capacity Table 7 is similar to Table 5, and the selection method is the same as discussed previously.

MULTINOZZLE EDUCTORS Figure 15 illustrates an annular multinozzle eductor designed for special applications where the suction fluid contains solids or semisolids. It is used primarily for large flows at low discharge heads. Because these units have relatively large air-handling capacities, they are well suited for priming large pumps, such as dredging pumps, where air pockets can cause these pumps to lose their prime. These eductors are designed by using the basic equations for head ratio. The appropriate efficiency factor is selected from Figure 2, and the volumetric flow ratio is calculated. Figure 4 is used to size the eductor after discharge flow has been determined.

DEEP-WELL EDUCTORS The eductor illustrated in Figure 16 is typical of those used in conjunction with a mechanical pump for commercial and residential water supply from a deep well. The eductor is used to lift water from a level below barometric height up to a level where the suction of the motive pump at the surface can lift the water the remaining distance.

In operation, the eductor is fitted with hoses connected to the suction and discharge of the motive pump and dropped into the well casing. An initial prime is required, which is maintained by the foot valve at the suction of the eductor. When the surface pump is activated, pressure water through the eductor entrains water from the well, lifting it high enough to enable the mechanical pump to carry it to the surface. A bypass valve at the surface diverts the suction quantity to a receiving tank.

Capacities of these units depend on the depth of the well and the centrifugal pump. The standard commercial unit has 1-in (25-mm) pressure and $1\frac{1}{4}$-in (32-mm) discharge connections and is available with a variety of nozzle and diffuser combinations for use with standard centrifugal pumps at varying depths. The following example illustrates how to calculate this type of unit.

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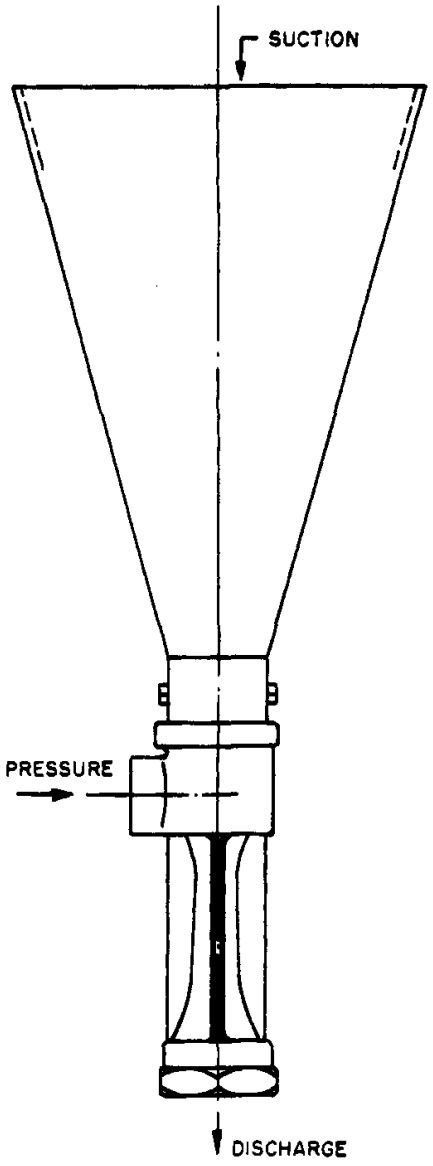


FIGURE 14 Annual eductor (Schutte and Koerting)

EXAMPLE 4 A centrifugal pump with a capacity of 100 gpm (22.7 m³/h) at a total discharge head of 150 ft (45.7 m) and requiring 10 ft (3.05 m) NPSH is available to operate an eductor to pump water at 50 ft (15.2 m) below grade. Find the quantity of water that can be delivered at 60 lb/in² (4.1 bar) gage (see Figure 17).

Solution The available operating head is 138.6 ft + 50 ft − frictional loss (42.2 + 15.2 − frictional loss). As a first assumption, the frictional loss is ignored and the head ratio is 188.6/40.83 = 4.62 (57.4/12.38 = 4.62). From Figure 2 at NPSH 33.6 ft (10.24 m), $\varepsilon = 0.9$ and

$$R_w = R_q = 0.9\sqrt{4.62} - 1 = 0.934$$

(from Eqs. 5 and 6)

With Q_R fixed at 100 gpm (22.7 m³/h),

TABLE 7 Relative capacities (flow rates) of annual eductors

Capacity of standard $1\frac{1}{2}$-in (38.1-mm) mixing eductor, 5 lb/in² (0.34 bar) gage discharge pressure					
Motive pressure, lb/in²(bar) gage	30 (2.1)	40 (2.8)	60 (4.1)	80 (5.5)	100 (6.9)
Entrainment, ft³/h (m³/h)	2.6 (0.07)	7.1 (0.20)	17.9 (0.51)	22.0 (0.62)	23.8 (0.67)
Motive flow, gpm (m³/h)	12.7 (2.88)	14.6 (3.31)	17.9 (4.06)	20.7 (4.70)	23.1 (5.24)

Relative capacities of standard sizes						
Size, in (mm)	$1\frac{1}{4}$ (31.8)	$1\frac{1}{2}$ (38.1)	2 (50.8)	$2\frac{1}{2}$ (63.5)	3 (76.2)	4 (102)
Capacity ratio	0.62	1.00	1.43	2.86	4.76	8.80

Source: Schutte and Koerting.

in USCS units $$Q_1 = \frac{100}{1 + R_q} = \frac{100}{1.934} = 51.7 \text{ gpm}$$

in SI units $$Q_1 = \frac{22.7}{1.934} = 11.7 \text{ m}^3/\text{h}$$

The motive line size is now chosen by selecting a reasonable velocity and frictional loss. Choosing a 2-in (51-mm) pipe size, velocity is 5.57 ft/s (1.70 m/s) and frictional loss is 6.1 ftH_2O (1.86 mH_2O). The revised operating head becomes 188.6 − 6.1 = 182.5 ft (57.4 − 1.86 = 55.6 m) and

in USCS units $$R_q = 0.9\sqrt{182.5/40.83} - 1 = 0.9$$

in SI units $$R_q = 0.9\sqrt{55.6/12.38} - 1 = 0.9$$

then Q_1 = 100/1.9 = 52.6 gpm (22.7/1.9 = 11.9 m³/h). (This value is close enough so that a third trial is not necessary.) The suction flow that can be delivered is then

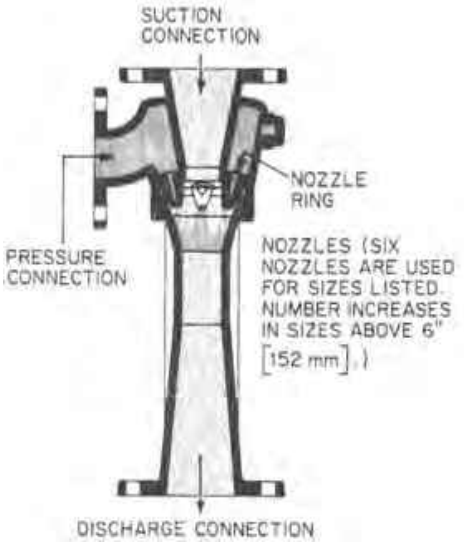


FIGURE 15 Multinozzle eductor (Schutte and Koerting)

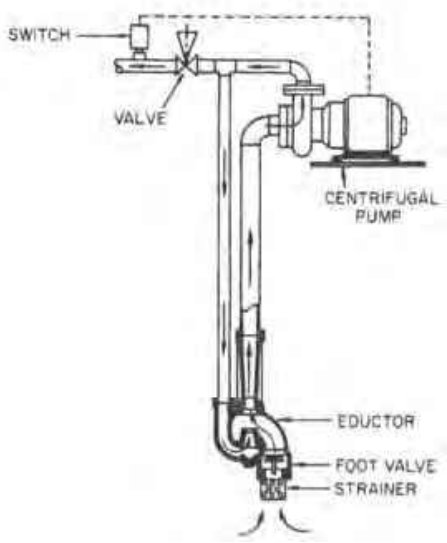


FIGURE 16 Centrifugal-jet pump combination

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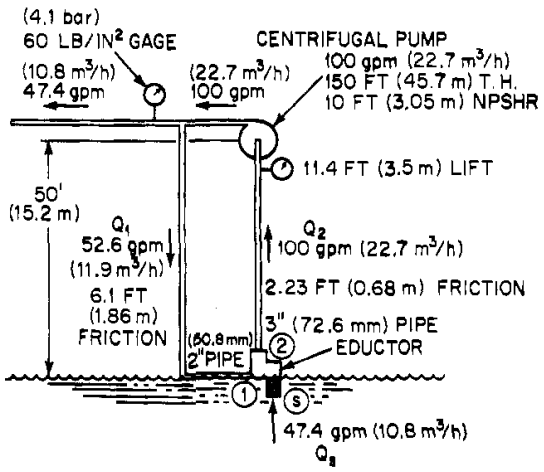


FIGURE 17 Centrifugal-jet pump for Example 4

in USCS units	$100 - 52.6 = 47.4$ gpm
in SI units	$22.7 - 11.9 = 10.8$ m³/h

PRIMING EDUCTORS—WATER-JET EXHAUSTERS Eductors are often used as priming devices for mechanical pumps. In this application, the eductor is used to remove air rather than water. Liquid jets are not well suited for pumping noncondensables; therefore, the capacities are low. However, the volume being primed is usually small, and so the low capacity is not a factor. When larger volumes are involved, such as condenser water boxes, it is more feasible to use an exhauster. The water-jet eductor of Figure 6 is converted to a water-jet exhauster by replacing the jet nozzle with a solid-cone spray nozzle. Evacuating rates and capacity tables for such a unit are shown in Figure 18 and Table 8. Eductors have approximately one-fifth the air-handling capacities of water-jet exhausters when supplied with similar motive quantities and pressures.

EXAMPLE 5 From Figure 18 and Table 8, determine size and water consumption to exhaust 15 standard ft³/min (0.42 m³/min) of air at 20 inHg (508 mmHg) abs discharging to atmosphere using 60 lb/in² (4.14 bar) gage motive water at 80°F (27°C).

Solution Enter Figure 18 at 80°F (27°C) (1); read horizontally to the suction pressure 20 inHg (508 mmHg) abs (2); project vertical line to 60 lb/in² (4.14 bar) gage motive pressure (3); project a horizontal line for the capacity of a 1-in (25-mm) exhauster (4); divide desired flow by the capacity of a 1-in (25-mm) unit, which is 1.9 standard ft³/min, (0.054 m³/min), to find capacity ratio: 15/1.9 = 7.9 (0.42/0.054 = 7.9).

The capacity ratio table shows that a 3-in (76-mm) exhauster with a capacity ratio of 9.0 is required. The motive water quantity from Table 8 is 86 gpm (19.5 m³/h). *Note:* Table 8 gives water consumption at 15 inHg (381 mmHg) abs; because flow varies as the square of pressure differential across the nozzle, the exact flow is obtained as follows:

Nozzle upstream pressure:

in USGS units	$60 + 14.7 = 74.7$ lb/in² abs
in SI units	$4.14 + 1.01 = 5.15$ bar abs

Nozzle downstream pressure:

in USCS units $$20 \text{ inHg abs} \left(\frac{14.7 \text{ lb/in}^2}{30 \text{ inHg}} \right) = 9.8 \text{ lb/in}^2 \text{ abs}$$

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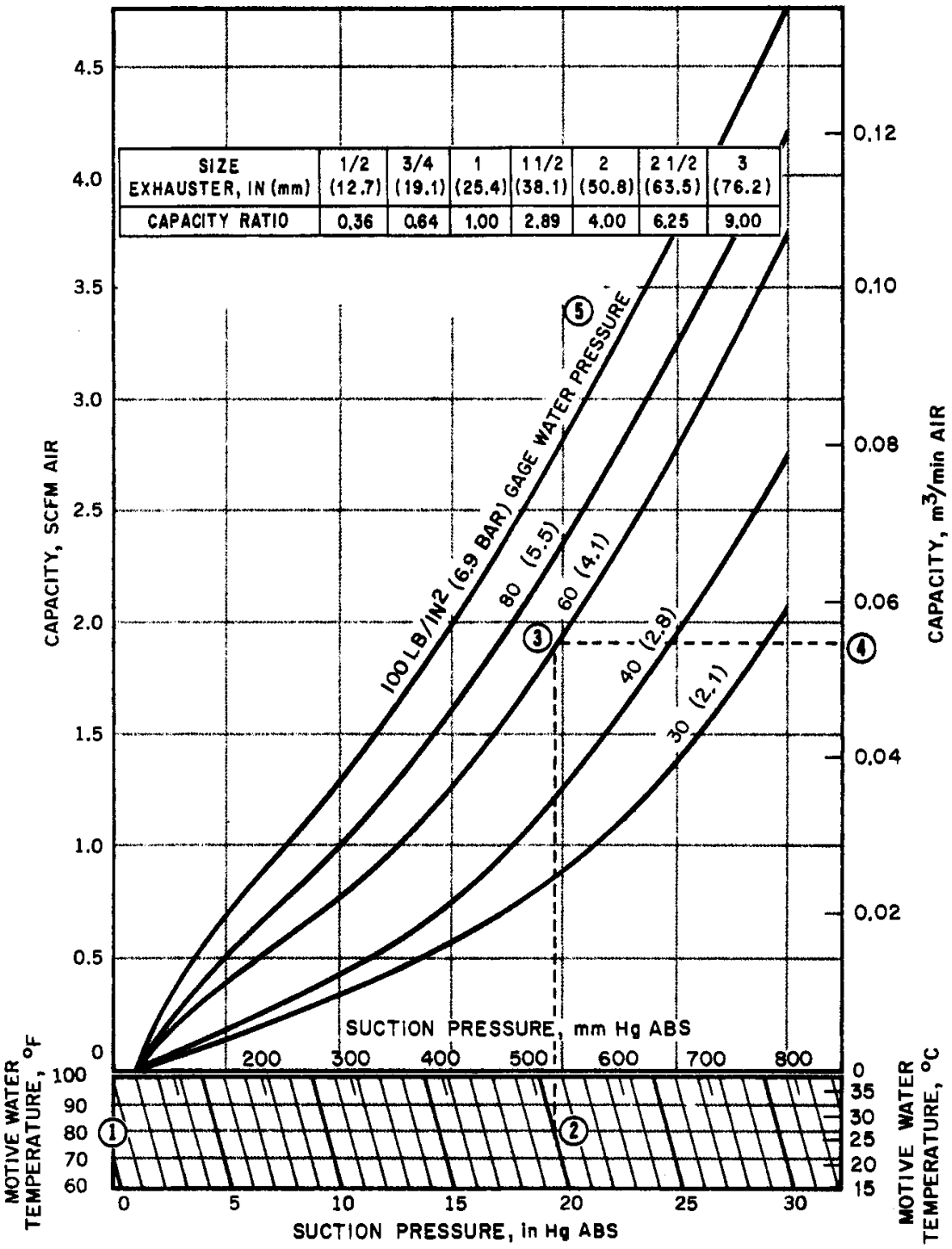


FIGURE 18 Capacity curve of water-jet exhausters (Schutte and Koerting)

in SI units $508 \text{ mmHg abs} \left(\frac{1.01 \text{ bar}}{762 \text{ mmHg}}\right) = 0.67 \text{ bar abs}$

Operating differential:

in USCS units $74.7 - 9.8 = 64.9 \text{ lb/in}^2$

in SI units $5.15 - 0.67 = 4.48 \text{ bar}$

TABLE 8 Approximate water consumption of water-jet exhausters, gpm[a, b]

	Water pressure, lb/in² (bar) gage				
Size, in (mm)	30 (2.1)	40 (2.8)	60 (4.1)	80 (5.5)	100 (6.9)
½ (12.7)	2.6	2.9	3.4	3.8	4.2
¾ (19.1)	4.6	5.3	6.4	7.4	8.3
1 (25.4)	6.2	6.8	8.1	9	10
1½ (38.1)	20	23	27	30	32
2 (50.8)	28	31	36	41	45
2½ (63.5)	46	51	60	67	73
3 (76.2)	66	73	86	96	106

[a] gpm × 0.227 = m³/h
[b] All flows at 15 in (381 mm) Hg abs.
Source: Schutte and Koerting.

Table differential:

in USCS units $$74.7 - 15\left(\frac{14.7}{30}\right) = 67.35 \text{ lb/in}^2$$

in SI units $$5.15 - 381\left(\frac{1.01}{762}\right) = 4.64 \text{ bar}$$

Actual flow:

in USCS units $$86\left(\frac{64.9}{67.35}\right)^{1/2} = 84.4 \text{ gpm}$$

in SI units $$19.5\left(\frac{4.48}{4.64}\right)^{1/2} = 19.2 \text{ m}^3/\text{h}$$

SIPHONS

Operation As previously defined, the term *siphon* refers to a jet pump utilizing a condensable vapor to entrain a liquid and discharge to a pressure intermediate between motive and suction pressure. The principal motive fluid is steam.

In an eductor, the high-pressure motive fluid enters through a nozzle and creates a vacuum by jet action, which causes suction fluid to enter the mixing chamber. The siphon of Figure 19 is identical to the eductor of Figure 6 except that, unlike the eductor, the siphon motive nozzle is a converging-diverging nozzle to achieve maximum velocity at the nozzle tip. The velocity is supersonic at this point. The motive fluid is condensed into the suction fluid on contact and imparts its energy to the liquid, thus impelling it through the diffuser. The diffuser section is the same as an eductor diffuser, and it converts the velocity energy to pressure at the discharge. To achieve maximum performance, the siphon nozzle must be expanded to the desired suction pressure in order to achieve the highest possible velocity. Because negligible radiation losses are encountered, the siphon is 100% thermally efficient in that the heat in the incoming water plus the heat in the operating steam must equal the heat of the mixture plus its mechanical energy. Furthermore, the momentum of the incoming water plus the momentum of the expanded steam is equal to the momentum of the discharge mixture less impact and frictional losses.

It is important that the motive steam be condensed in the suction liquid prior to the throat for proper operation. If condensation does not occur, full available energy is not transferred. Furthermore, energy must be expended to recompress the uncondensed

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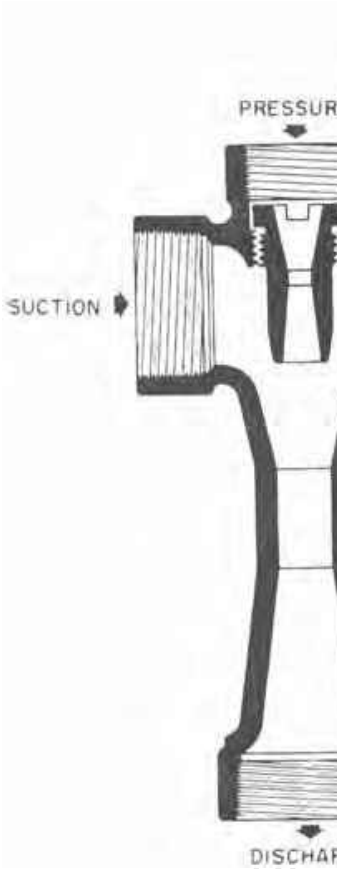


FIGURE 19 Standard siphon (Schutte and Koerting)

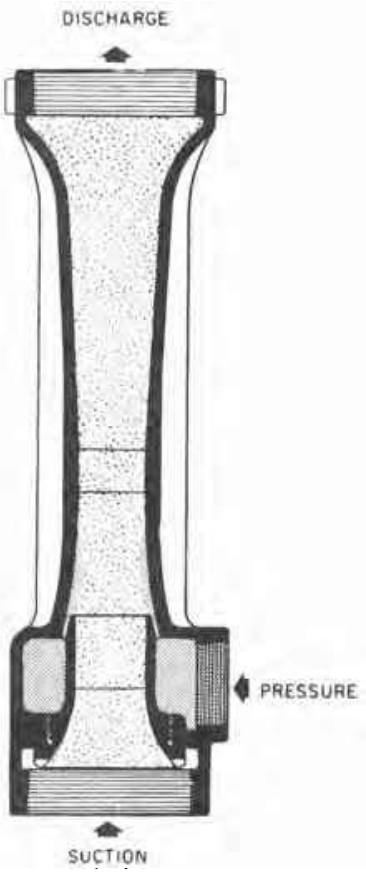


FIGURE 20 Annular siphon (Schutte and Koerting)

steam. For this reason, discharge temperature cannot exceed the boiling point at the discharge pressure. The fact that energy is required to recompress any uncondensed steam explains why air is a very poor motive fluid for liquid pumping.

STANDARD SIPHONS Tables 9 and 10 illustrate the capacity and operating characteristics of standard siphons. These tables are similar to the eductor capacity tables. To size a unit, read the suction capacity of a $1\frac{1}{2}$-in (38-mm) unit from Table 9 at the appropriate motive steam pressure, suction lift, and discharge head. Divide the desired suction flow by this capacity to find the capacity ratio. From Table 10, find the unit with the next largest capacity ratio. Read the motive steam required under the proper motive pressure.

Standard materials of construction are cast iron, bronze, stainless, steel, and Pyrex®. If desired, special capacity ratios can be achieved by using a custom-designed unit. Sizes over 6 in (152 mm) can be fabricated of any suitable material.

ANNULAR SIPHONS Figure 20 illustrates an annular siphon. This unit is identical to the eductor of Figure 14 except that steam is the motive fluid. Capacity Table 11 is used in the same manner as previous examples. This type of siphon is used when inline flow is desired or when the suction liquid contains some solids. Units are available in cast iron through an 8-in (203-mm) size. Special materials or fabricated designs are also available.

OTHER JET PUMP DEVICES

Air Siphons As previously mentioned, air is a very poor motive fluid for entraining a liquid because energy must be expended in compressing the air back to the discharge pressure. There are, however, applications where it is necessary to sample a liquid with no dilution.

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TABLE 9 Relative capacities of steam-jet siphons, gpm[a]

Suction lift, ft (m)	Suction temp., °F[b]	Operating steam pressure, lb/in² (bar) gage								Operating steam pressure, lb/in² (bar) gage							
		40 (2.8)	50 (3.4)	60 (4.1)	80 (5.5)	100 (6.9)	120 (8.3)	160 (11)	240 (16)	40 (2.8)	50 (3.4)	60 (4.1)	80 (5.5)	100 (6.9)	120 (8.3)	160 (11)	240 (16)
		0-ft (m) discharge head								20-ft (6.1-m) discharge head							
	70	52	51	51	49	46	43	37	30	36	47	48	48	45	41	35	29
	90	45	44	43	42	40	37	33	26	37	44	43	43	41	38	33	26
1 (0.3)	110	40	38	36	36	35	33	28	22	38	39	37	37	36	34	30	23
	130	35	32	30	30	29	29	25	...	35	33	31	31	30	29	25	...
	150	26	25	24	24	24	24	21	...	26	25	24	24	23	22	20	...
	165	17	17	17	18	18	17	17	...	17	17	17	17	17	17	16	...
	70	38	38	37	35	30	28	25	20	27	36	37	35	31	29	25	19
	90	34	34	33	30	27	25	21	17	26	32	32	29	26	23	20	17
10 (3.0)	110	28	27	26	25	23	21	18	...	26	27	27	25	23	21	18	...
	130	21	21	21	20	18	16	14	...	21	22	22	21	18	16	14	...
	145	16	16	16	16	14	12	...	...	16	16	16	16	14	12	...	...
	70	34	32	30	26	23	21	18	14	24	33	32	27	24	23	19	15
	90	29	28	26	23	20	18	16	12	23	28	27	23	20	19	16	12
15 (4.6)	110	24	23	22	19	17	15	13	...	23	23	22	19	17	15	13	...
	130	17	17	17	15	13	11	...	...	17	17	17	14	13	...	...	...
	145	10	12	11	9	...	...	...	...	11	11	10	10	...	...	...	...
	70	26	23	21	18	16	15	13	...	24	24	22	19	17	15	12	...
	90	22	19	17	15	14	12	11	...	19	20	18	15	14	12	11	...
20 (6.1)	110	18	16	14	12	11	10	...	...	17	16	14	12	11	...	...	...
	125	13	12	11	...	...	...	...	...	12	11	10	...	...	...	...	...

(*Continued*)

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TABLE 9 Relative capacities of steam-jet siphons, gpm[a] (*Continued*)

Suction lift, ft (m)	Suction temp., °F[b]	Operating steam pressure, lb/in² (bar) gage								Operating steam pressure, lb/in² (bar) gage							
		40 (2.8)	50 (3.4)	60 (4.1)	80 (5.5)	100 (6.9)	120 (8.3)	160 (11)	240 (16)	40 (2.8)	50 (3.4)	60 (4.1)	80 (5.5)	100 (6.9)	120 (8.3)	160 (11)	240 (16)
		40-ft (12.2-m) discharge head								50-ft (15.2-m) discharge head							
1 (0.3)	70	...	20	33	47	44	41	36	29	...	...	18	44	44	41	36	28
	90	...	18	36	43	42	39	34	27	...	...	21	42	42	39	34	27
	110	...	20	36	37	37	35	30	24	...	...	24	37	36	34	30	24
	130	...	23	31	31	30	29	25	...	...	...	26	30	30	29	25	...
	150	...	24	24	24	24	24	20	...	...	...	24	24	24	24	20	...
	165	...	17	18	18	18	18	16	...	...	...	18	18	18	18	16	...
10 (3.0)	70	...	...	24	34	30	27	24	18	...	...	...	35	30	27	23	18
	90	...	...	26	29	26	24	20	17	...	...	...	29	27	24	21	17
	110	...	...	27	26	23	20	18	...	...	...	...	25	23	20	18	...
	130	...	...	22	21	19	16	14	...	...	...	...	21	19	17	14	...
	145	...	...	16	16	14	12	...	...	...	...	...	16	14	13	...	...
15 (4.6)	70	...	...	23	28	24	22	19	15	...	...	...	27	24	21	18	14
	90	...	...	20	24	20	18	16	...	...	...	...	24	21	18	16	...
	110	...	...	21	19	17	15	...	...	...	...	...	19	17	15	...	...
	130	...	...	17	14	12	...	...	...	...	...	...	14	...	...	...	...
	145	...	...	11	...	...	...	...	...	...	...	...	...	...	...	...	...
20 (6.1)	70	...	...	21	19	16	15	12	...	...	...	...	...	16	15	...	...
	90	...	...	18	16	14	13	11	...	...	...	...	...	15[c]	13[c]	...	...
	110	...	...	...	11	10	...	...	...	...	...	...	...	...	...	...	...

[a] gpm = 0.227 m³/h

[b] °C = (°F − 32)/1.8

[c] Suction temperature 85°F (29.4°C)

Source: Schutte and Koerting.

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TABLE 10 Steam consumption of steam-jet siphons, lb/h[a]

Siphon size, in (mm)	Capacity ratio	Operating steam pressure, lb/in² (bar) gage							
		40 (2.8)	50 (3.4)	60 (4.1)	80 (5.5)	100 (6.9)	120 (8.3)	160 (11.0)	240 (16.5)
½ (12.7)	0.125	40	47	54	69	83	97	126	84
¾ (19.1)	0.222	70	83	96	122	147	173	222	322
1 (25.4)	0.346	110	130	150	190	230	270	350	510
1½ (38.1)	1.000	318	376	434	550	665	780	1,012	1,475
2 (50.8)	1.38	440	520	600	761	920	1,080	1,400	2,040
2½ (63.5)	2.0	635	750	865	1,100	1,329	1,558	2,020	2,940
3 (76.2)	3.11	990	1,170	1,350	1,710	2,065	2,425	3,145	4,590
4 (102)	5.54	1,760	2,085	2,400	3,045	3,685	4,320	5,500	8,170
6 (152)	12.45	3,960	4,680	5,400	6,850	8,280	9,710	12,600	18,360

[a]Lb/h × 0.454 = kg/h
Source: Schutte and Koerting.

Small units (less than 1 gpm, 0.23 m³/h) can be supplied for limited discharge pressures, as indicated by Figure 21. With air as the motive fluid, the suction liquid can be very close to its boiling point and only a very slight NPSH is required.

When air is used as a motive fluid, the smaller sizes operate more efficiently because the air is more intimately mixed with a suction fluid. In larger sizes, the tendency is for the fluid to be discharged in slugs because intimate mixing does not readily occur. This has a detrimental effect on the performance and especially on available discharge head.

Air-Lift Eductors Air-lift pumps are frequently used for difficult pumping operations. Compressed air is forced into the bottom of a pipe submerged in the liquid to be pumped. The expanding air, as it rises up the pipe, entrains the suction fluid.

If compressed air is not available, it is possible to lift water higher than 34 ft (10 m) with the use of an eductor-air-lift combination. Figure 22 illustrates the suction capacity

TABLE 11 Relative capacities of annular siphons

Capacity of standard 3-in (76.2-mm) siphon, water temperature 100°F (37.8°C), 0 suction lift				
Steam pressure, lb/in² (bar) gage	50 (3.45)	75 (5.17)	100 (6.90)	125 (8.62)
Steam consumption, lb/h (kg/h)	1180 (535)	1620 (735)	2060 (934)	2490 (1130)
Max back pressure, lb/in² (bar) gage at zero flow	12 (0.83)	18 (1.24)	22 (1.52)	35 (2.41)
Suction capacity, gpm (m³/h)	140 (31.8)	130 (29.5)	120 (27.2)	110 (25)
Discharge pressure, lb/in² (bar) gage	5 (0.34)	8 (0.55)	12 (0.83)	30 (2.07)

Relative capacities of standard sizes								
Size, in (mm)	1¼ (31.8)	1½ (38.1)	2 (50.8)	2½ (63.5)	3 (76.2)	4 (102)	6 (152)	8 (203)
Capacity ratio	0.13	0.21	0.30	0.60	1.00	1.85	4.0	7.1

Source: Schutte and Koerting.

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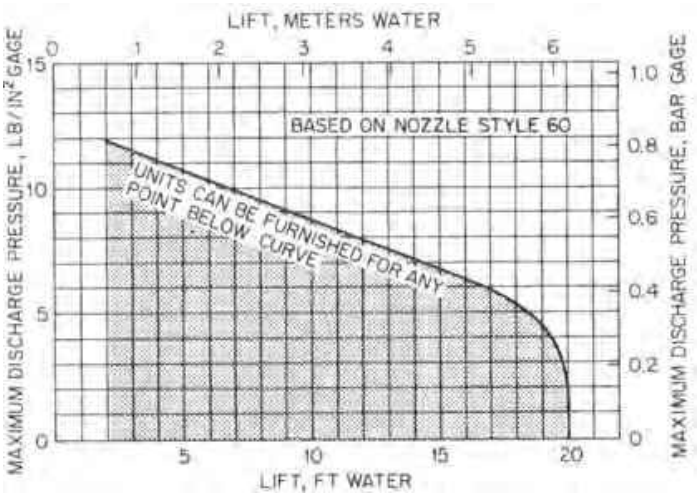


FIGURE 21 Air pumping liquid (Schutte and Koerting)

of a 2-in (51-mm) eductor drawing water from a 50-ft (15-m) depth and discharging it to the atmosphere. In operation, an air line from the atmosphere enters the suction pipe near the water level. As the eductor creates a vacuum in this line, atmospheric pressure forces air into the suction pipe. After it is in the line, the rising air carries the suction fluid to the surface and both fluids are discharged to the atmosphere through the eductor.

No sizing data are presented because this type of pump is best specified according to specific conditions.

Boiler Injectors The boiler injector is a jet pump utilizing steam as a motive fluid to entrain water, and it is used as a boiler feedwater heater and pump. It differs from a siphon in that the discharge pressure is higher than either motive or suction pressure. This is achieved by the double-tube design shown in Figure 23. In operation, the lower nozzle is activated by pulling the handle partway back. The lower jet creates a vacuum in the chamber, causing water to be induced into the unit. When water is spilling out the overflow, the handle is drawn back all the way. This closes the overflow and simultaneously admits motive steam to the upper jet. This second jet, which is of the straight or forcing type, picks up the discharge from the first jet and imparts a velocity to the water

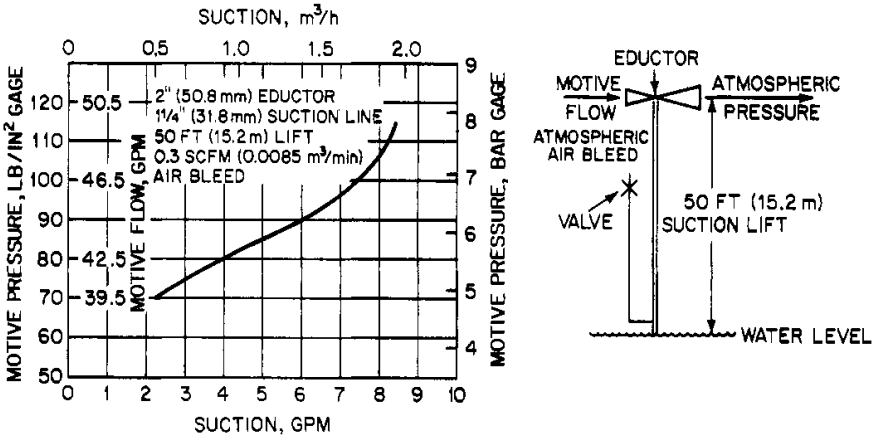


FIGURE 22 Suction capacity of eductor-air-lift combination (gpm × 0.227 = m³/h)

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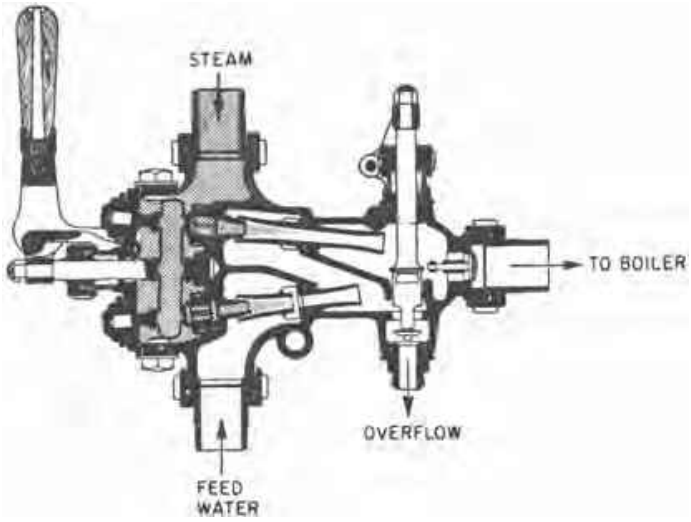


FIGURE 23 Boiler injector, starting position (Schutte and Koerting)

through the discharge tube. The energy contained is sufficient to open the check valve and discharge against the boiler pressure.

The now obsolete steam locomotives were the largest users of this type of injector. Principal use at present is as a backup to a regular boiler-feed pump. Capacity Table 12 illustrates the range of capacities available for the double-tube injector.

TABLE 12 Capacities of boiler injectors, gph[a]

Size no.	Size iron pipe conn., in (mm)	Size copper pipe OD, in (mm)	Size overflow (drip funnel) pipe, in (mm)	Steam pressure, lb/in² (bar)				
				50 (3.45)	100 (6.90)	150 (10.34)	200 (13.79)	250 (17.24)
0	¼ (12.7)	⅜ (9.53)	¼ (12.7)	80	100	110	100	90
1	⅜ (9.53)	½ (12.7)	¼ (12.7)	110	140	180	140	130
2	½ (12.7)	⅝ (15.9)	⅜ (9.33)	170	210	230	200	190
3	¾ (19.1)	⅞ (22.2)	½ (12.7)	280	340	400	340	320
3½	¾ (19.1)	⅞ (22.2)	½ (12.7)	400	470	550	470	440
4	1 (25.4)	1⅛ (28.6)	¾ (19.1)	530	620	720	620	590
5	1¼ (31.8)	1½ (38.1)	1 (25.4)	680	800	920	800	750
6	1¼ (31.8)	1½ (38.1)	1 (25.4)	820	990	1130	990	930
7	1½ (38.1)	1¾ (44.4)	1¼ (31.8)	1070	1370	1610	1370	1290
8	1½ (38.1)	1¾ (44.4)	1 (31.8)	1400	1800	2100	1800	1700
9	2 (50.8)	2¼ (57.2)	1½ (38.1)	1700	2100	2500	2100	2000
10	2 (50.8)	2¼ (57.2)	1½ (38.1)	2000	2500	2900	2500	2300
11	2½ (63.5)	2¾ (69.8)	2 (50.8)	2500	3000	3500	3000	2800
12	2½ (63.5)	2¾ (69.8)	2 (30.8)	3000	3600	4300	3600	3400
14	3 (76.2)	3¼ (82.6)	2½ (63.5)	3900	4600	5500	4600	4400
16	3 (76.2)	3¼ (82.6)	2½ (63.5)	5000	6000	7000	6000	5700

[a]gph × 0.00379 = m³/h

Source: Schutte and Koerting.

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C • H • A • P • T • E • R • 8

MATERIALS OF CONSTRUCTION

SECTION 8.1
METALLIC MATERIALS AND DAMAGE MECHANISMS

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A successful pump installation depends on how well it satisfies the requirements of *performance* and *life*. Performance is the rating of the pump head, capacity, and efficiency. Life is the total number of hours of operation before one or more pump components must be replaced to maintain an acceptable performance. The initial performance is the responsibility of the pump manufacturer and is inherent in the pump design. Life is primarily a measure of the resistance of the materials of construction to corrosion, erosion, wear, and other factors that can influence the materials when the pump has been placed in service. The need to maximize reliability and extend the pump life makes the selection of appropriate materials of construction crucial.

The selection of materials that are both cost-effective and technically suitable for the application requires a knowledge not only of the pump design and manufacturing processes, but also of the engineering properties of the material, particularly its corrosion and wear resistance properties when subjected to the conditions encountered in the pump. Sufficient information is available in the corrosion and metallurgical literature as well as from the experience of pump manufacturers to make appropriate material choices for virtually any pumping application.

It is known that several factors lead to a long pump life. These include:

- Neutral liquids at near-ambient temperatures
- Appropriate material selections for pumps in aggressive services
- The absence of abrasive particles
- Continuous operation at or near the maximum efficiency capacity of the pump
- An adequate margin of available NPSH over NPSH required as stated on the manufacturer's rating curve
- A low velocity (developed head/rotative speed)

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Pumping installations that satisfy all these criteria should have a long life. A typical example would be a waterworks pump. Some waterworks pumps with bronze impellers and cast-iron casings have a life of 50 years or more. At the other extreme might be a chemical pump handling a hot corrosive liquid with abrasive particles carried in suspension. The life of this pump might be measured in months rather than in years, despite the fact that construction was based on the most resistant materials available.

Most pumping applications fall somewhere between these two extremes. The pump designer needs to be familiar with the various types of degradation that can affect the components of the pump and reduce its useful life. These can be grouped into the general categories of corrosion, wear, and fatigue, with corrosion and wear being the predominant life-limiting mechanisms.

TYPES OF CORROSION

General Corrosion General corrosion is corrosion that proceeds without an appreciable localization of attack. This type of corrosion occurs on metals or alloys that do not develop an effective passive film on the surface. Usually, the corrosion mechanism is oxidation with the formation of metal oxide corrosion products. General corrosion is most often encountered in pumps with carbon steels and copper-based alloys. Cast irons also experience a specialized form of general corrosion, known as *graphitic corrosion,* which will be considered separately.

Carbon steel does not develop a protective oxide film and will corrode at a rate dependent upon several characteristics of the water or other fluid, including temperature, oxygen content, pH, and fluid chemistry. Several empirical indices based on water chemistry exist and can be used to calculate the relative corrosivity of natural waters to carbon steel and similar ferrous alloys. The Langelier Index is best known. The rate of corrosion is also very dependent on velocity and increases with an increasing velocity. In most pump applications, with the notable exception of hydrocarbons, the corrosion rate of carbon steel is too high for this material to provide a useful life. However, carbon steel is frequently used, particularly in vertical pumps, with some form of protective coating to prevent corrosion. Coal tar epoxy is a preferred coating for many water services, with the notable exception of potable water, where there are health-related concerns.

Copper alloys, including both brasses and bronzes, are also subject to general corrosion in the water applications where they are most commonly used in the pump industry. The corrosion rate will be increased by the presence of small amounts of sulfides in the water. Copper alloys gradually develop a protective copper oxide corrosion film in most applications. The corrosion rate gradually decreases over time as this film develops. The rate of general corrosion varies with the specific type or grade of copper alloy. Among the alloys commonly used in pumps, nickel aluminum bronzes have the lowest corrosion rate and best tolerance for higher velocities.

The general corrosion of a Ductile Ni-Resist casing from a vertical pump is shown in Figure 1. A metallographic cross section was removed to show the depth of the corrosion attack.

Localized Corrosion There are generally considered to be two types of localized corrosion, pitting and crevice corrosion. Localized corrosion is most commonly found in materials that tend to develop a relatively stable oxide film, such as stainless steels, nickel-based alloys, and aluminum alloys. Pitting, or crevice corrosion, will develop at microscopic flaws or discontinuities in the oxide film. Some corrodent, usually chloride or another halogen ion, initiates the corrosion. Very often, a microgalvanic couple is set up between the interior of the pit and the external unattacked surface. The interior contains acidic, hydrolyzed salts, which are quite corrosive in comparison with the bulk solution. An anode is established within the pit, with the surrounding surface acting as the cathodes. This is particularly the case for alloys that rely on a resistant, passive film for protection, such as stainless steel titanium, and aluminum.

Stainless steels are extensively used in pumps. Localized corrosion is the most common type of corrosion experienced by these materials. Attack by aqueous chlorides is the usual

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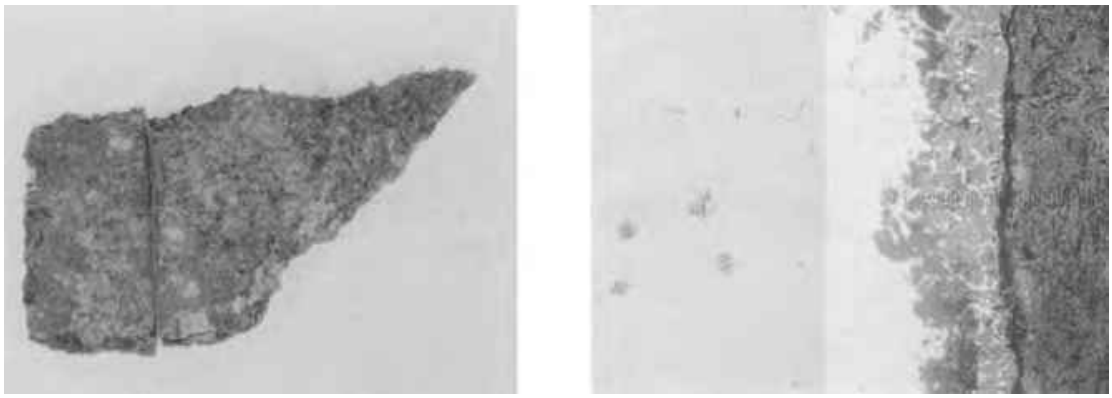


FIGURE 1 A small fragment of Ductile Ni-Resist from the lower casing of a vertical pump. The microstructure is also shown on the right side of this figure, illustrating the depth of the corrosion's penetration. This is a classic example of general corrosion (right photo at 100×).

cause for the corrosion. Susceptibility to localized corrosion is directly related to the alloy content of these steels. An empirical relationship, known as the pitting resistance equivalent (PRE), has been developed that relates the pitting resistance in seawater to the chemical composition of the stainless steel by the following relationship:

$$PRE = \%Cr + 3.3\%Mo + 15\%N$$

where Cr = chromium
Mo = molybdenum
N = nickel

This formula is applied primarily to austenitic and duplex grades of stainless steel. Test data and field experience indicate that stainless steels with a PRE of 40 or higher will be fully resistant to pitting corrosion in seawater at ambient temperature.

Crevice corrosion occurs in constricted areas where free access to the surrounding liquid is restricted. Among the more common locations for crevice corrosion to occur in pumps are various types of metal-to-metal surfaces, such as threaded fasteners, couplings, or joints. Crevice corrosion also occurs at metal to nonmetal surfaces, such as under gaskets. A specialize form of crevice corrosion is deposit attack, which suggests that the crevice has been formed by a discontinuous deposit of a foreign substance from the environment adhering to the surface.

There are several commonly used methods for evaluating the resistance of alloys to crevice corrosion. The ASTM G78 standard describes a method for conducting crevice corrosion tests, and identifies factors that may affect results and influence conclusions. This method uses segmented nonmetallic washers, having a series of grooves and plateaus, which are attached to a corrosion coupon to create a multiple crevice assembly. Samples are evaluated based on the number of sites showing initiation of crevice corrosion, and the depth of this corrosion. The results can be used to determine a crevice corrosion index, which indicates the relative resistance of the alloy to crevice corrosion.

Another technique for determining resistance to crevice corrosion is determination of the critical crevice temperature (CCT). This is the temperature at which crevice corrosion initiates in a ferric chloride solution in a precisely controlled twenty-four hour test. The results of this test correlate well with crevice corrosion resistance in aerated seawater.

Dealloying Dealloying is the preferential removal of one phase from a multiphase alloy, or one element from a material. Several types of dealloying occur in the pump industry. One of the most common is the graphitic corrosion of gray cast iron. This material is low cost, easy to machine, and well suited for a variety of applications, especially in the waterworks industry. It is probably the most widely used material in the pump industry.

Gray cast iron corrodes by a fundamentally different mechanism than carbon steel or ductile cast iron. The structure of gray cast iron consists of interconnected graphite flakes

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FIGURE 2 The interface between the advancing graphitized front and the sound base metal. Graphitic corrosion propagates along the path of the graphite flakes (50×).

in a matrix that is predominantly iron. In the presence of an electrolyte, which is usually water, a galvanic cell is established between the iron and graphite. The iron corrodes, and the corrosion products are largely flushed away with the fluid passing through the pump. The original casting is gradually reduced to a porous graphite structure that may contain some iron oxide corrosion product. This is frequently referred to as *graphitization.* The surface of a gray iron casting that has suffered graphitic corrosion will retain its original shape and dimensions, but the surface will be largely graphite, which can be cut with a knife. The casting will lose some fraction of its mechanical properties and become increasingly susceptible to brittle failure, resulting from modest shock or impact loads. This is also the corrosion mechanism for Ni-Resist in seawater. Figure 2 shows the interface between the sound base metal and the graphitized surface layer.

It is important to recognize that the rate of graphitic corrosion varies with the water chemistry, and that this type of corrosion can occur in both fresh and salt waters. The high conductivity of salt water corresponds to a higher corrosion rate. Graphitic corrosion will proceed at a slower pace in waters that have a high mineral content. Minerals tend to plug the graphitic layer on the surface, sealing off the base metal from exposure to the fluid, thereby reducing the corrosion rate.

As the surface of a cast-iron component, such as a pump casing, gradually graphitizes, the galvanic relationships with other components within the pump will be altered. It has been observed that the bronze impeller originally supplied in a cast-iron pump handling seawater will provide a significantly longer life than bronze impellers that are installed after the pump has been in service for several years. The reduced life of the replacement impellers is caused by an altered galvanic relationship with the pump casing. Initially, the casing was cast iron, which is anodic to a bronze impeller. With time, as the casing graphitizes, it gradually becomes cathodic, due to the influence of the graphite. The bronze impeller is now the anode and corrodes at a much higher rate. This example highlights the influence that graphitic corrosion can have on other components within the pump and the importance of carefully selecting materials for use in conductive fluids, such as saltwater.

Several other types of dealloying can also occur in pumps. Brass and bronze alloys containing more than about 14% zinc are subject to a form of dealloying known as *dezincification.* The zinc is preferentially corroded from the matrix of the material, leaving a spongy, copper-rich residue. Dezincification can occur either uniformly in a shallow layer over the surface of the casting or as a distinct plug confined to a small area. Plug-type de-zincification is a more serious problem because the plug is weak and will cause leakage if it penetrates a pressure boundary. It should be emphasized that copper alloys containing less than 14% zinc are not susceptible to this form of corrosion. Consequently, the requirement often imposed upon pump manufacturers for zinc-free bronzes to avoid dezincification is without technical justification. Figure 3 shows the dealloying of an impeller.

The final type of dealloying that occasionally occurs in pumps is dealuminification in aluminum bronzes. These are metallurgically complex materials. Some compositions can form an aluminum-rich phase that can be preferentially corroded in aggressive fluids.

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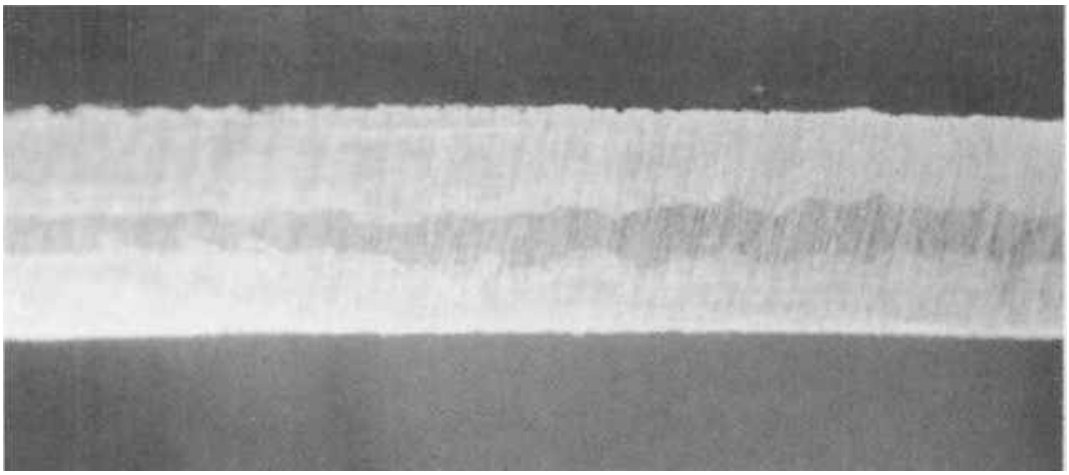


FIGURE 3 The dealloying of a vertical turbine pump impeller. Note the change in color across the cross section. The unaffected bronze (light color) material is surrounded by a dezincified layer (1.3×).

especially seawater. The detrimental phase can be mitigated by a special heat treatment known as *temper annealing*. This heat treatment must be specified by the designer for susceptible compositions, because it is not a mandatory requirement of national material specifications. The chemistry of some aluminum bronze alloys from Europe has been adjusted to preclude the formation of the detrimental aluminum-rich phase without the need for the temper annealing heat treatment. The temper anneal can serve as a stress-relief operation for fabricated aluminum bronze structures, which is a secondary benefit for products in this category.

Galvanic Corrosion *Galvanic corrosion* refers to the corrosion that occurs when one alloy is electrically coupled to another and exposed in a conductive liquid. Usually, the corrosion rate of the more noble alloy will be less than if it were exposed uncoupled. The corrosion rate of the less noble material will be greater than if it were exposed uncoupled.

Several factors influence the rate of galvanic corrosion of both metals. This corrosion is greatly influenced by the conductivity of the fluid. In a fluid such as freshwater, which has a low conductivity, galvanic corrosion will be less severe and generally confined to the immediate location where the metals contact one another. However, in a highly conductive fluid, such as seawater, galvanic corrosion will be more severe and will occur over a wider area. The pump designer needs to consider the possibility of such corrosion when using dissimilar metals in a conductive fluid.

Galvanic corrosion problems in seawater and other conductive fluids can be avoided by the careful use of materials. Galvanic corrosion is related to the area ratios of the coupled metals. It is always desirable to have the area of the anode, or less noble metal, equal to or greater than that of the more noble metal. In this way, the additional corrosion experienced by the less noble metal will be spread over a relatively large area and will not be excessive because of being coupled. An example of the effective use of this galvanic relationship involves centrifugal pumps having a Ni-Resist casing and austenitic stainless steel internals. This combination is often specified for seawater services. The Ni-Resist is anodic to the stainless steel and will protect it from localized corrosion when the pump is shut down and contains stagnant water. The area of Ni-Resist is considerably larger than that of stainless steel. The increased galvanic corrosion of the Ni-Resist is spread over a large area and is negligible.

The amount of corrosion that will occur in a galvanic couple also depends on the freely corroding potentials of the coupled metals. Less corrosion-resistant metals, such as zinc, cast iron, and steel will usually have more negative potentials when measured against a standard reference electrode. More corrosion-resistant metals, such as stainless steels, will have less negative potentials.

The corrosion potentials for many commonly used engineering alloys in slowly moving seawater are shown in Table 1. The alloys are listed in the order of the potential that they

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TABLE 1 Corrosion potentials in flowing seawater (8 to 13 ft/s, 50 to 80°F/2.4 to 4.0 m/s, 10 to 26°C)

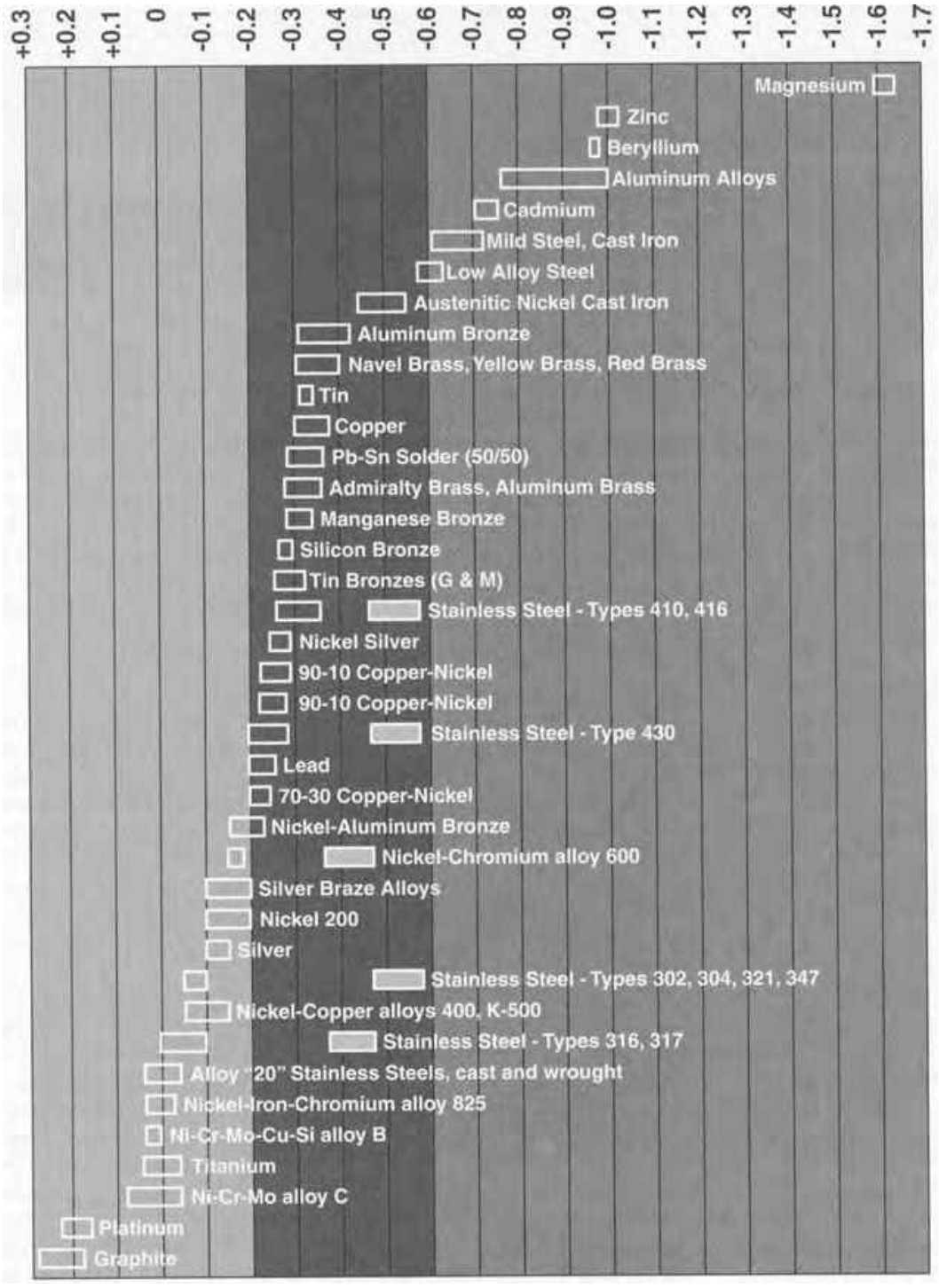


exhibit in flowing seawater. Certain alloys (indicated by solid colored boxes preceding the name of the alloy) in low-velocity or poorly aerated water and at shielded areas may become active and exhibit a potential near – 0.5 volts. The extent of galvanic corrosion that will occur when two metals are electrically coupled will depend on the potential difference between the metals. The corrosion rate of zinc coupled to stainless steel will increase

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dramatically because of the large potential difference between these two metals. A nickel aluminum bronze coupled to austenitic stainless steel will experience little galvanic corrosion because the potentials of these two metals are close to one another. The pump designer needs to be aware of the corrosion potentials of dissimilar metals used in conductive fluids in order to avoid unanticipated galvanic corrosion problems.

The use of coatings can decisively alter the galvanic relationships in a pump. If the more anodic component, such as a steel casing, is coated, one can expect a high rate of corrosion at those locations where the coating eventually begins to fail. This will be caused by a very unfavorable area ratio, with a small area of exposed carbon steel coupled to a large area of some more noble metal, such as stainless steel or bronze. Experience has shown that this type of galvanic corrosion problem can occur even in relatively low conductivity fluids, such as freshwater. For this reason, coatings should be employed with caution in pumps handling conductive fluids that are constructed of dissimilar metals. It is generally advisable in these applications not to coat the anodic component, usually the casing. If coatings are used, periodic maintenance will be necessary to avoid galvanic corrosion. Figure 4 documents the galvanic corrosion on the interior diameter of a carbon steel flange connected to a stainless steel shroud. The accelerated corrosion is due to the unfavorable ratio of stainless steel to carbon steel in this component.

Stress Corrosion Cracking *Stress corrosion cracking* (SCC) is a particularly dangerous form of corrosion because it is not easily detected before it has progressed to such an extent that it can cause sudden catastrophic damage. Although relatively uncommon in pumps, it can occur in several classes of materials. The pump designer should be aware of the potential combinations of material and environment that can cause SCC.

Stress corrosion requires that several factors be present. These include tensile stress, which can be either residual or applied, a susceptible material, an environment capable of causing stress corrosion, and time.

The materials used in the pump industry that may experience SCC include austenitic and martensitic stainless steels, some copper-based alloys, and, occasionally, Ni-Resist. The austenitic stainless steels are susceptible to stress corrosion in aqueous chlorides at temperatures above about 140°F (60°C). Cast alloys, which contain some fraction of ferrite in the microstructure, are significantly more resistant to stress corrosion than their wrought counterparts. The possibility of cracking is increased in situations where chlorides are concentrated, as by evaporation. High residual stress, often present in as-welded



FIGURE 4 Galvanic corrosion is evident on this pump section. Note the high corrosion rate on the interior diameter of the carbon steel flange that is attached to the stainless steel shroud.

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structures, also enhances the possibility of cracking. Increasing nickel content in austenitic stainless alloys enhances the resistance to SCC. The high nickel grade, commonly known as Alloy 20, is often used in chemical applications where the optimum resistance to stress corrosion is necessary. The SCC of austenitic stainless steels in pumps is relatively uncommon.

Martensitic stainless steels are susceptible to cracking in the presence of hydrogen sulfide and is often referred to as *sulfide stress corrosion cracking* (SSC). These steels, particularly CA-15 and CA-6NM, are commonly used in pumping applications in oil production and refining where hydrogen sulfide can be present. SCC can be avoided by giving these materials a special heat treatment intended to reduce hardness below a certain threshold level, below which cracking will not occur. This has also been correlated to the yield strength of a material. It is often seen in literature that ferrous materials used in these services should have a hardness no greater than 22 R_c or a yield strength no higher than 90,000 lb/in^2 (620 MPa). Technical standards, including API 610 and NACE MR-01-75, can be used to specify appropriate requirements for martensitic steels, which will be used in environments containing hydrogen sulfide.

Copper alloys are susceptible to SCC in the presence of ammonia, although considerable variations take place in the susceptibility of the various types of bronzes, with aluminum bronzes being the most resistant. Polluted natural waters can contain ammonia, and for this reason, bronze pumps are usually not a good choice for these applications.

High-strength manganese bronzes are susceptible to cracking in natural waters. Cast impellers in these alloys have been known to suffer severe cracking. Residual stress in the casting may also be sufficient to induce cracking. These alloys should not be used in pumps because of their susceptibility to such problems.

Ni-Resist is an austenitic cast iron that contains 15 to 20% nickel. This material is commonly used in large, seawater vertical pumps. Experience has shown that it is subject to SCC, especially in the diffuser section of these pumps, unless the castings are furnace stress-relieved. This must be specified by the purchaser, as it is not a requirement of national material specifications.

Hydrogen Embrittlement Hydrogen damage is a form of environmentally assisted failure that results from the combined action of hydrogen and residual or applied tensile stress. Hydrogen damage to specific alloys or groups of alloys manifests itself in many ways, such as cracking, blistering, hydriding, or as a loss of tensile ductility. Collectively, these various forms of damage are often referred to as *hydrogen embrittlement.*

Damage caused by hydrogen is occasionally encountered in pumps. Some plating processes, such as chrome plating, which is often used to rebuild pump shafts, generate hydrogen. This hydrogen can enter the surface of the metal. Microscopic cracks can occur in higher strength steels (greater than a 90,000-lb/in^2 or 620-MPa yield strength). Abusive grinding can work-harden the surface of lower strength steels and increase the probability that hydrogen will cause cracking. Microscopic cracks resulting from hydrogen damage act as stress risers and can propagate failure catastrophically by mechanical fatigue. This problem can be avoided by utilizing proper grinding practices before plating. Higher strength steels should be baked, to drive off hydrogen, immediately after plating.

Hydrogen can also be introduced into metals during welding. In order to avoid the hydrogen damage associated with welding, ferritic and martensitic steels should be welded with low hydrogen electrodes. Coated electrodes should be baked, in accordance with manufacturer's instructions, prior to usage in order to drive off moisture, which is the major source of hydrogen contamination of welds.

Microbiologically Induced Corrosion Living organisms can promote corrosion in many different environments. A variety of biological organisms thrive in both aerobic and anaerobic environments. Corrosion attributable to microbiological activity occurs most frequently in stagnant water, which remains in a pump when it is shut down for an extended length of time.

Sulfate-reducing bacteria are found in many waters. They will form slimy, reddish, hemispherical-shaped mounds or colonies on cast iron or carbon steel. These are known as *tubercles.* If scraped off, there will invariably be a saucer-shaped pit beneath the tubercle.

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The inside of the pit will contain a wet, black deposit. The pitting is caused by traces of sulfuric acid excreted by the bacteria. This type of corrosion will usually not result in premature failure.

Several more serious types of microbiologically induced corrosion afflict stainless steels. A certain class of metal ion concentrating/oxidizing microbes appears to concentrate ferric and manganic chlorides, both of which are potent pitting agents. These bacteria form colonies preferentially at welds in austenitic stainless steels and are capable of causing severe pitting corrosion in a relatively short time. This problem has been encountered in a variety of equipment in both salt and freshwater. It is often discovered only when the welds begin leaking. Pumps employing welded stainless steel fabrications can be afflicted by this problem if permitted to sit idle with stagnant water, either fresh or salt, for an extended period. Biocides can be used to mitigate this problem in some instances.

Finally, the decay of biological organisms can generate hydrogen sulfide, which adversely affects the protective oxide film on copper-based alloys. The enhanced biological activity in warmer tropical waters, especially under stagnant conditions, can impair the corrosion resistance of bronzes and reduce the threshold velocity at which accelerated corrosion will occur. Bronzes should be used with caution in applications where macrobiological activity is anticipated and the possibility of extended shutdowns is possible.

Stray Current Corrosion This type of corrosion is sometimes assumed, incorrectly, to be synonymous with galvanic corrosion. In fact, while the resulting corrosion can have some similarities, the cause for stray current corrosion is much different. Stray current corrosion is caused by an extraneous source of dc current passing through the pump. The most frequent source is some defect in the motor, which allows current to escape, pass through the pump shaft, and exit into the fluid. Other possible sources of stray current include defective or poorly designed cathodic protection systems, and grounding of a heavy-duty electrical device, such as a welding torch, through a pump. In the special case of submersible pumps, there have been instances where the insulation on the power cable is cut or damaged, usually during installation of the pump. Current will escape at this location, enter the pump, and cause extensive corrosion. Figure 5 shows a submersible pump bowl perforated in several locations as a result of stray current emanating from a damaged power cable.



FIGURE 5 A submersible pump bowl perforated in several locations as a result of stray current emanating from a damaged power cable

Stray current is often severe. The corrosion occurs at the location where the current exits a metal surface and enters the fluid. The extent of corrosion can be calculated using Faraday's law:

$$\text{Weight of metal reacting} = KIT$$

where I is in amperes, T in seconds, and K is a constant known as the electrical equivalent.

The corrosion often occurs as large, saucer-shaped pits. The base of the pit is smooth and shiny, with no buildup of corrosion products. Stray current should be suspected when severe pitting develops in a pump that is handling a noncorrosive fluid. One should also consider the possibility of stray current corrosion when only one of several pumps develops corrosion, or when corrosion develops in an application where it had not occurred previously, or where there is no other plausible explanation for the corrosion. Although stray current is more often seen in submersible or vertical pumps, it has also been found in horizontal split case pumps. The nominal corrosion resistance of the alloy plays no role in determining the extent of stray current corrosion, which is governed solely by Faraday's law.

Intergranular Corrosion This infrequent type of corrosion preferentially attacks a material at the grain boundaries. This is caused by local chemical differences such as the chrome-depleted regions of an austenitic stainless steel. Bronze alloys susceptible to this type of corrosion include aluminum brasses, silicon bronzes, Muntz metal, and admiralty metal. Two things are necessary: a sensitized material and a corrosive media, such as seawater. Sensitization refers to chrome carbide precipitation at the grain boundaries in austenitic stainless steels and can occur during heat treatment or more commonly during weld repair. This type of corrosion often leads to corrosion-assisted fatigue cracks when cyclic loading is present.

The improper heat treatment of 300 series austenitic stainless steels can result in sensitization and subsequent intergranular corrosion. Sensitization occurs when stainless steels that contain more than 0.03% carbon are held at temperatures between 800 and 1550°F (between 425 and 850°C). At these temperatures, chrome carbides precipitate along the grain boundaries, resulting in chrome depletion in the adjacent areas. These adjacent areas have reduced corrosion resistance. Austenitic stainless steels contain approximately 16 to 18% chrome. The chromium content in the areas surrounding a chrome carbide particle can drop below the 12% necessary to maintain a passive state. A galvanic cell is set up with a large cathode (grains) and a small anode (grain boundaries). In this undesirable scenario, corrosion occurs along the anodic grain boundaries. The extent of the corrosion damage depends on the length of time held within the sensitization temperature range. The degree of sensitization is a function of the carbon content; the higher the carbon content, the shorter the period of time the material can be held within this range without sensitization occurring. A graph of the temperature versus time for various carbon contents illustrates this point in Figure 6. Intergranular corrosion of an improperly heat-treated stuffing box cover is shown in Figure 7.

Austenitic stainless steels can also be sensitized during normal welding procedures. Care must be taken to avoid the sensitization range during welding followed by proper postweld heat treatment when necessary.

Sensitization can be avoided or corrected by several methods:

- Heat the material to a temperature high enough to dissolve the chrome carbides, typically 1900 to 2100°F (1040 to 1150°C), followed by rapid cooling through the sensitization range. Localized heat treatment of welded areas will not desensitize a material.
- Use a stainless steel that is stabilized by the addition of niobium or titanium. These two elements will tie up the carbon, thus preventing chrome carbides.
- Reduce the carbon content to a low level (less than 0.03%). The lower the carbon content, the longer it takes chrome carbide precipitation to occur.

Austenitic stainless steel castings should always be supplied in the solution annealed and quenched condition. If low carbon grade material is used, (i.e., CF-3M), postweld heat

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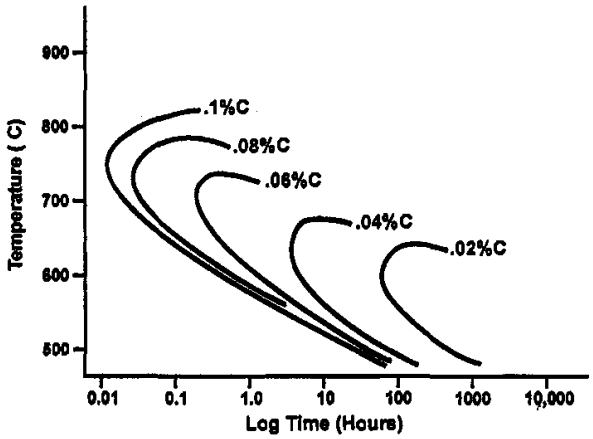


FIGURE 6 Time-temperature sensitization curves as determined by the Strauss test for 18-8 stainless steel. Note that a low carbon grade of stainless (0.03% C) requires 5 to 10 h exposure, while a standard grade (0.08%) needs only minutes of exposure time (°F = 1.8 × °C + 32).

treatment (to eliminate sensitization) will not be necessary following repair welding. This is a significant benefit in the event welding is required on a finish-machined casting.

Cavitation Erosion Cavitation erosion is primarily a mechanical process, although it acts synergistically with corrosion and is often considered with other forms of corrosion. Cavitation erosion can be defined as metal removal from the surface caused by high stresses associated with the collapse of vapor bubbles in the fluid. Cavitation occurs in a pump when the local pressure of the fluid is reduced to the vapor pressure. In a multistage pump, vapor bubbles form in the low-pressure areas at the impeller inlet and are swept by the flow into regions of higher pressure where they collapse. A great many bubbles may form and collapse in a small area, producing many microjets of high kinetic energy. The energy released by the bubble collapse is expended as impact loading on the metal surface. This situation is aggravated if protective oxide films are present because these are damaged, exposing fresh metal to the corrosive action of the fluid. This cyclic loading eventually causes the formation of microscopic fatigue cracks. These cracks propagate and intersect, resulting in the removal of metal from the surface and the characteristic



FIGURE 7 The surface of a stuffing box cover that experienced intergranular corrosion due to sensitization. The grains are clearly evident on the interior of the bore as well.

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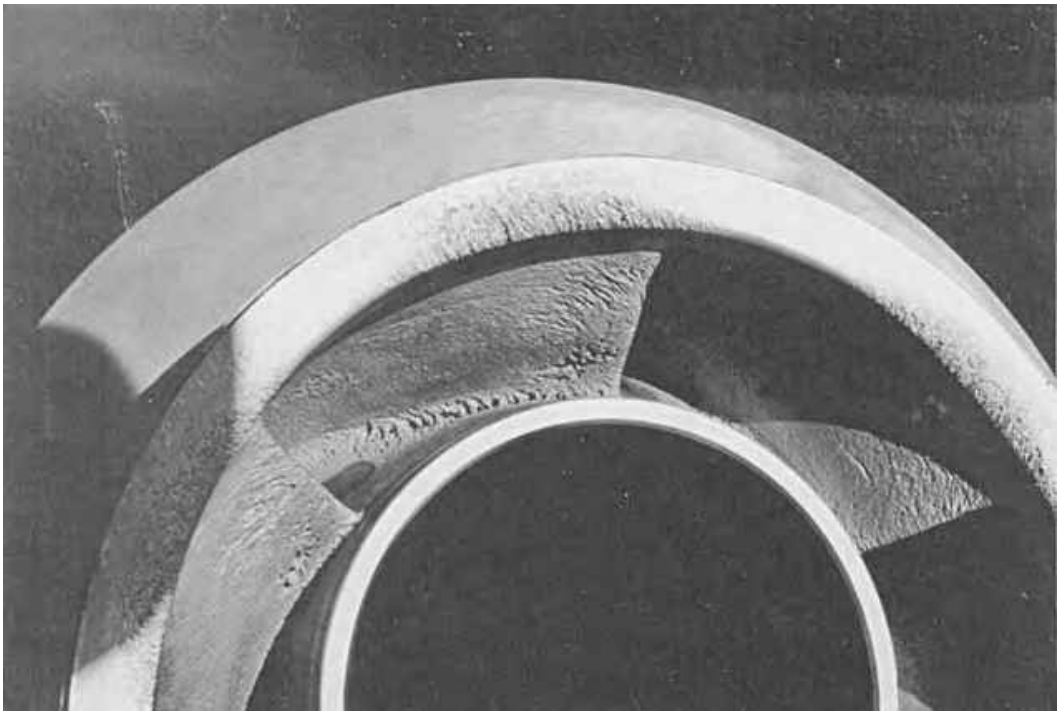


FIGURE 8 Cavitation erosion of an impeller, indicated by the porous appearance of cavitated regions on the surface

spongy or porous appearance of cavitation damage. An example of a cavitated impeller is shown in Figure 8.

It was conventional wisdom in the pump industry until recent years that the cavitation resistance of a material was directly related to its hardness. A more sophisticated understanding has been developed in recent years, leading to the development of a new class of nonstandard stainless steels with exceptional cavitation resistance.

The relationship between cavitation resistance and hardness was first critically investigated in the 1970s when it was observed that cobalt-based alloys of a modest hardness developed a very high resistance to cavitation damage. Cavitation resistance was related to the capability of the material to transform at the surface when subject to cavitation loading into a harder, more resistant metallurgical phase. This work was extended to austenitic stainless steels, whose chemical composition was adjusted to promote the formation of a stress-induced martensite under cavitation loading. New alloys were developed initially as weld filler metals to repair cavitation damage and later as impeller castings for pumps. These alloys have relatively low hardness in the solution-annealed condition, comparable to standard austenitic grades, but transform to a much harder martensite at the surface upon exposure to cavitation loading. The hard surface layer resists the initiation of fatigue cracks. If these cracks eventually develop after extended exposure to cavitation bubbles, propagation into the soft ductile base metal is difficult. Cavitation-resistant austenitic stainless steel castings, alloyed with chrome and manganese, develop cavitation resistance similar to that of cobalt-based alloys.

Extensive laboratory tests of the resistance of a wide range of materials to cavitation erosion have produced data for all the materials commonly used in centrifugal pump construction. It is possible to make a good correlation between the laboratory data and field experience to develop the following tabulation of the cavitation-resistance properties of pump materials, listed in order of decreasing cavitation resistance:

- Stellite
- Chrome-manganese austenitic stainless
- Carburized 12% chrome stainless casting
- Titanium 6AL-4V

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- Cast nickel-aluminum bronze
- Cast duplex stainless steel
- Cast precipitation hardening stainless steel
- Ductile Ni-Resist
- Cast CF-8M
- Cast CA6-NM
- Cast CA-15
- Monel
- Manganese bronze
- Carbon steel (cast)
- Leaded bronze
- Cast iron

Selecting materials with adequate cavitation resistance will afford the pump designer much greater leeway in the range of conditions under which the pump can be operated. It also permits the design of smaller, lighter pumps that can be operated at higher speeds. The judicious use of materials significantly extends the time between outages caused by cavitation damage and can dramatically reduce maintenance costs.

TYPES OF WEAR

Rotating equipment, including pumps, can suffer from damage as a result of mechanisms unrelated to corrosion. The relative motion between parts that are in close proximity to each other can produce wear when these components come into contact with one another.

Catastrophic damage may occur if the parts make contact under high loading conditions or when foreign bodies are entrapped between the rotating and stationary components. An accelerated material loss or catastrophic seizure of these components can result in costly repairs or replacements. Erosion, due to the presence of solid particles in the liquid being pumped, can also limit the life of internal pump components.

Wear mechanisms have been categorized into more than 20 individual processes.[1] However, only a few mechanisms are frequently recognized as damaging to a pump:

- Adhesive wear: material-to-material contact
- Abrasive wear: solids interacting with internal components
- Erosion: solid particle impingement
- Fretting: small amplitude motion of parts causing oxidation damage

Identifying the wear mechanism is somewhat difficult at times as wear, or the loss of material, within a pump can result from more than one mechanism at a time.

The study of friction and wear as a science, known as *tribology,* had its beginning in the late 1930s. These early studies fostered an increased awareness of wear damage mechanisms that, in addition to corrosion and material fatigue, account for the life-limiting factors of pumps. Additional information on the study of wear can be found in current trade journals and texts.

Adhesive Wear One of the primary causes of material loss on rotating components in a pump handling clear liquids (with no solids entrained in the fluid stream) is adhesive wear. This material loss is due to metal-to-metal contact producing mechanical damage on the surface, a transfer of material, and eventually galling, which is considered a severe case of adhesive wear. Two important characteristics to consider for a pair of materials that may come into contact are their adhesive wear traits and their galling threshold.

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The wear of two surfaces in relative motion is complex. Some alternative theories of sliding wear have been proposed in addition to the adhesive wear model. They are the delamination theory, the oxidation theory, the surface delamination theory, a fatigue model, and combinations of several of these theories. However, only the adhesive wear theory offers a general wear equation that quantitatively predicts wear, thus providing a means to rank materials with respect to their wear characteristics.

A multitude of adhesive wear tests exist, including ring and block, pin and vee block, 4-ball, and pin on disk. Wear tests are designed to simulate, as closely as possible, the actual service conditions and parameters, and are useful for developing a quantitative ranking of various alloys.

Wear testing and many years of field experience in pumping applications has led to several guidelines for the selection of materials for close clearance rotating and stationary components that will have adequate resistance to adhesive wear.

1. Austenitic stainless steels have poor galling resistance. Wide clearances are advisable, unless one of the two components is Nitronic 60, a special high manganese austenitic grade with antigalling characteristics.
2. Martensitic stainless steels will also have poor galling resistance. These are heat treatable materials. If heat treated to a minimum hardness difference of 50 BHN, galling resistance will be adequate. Also, if both parts are heat treated to a minimum hardness of 400 BHN, galling resistance will be acceptable.
3. Bronzes have reasonably good galling resistance, especially those containing several percent lead, which acts as a lubricant.
4. Cast iron also has good antigalling characteristics, with the graphite in the material promoting lubricity.
5. Several nonmetallics, including graphite composites and reinforced polyetheretherketone (PEEK) are excellent antigalling alloys, and useful for low specific gravity fluids.

Based upon extensive empirical testing and field experiences, several sound rules of thumb have been developed through the years when selecting pump wear ring materials. Three factors are used to select materials for wear surfaces in clear liquid environments:

- Corrosiveness of the fluid
- Amount of wear allowed
- Galling stress

Corrosion determines the class of material to be used. These classes generally fall into three groupings: noncorrosive, mildly corrosive, and corrosive. Of course, additional constraints occur when selecting an appropriate material within the corrosive material grouping that will need to be addressed by application experience.

A general guide for selection of suitable antigalling material combinations in several environments is as follows:

Environment	Materials	Hardness
Noncorrosive	Cast iron/leaded bronze	Unimportant
Mildly corrosive	Martensitic stainless steels (locally or through hardened)	Less than 400 BHN, 50-point differential
Corrosive	Corrosion-resistant, nongalling austenitic stainless steel (Nitronic 50/Nitronic 60 or Waukesha 88/Nitronic 50)	Greater than 400 BHN, same hardness acceptable
Severely corrosive	Highly alloyed austenitic stainless steel with hard-faced materials such as stellite or colmonoy	Not applicable

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Using these industry-wide accepted rules of thumb will help avoid catastrophic damage normally resulting in costly repairs.

Some applications are particularly susceptible to adhesive wear problems and require special materials to resist wear. These include low specific gravity applications where the use of mechanical carbon materials is desirable because of the nonlubricating nature of these fluids. Common practice is to make the stationary component metal-filled graphite if the specific gravity is 0.5 or less. Stationary mechanical carbon components are also used in liquid CO_2 services and other potential dry start applications, such as the upper bearing in vertical pumps. Currently, nonmetallic wear components, such as advanced polymers and especially reinforced versions of PEEK are being employed to solve nagging problems in a variety of applications..

Fretting Fretting can be considered a special case of adhesive wear. It occurs when two parts in contact experience a repeated, small amplitude relative motion between close-fitting surfaces such as a loose impeller on a shaft. Researchers have described fretting damage as a four-stage event:[2]

1. Adhesive wear of the asperities on the mating materials
2. Abrasive wear caused by the wear debris produced in step one
3. Abraded particles filling the asperity valleys
4. Elastic contact producing cold working of the surface and micropitting

In a pump, there is the potential for small amplitude motion at loose fitting impellers, beneath loose bearings, and between impeller wear rings and the impeller hub. The design engineer does not intentionally create a circumstance that will generate this type of motion, but when it occurs, fretting damage can lead to other problems.

Fretting can be identified by a red powdery oxide that forms along the fretted surface. In a pump, the red-colored debris is often washed away, but a distinct damaged surface appearance will develop on the fretted surfaces. This damage is often described as having a mottled appearance and is best depicted as a flat, eroded surface with no directionality to the damage. Although the oxide may be washed from the surface, some staining of the adjacent component can be observed after disassembly of the pump. This has led to the misinterpretation that fretting is a corrosion mechanism, but it is actually a special wear phenomenon.

Figure 9 shows the fretting damage of a pump shaft along the impeller-fitted area where a loose fit enables the oscillation of the impeller during operation. Since the motion necessary to cause fretting can be of a small amplitude, large vibrations in the pump may not be present. This makes the detection of fretting during any operation impossible. The impeller in this example would have similar damage along its bore.

Fretting damage can be avoided with a few relatively simple guidelines. One should eliminate or prevent the possibility of motion between the two components by either tighter clearances, or shrink fitting the assembly, which increases the clamping force. If fretting is unavoidable in a particular design, methods of mitigation can be used. These include various coatings or providing the contact zone with an appropriate lubricant. Coatings that may be used include flame-sprayed high-nickel alloys, silver plating, or possibly adding a thin, dense chrome plating to one or both of the faces in contact.

Abrasive Wear Abrasive wear is often categorized into two main classifications: *two-body* and *three-body wear.* The name indicates the mechanism of wear. Usually, three-body abrasive wear is the primary mechanism of damage in centrifugal pumps. This can occur when hard solid particles entrained in the fluid enter between ring fit areas or impeller keyway faces. In fluids with high concentrations of solids, another form of three-body wear is produced. Solids carried in the fluid stream can strike the internal pump surfaces. This is more commonly referred to as *erosion.* This type of damage is observed in the impeller and cutwaters of the casing. The degree of material damage, due to this mechanism, depends upon the bulk hardness of the material, the carbon content, and the characteristics of the solids present. Important particle characteristics include size, shape, hardness, and mass.

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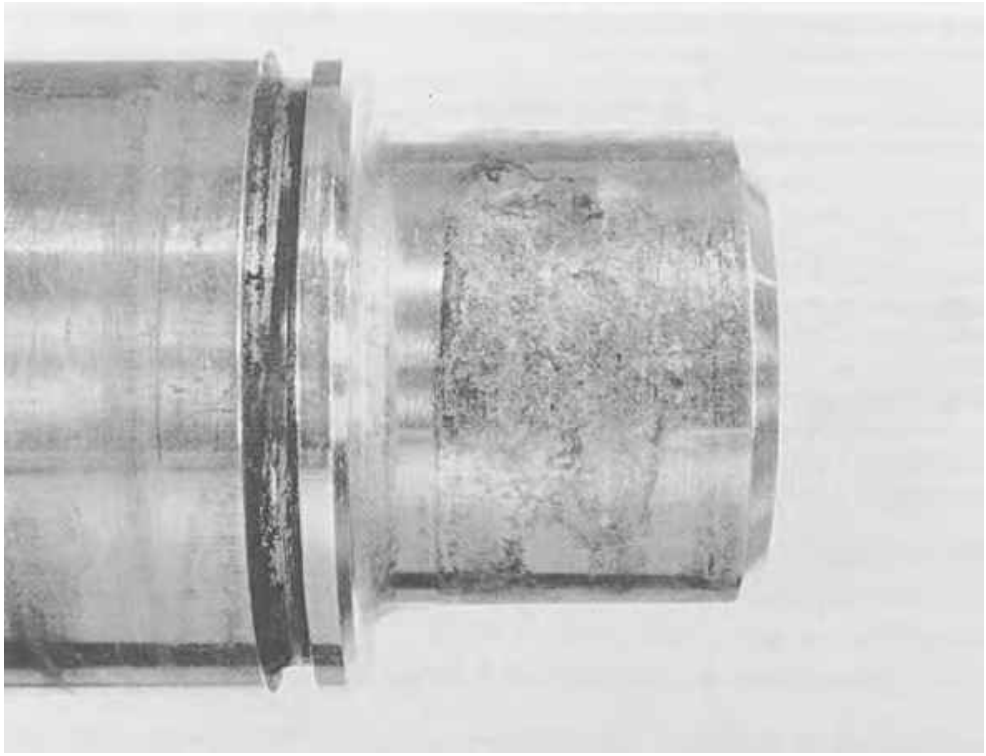


FIGURE 9 The fretting damage of a shaft beneath an impeller that experienced small amplitude motion. The mottled appearance is typical of the damage caused by fretting (2.2×).

To minimize three-body abrasive wear, a couple of variables must be taken into consideration. The wear ring clearance influences damage. The relationship between the size of the particles in the fluid stream and the gap into which they can enter is important. This is graphically illustrated in Figure 10, which shows three types of particle-to-gap relationships. Condition A is logically the most damaging three-body abrasive case. A high rate of damage will result as these particles are entrapped between the two components. In condition B,

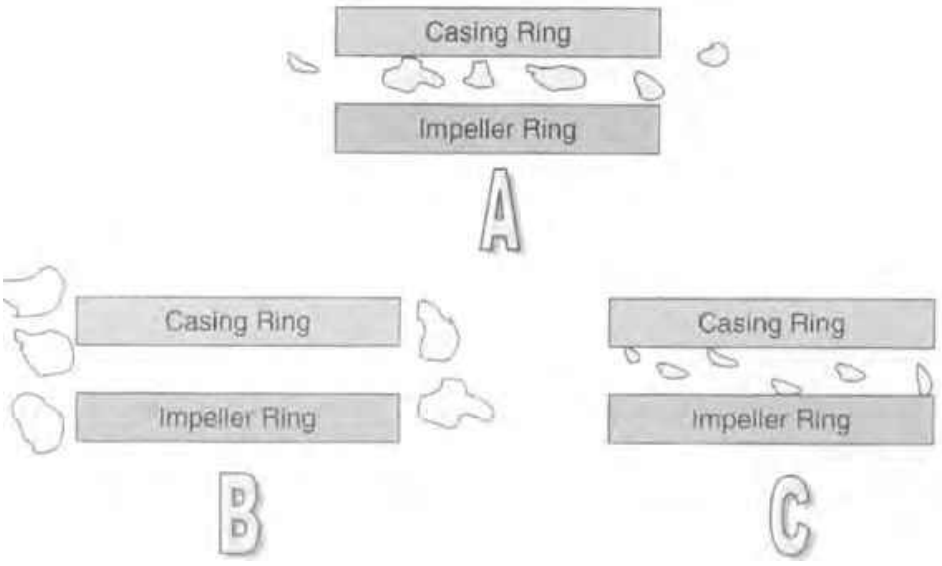


FIGURE 10 Three possible conditions between wear surface clearances and solid particle size. Condition "A" is conducive to maximum three-body abrasive wear.

large particles relative to the ring clearance will not enter and produce damage. This condition enables the particles to flow with the fluid stream through the eye of the impeller and exit the pump. In condition C, very fine or relatively small particles will not be entrapped and ground between the rings and will not result in collateral damage of the components.

Usually, particles in a fluid service will be in a range of sizes, so all the conditions will exist. Typically, a particle size and distribution analysis is performed to characterize the amount of particles that will cause condition A to exist. This is relatively simple to accomplish by extracting the solids from a fluid sample and performing a sieve analysis. The percentage of solids present in the fluid stream is extremely important for determining the appropriate material and design considerations. This will be addressed later with guidelines given for appropriate material selections.

The hardness of entrained solids is also extremely important. If particles are soft and friable, such as talc, little damage would be expected to occur on metal pump components because of three-body abrasive wear. The amount of damage is expected to be greater if the particles are extremely hard. These particles include welding scale or silicon dioxide (SiO_2), which is sand. The particle geometry also contributes to the amount of damage that can result in three-body abrasive wear. Often, particles of SiO_2 are found in a rounded condition. Pumps used to handle river water or seawater on ships frequently encounter these configurations. Hard, round particles are less damaging than particles of equal hardness with sharp, angular configurations. Fly ash, a very hard, sharp, angular particle, is one of the most abrasive services encountered in the pump industry.

A material's resistance to abrasive wear can be characterized by a standard ASTM test procedure. Each testing procedure attempts to simulate the mechanism that most appropriately addresses the class of abrasive wear. In general, materials that are resistant to two-body abrasive wear are resistant to three-body abrasive wear also.

Test results show that the primary property responsible for increasing resistance to abrasive wear is the hardness of a metal alloy. Small microstructural differences, alloying, and surface-condition differences within alloy groups also can influence the abrasion resistance of a material.

An example of three-body abrasive wear is shown in Figure 11. It shows a laser-hardened shaft sleeve after approximately one year of service in a mine dewatering operation where abrasive wear caused a significant wear of other material combinations. The abrasive wear was caused by fine tailings in this gold mine application. To increase the life of rings in services like this, the use of hardened wear rings is recommended. As an alternative to through hardening, pump producers will often use coated rings in applications where significant abrasive wear is anticipated. Depending upon the severity of the service, a coating containing carbides or other hard intermetallic compounds may be necessary.

For mildly abrasive services, the following materials should be considered:

- Ni-Resist—Its resistance is due to chromium carbides in the matrix. It has good adhesive wear resistance also.
- Selectively hardening the surface of AISI 420 (laser hardened 50 to 55 *Rc*). Surface hardening is not susceptible to hydrogen embrittlement or SCC.
- Carburized and hardened 12% chromium stainless steel.

For more abrasive services, the following is often considered:

- Hardened AISI-440C (50 to 55 *Rc*)
- Stellite or colmonoy-coated (hard-faced) austenitic stainless steel
- Solid stellite
- Tungsten carbide
- Silicon carbide
- Partially stabilized zirconia (PSZ)



FIGURE 11 The three-body abrasive wear of a laser-hardened shaft sleeve in an abrasive service. Note the fine concentric scoring of the hardened surface. The helix pattern is the laser-beam overlapped zone produced by the laser process.

Recent advances involving the use of ceramics, metal-matrix composite materials, laser-surface alloying, and laser-surface modifications to a substrate that normally could not survive in an abrasive service are examples of ongoing material developments.

Erosion Most fluids handled by pumps are considered clear liquids, meaning they do not have significant amounts of solid particulates present. The corrosive nature of these fluids dictates the required pump materials. Guidelines for many of these services are embodied in the "Corrosion Data Survey."[3]

However, many fluid-handling applications requiring pumps are far from clear liquids. Solid particulates can be removed with costly filtration systems that must work flawlessly at all times. Fabricated piping systems may introduce suspended solids from weld slag and pipe burn. Naturally occurring suspended solids are those found in water sources such as river water or seawater, as mentioned previously in the abrasive wear section.

The following factors should be considered during the material and pump selection phase of the procurement process:

- The hardness of the particles
- The quantity of particles
- Size distribution
- Nature (geometry)
- The velocity of the pumpage
- The angle of fluid impingement

The first four items listed deal with the suspended solids. These variables can vary from application to application. The hardness of the particles is important to understand

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in determining the materials necessary to yield an acceptable pump life span. Hardness can range from relatively soft substances, such as cellulose fiber in pulp and paper applications, to very hard abrasive particles such as silicon or rock in mining pumps. The Miller number index, as described in ASTM G75,[4] is used to characterize the abrasivity of hard particles.

The Miller number was developed to determine the relative abrasivity and attrition of solid particles making up a slurry. In a closed loop test, the abrasivity of the particles becomes less damaging with time due to the fracturing and rounding (or friability) of the particles as they strike each other and/or impinge on a pump or casing wall.

The Miller number is therefore reported with two numbers. The first number characterizes the abrasivity of the particles and the second is the loss of abrasivity (attrition) of the particles during the slurry test. The abrasivity portion of the Miller number is useful in practical applications because this more closely characterizes a slurry's damaging potential. The attrition number has found little use other than characterizing a test loop's influence on a slurry. A slurry with a Miller number less than 50 is not considered abrasive in a reciprocating pump. Examples of slurries with a Miller number below 50 are limestone, sulfur, and detergent. It has been determined that a slurry consisting of finer particles is less abrasive than one containing larger particles. Test data show that Corundum at 220 mesh is about four times as abrasive as the same material at 400 mesh.[4]

Particle velocity plays a major role in the degree of damage that occurs in a pump handling slurries. In this case, the potential energy is converted into kinetic energy, producing a material loss by the transfer of energy from the particle to the component. The amount of material damage on an individual particle scale depends specifically upon particle velocity, v, and mass, m (kinetic energy, defined as mv^2). This is demonstrated by Finnie's equation[5] for hard materials:

$$\text{Wear rate} = (\#\text{ of impinging particles}) \times (\text{average particle mass}) \times (\text{impingement velocity})^2 \times (\text{angle of impingement})$$

Of course, the pump part that absorbs the kinetic energy resulting from the particle impact has a role to play also. The material hardness and/or resilience of the pump component in absorbing the particle's impact energy will also determine the amount of material loss.

Chen and Hu[6] have performed laboratory tests on materials while changing the particle variables previously described. Their test results show the following:

- An increased particle hardness increases the material loss up to 1700-kg/mm^2 microhardness (greater than 75 R_c). Beyond this hardness, a decrease in wear occurs. This is most likely the result of the hard, brittle particle fracturing, which absorbs some of the kinetic energy.
- Sharp, angular particles increase the erosion rate over round particles.
- Erosion increases with increasing concentrations of abrasive particles.
- An increased fluid (and particle) velocity increases the erosion rate.
- Minimal erosion occurs at an impingement angle of 0° (tangent to the target surface) and increases to a maximum amount of wear at a 65° angle.

A review of the literature shows that several authors have plotted the solid particle impingement angle versus the amount of erosion.[7,8] These plots show that for ductile materials, erosion increases with the increasing impingement angle to a maximum material loss at an angle of 25°. Then the erosion damage decreases to the 65° impingement angle previously mentioned. Brittle materials, such as glass, are quite different. As the impingement angle increases from 0° to 90°, the volume of material loss continuously increases.

The characteristic features of erosion damage due to solid particle impingement are usually recognizable. However, when an aggressive fluid is present, the effects of solid particle impingement may not be easily identified. These effects can appear very much like corrosion-erosion, which is a fluid velocity-controlled damage mechanism where entrained solids are not present. If this damage is misdiagnosed, an improper material substitution can be made that may not solve the real problem. Conversely, a more likely situation is

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FIGURE 12 The severe erosion of a carbon steel casing in a 17% bauxite and sand service. Note the gouging due to the local turbulence of the slurry.

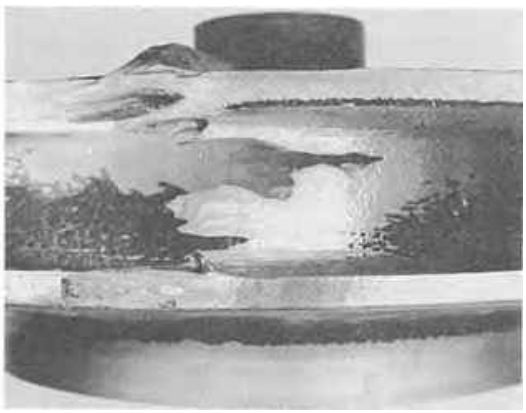


FIGURE 13 Erosion at the exit vane tips of a duplex, stainless steel CD4MCu impeller in a bauxite service

that the observed damage resulting from the erosion-corrosion is misinterpreted as solid particle erosion. A full understanding of the pumpage including the fluid velocity, fluid corrosiveness, content, and nature of the solid particles present is necessary for the appropriate action to be taken in improving the life of a damaged pump.

An example of solid particle erosion in a pump is shown in Figures 12 and 13. The severe erosion damage of a casing is illustrated by the gouging of surfaces that were directly impinged or scoured by glancing blows of the solid particles in the fluid stream. This pump handles a bauxite slurry where the percentage and velocity of alumina (Al_2O_3) and sand are too high for the carbon steel casing and CD4MCu impeller.

Figure 14 shows a CF3M impeller in a fly ash service. This shows that the greatest damage to the impeller is at the outer periphery, which corresponds to the highest velocity of the slurry. The least amount of damage is near the impeller inlet eye. Figure 14 also shows that the lower velocity region of the impeller inlet eye has the least damage. This confirms the laboratory data that show an increased erosion with an increased slurry velocity. Note that the damage increases near the outside of the inlet eye and is almost nonexistent at the impeller hub where the fluid velocities are lower.

Erosion damage can also be encountered in reciprocating pumps. Figure 15 shows extensive erosion of an AISI-type, 440C stainless steel ball from a ball valve after it became stuck and unable to rotate in a coal slurry application. This caused a slurry impingement on a concentrated region of the ball.



FIGURE 14 An austenitic stainless steel impeller in an abrasive fly ash service that shows severe erosion. Increased erosion occurs with an increasing fluid velocity near the periphery of the impeller.

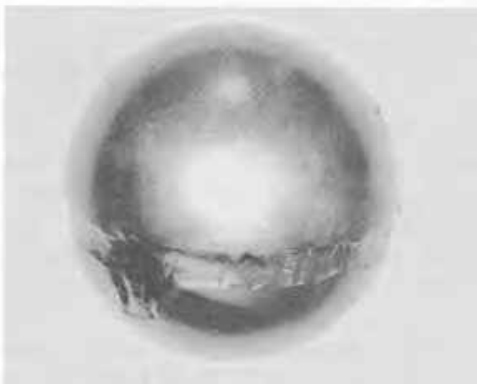


FIGURE 15 Erosion damage of an AISI-type, 440C stainless steel ball valve in a coal slurry service

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Erosion damage, once identified, has a limited number of solutions to prolong the longevity of pump materials. This can be accomplished by the selection of hard, wear-resistant replaceable liners, elastomeric liners, or, in cases where liners cannot be utilized, hard materials. Such metallic materials include white cast iron (such as Ni-Hard), high chromium (13 to 28%) alloy steels, cobalt-based super alloys (such as stellite), and nickel-based alloys. Slurry pumps tend to be large single stage pumps operated at low speed to minimize erosion damage. These pumps may use rubber liners to absorb the impingement energy of the abrasive particles, or be made of abrasion-resistant alloys.

FATIGUE

Centrifugal and reciprocating pumps are subjected to cyclical loading, which, if not considered during design, will result in a limited life due to material fatigue. In combination with a corrosive environment, material fatigue can be accelerated due to what is commonly referred to as *environmentally assisted fatigue.*

The one essential parameter in component fatigue is the presence of an alternating or cyclic load. In general, pumps are machines that have either fluid or mechanically induced cyclic loading on their components. Although centrifugal pumps are steady-state rotating equipment, pulsations or fluctuating applied stresses are encountered. The source of these cyclic stresses can be from fluid interaction between impeller exit vanes and diffuser vanes or, in a volute pump, the impeller vanes and the casing cutwater. Mechanically induced forces are due to bending moments acting on the pump shaft or possibly a component imbalance in the rotor assembly. Reciprocating pumps experience a cyclic loading of the internal and external components from the action of the machinery. In fact, these pumps can be thought of as large fatigue-testing machines due to the pulsating action of the pumping process.

When cyclic forces are applied to materials in a pump over a period of time, a crack may initiate, usually at a geometric stress concentration. After initiation, the crack will grow with continued cyclic loading until the part finally fractures. Fractures can occur, even though the loading produces stresses that are far less than the tensile strength of the material. Engineers have been aware of this potential mode of component fracture for many years and have developed design criteria that take this anomaly into account. The study of cyclic loading and material behavior based upon cyclic stress history and flaw size is beyond the scope of this text. It should be noted, however, that the field of fracture mechanics offers an engineering design tool that can predict the life of an engineered component.

Fatigue is a three-stage process consisting of: (1) crack initiation, sometimes associated with preexisting defects, (2) crack propagation, and (3) the final fracture, associated with crack instability, as suggested by Wohler.[9] The applied stress level, sample geometry, flaw size, and mechanical properties determine the existence and extent of these stages.

Fatigue was first studied by August Wohler in 1852.[10] Wohler's work included the concept of alternating applied stress, S, and the number of cycles, N, applied to a sample until a fracture occurs. This work is the basis for today's S/N curves used by design engineers. A laboratory-generated S/N curve is shown in Figure 16. This curve was generated by

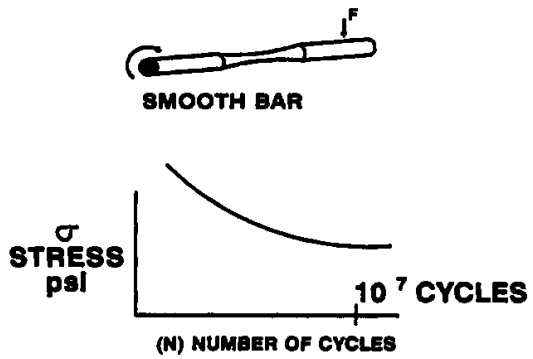


FIGURE 16 A laboratory-generated S/N curve for a smooth bar rotating beam test specimen

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smooth, rotating-beam test specimens. These specimens are machined carefully to avoid metallurgical notches on their surfaces that would lower the applied stresses required to produce a failure during testing.

When the fluid handled is corrosive, pitting, or other types of corrosion develop and act as stress risers. Since corrosion over time can increasingly damage the material, the endurance limit will correspondingly decrease with the exposure time. No true fatigue limit exists for materials in a corrosive environment. For this reason, the corrosion-assisted fatigue life of a material is usually published with cautionary statements. Given enough time, corrosion can penetrate completely through a fatigue test specimen, resulting in a data point of zero load and zero cycles. For this reason, corrosion-influenced fatigue test results usually specify the corrosive media, the test temperature, the details of the sample preexposure to the corrosive media, and the test frequency with respect to the applied cyclic loading.

Published data vary because laboratories that use a low frequency of applied stresses increase the influence that corrosion has upon the test specimens. This is in comparison to laboratories that conduct these tests at a high frequency that minimizes the influence of the corrosive media. Published values for the ultimate tensile strength and corrosion fatigue strength of various alloys are shown in Table 2.[11]

Wohler's investigation shows that a mechanical notch can reduce a material's fatigue.[9] This is shown in Figure 17 as a shift in the *S/N* curve below the curve produced by a smooth bar specimen. The severity of the notch determines the amount of divergence from the smooth bar curve. As shown, the surface degradation mechanisms lower the stress needed to produce the specimen failure after a certain number of cycles. This in turn reflects a lowering of the material's endurance limit.

The three stages of fatigue cracking can be observed on the fracture face, rendering them easily identifiable. This is especially true if no other secondary damage masks the characteristic appearance. Especially in fractures that occur over long periods, lines are visible on the fracture surface. These bands are sometimes referred to as *clamshell markings, crack arrest lines,* or *beach marks* and reflect different periods of crack growth. Ratchet lines, which represent the joining of two different crack fronts on different planes into one, are observed in multiple origin fatigue cracks. Multiple origin fatigue fractures are often associated with rotating components.

TABLE 2 Corrosion fatigue strength of alloys in seawater*

Alloy	UTS	CFS
Ti-6Al-4V	154	88
Inconel 718	189	60
Inconel 625	149	50
Hastelloy C	108	32
Monel alloy K-500	176	26
Ni Al bronze (cast)	115	15
304 Stainless	79	15
316 Stainless	85	14
304L Stainless	75	14
316L Stainless	79	13
17-4PH - cast		10
70-30 Cu-Ni (cast)	83	9
Ni Mn Bronze	82	9
Mn Bronze	73	8
D-2 Ni-Resist		7.5
Mild steel		2

*Test parameters: ambient temperature, 1750 rpm, 2 to 3 ft/s (0.6 to 0.9 m/s). Corrosion fatigue strength (CFS) given at 100,000,000 cycles. All values are in ksi; 1 ksi = 6.894759 MPa. (UTS is ultimate tensile strength of material in air.)

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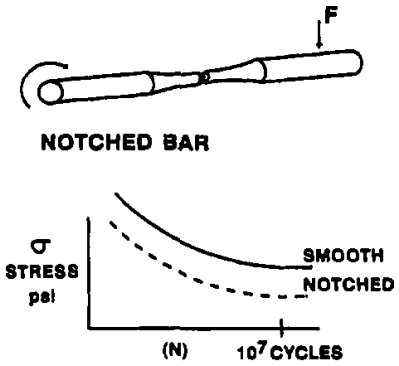


FIGURE 17 A shift in the *S/N* curve below the curve produced by a smooth bar specimen. The severity of the notch determines the amount of divergence from the smooth bar curve.



FIGURE 18 SEM photo of a fracture smooth fatigue test specimen with a single origin. The arrow indicates the fatigue crack origin.

Figure 18 shows a high magnified view of a smooth bar fatigue specimen after a fracture. The arrow shows a single origin of this specimen. This fatigue crack propagated across the entire specimen diameter until the final fracture occurred. The final fracture is sometimes referred to as the *ductile overload zone* or the *fast fracture zone.* An example of a fatigue fracture on a pump shaft is shown in Figure 19. The arrows in this figure indicate the location of many crack origins. The flat, smooth surface appearance of this fracture face is characteristic of most fatigue fractures. This flat fracture appearance is sometimes mistaken for a brittle fracture because no evidence of plastic deformation is observed on or near the break. Shown almost directly in the center of the fractured shaft section is a small area of ductile overload, marked A and B, which corresponds to the final fracture area. The relatively small size of this area indicates the crack propagated under low alternating loads. In other words, the only material holding the two halves of the shaft together was the last area to fracture. This is a mere fraction of the total cross-sectional area of this shaft. The multiple arrows at the OD of the shaft shows the many

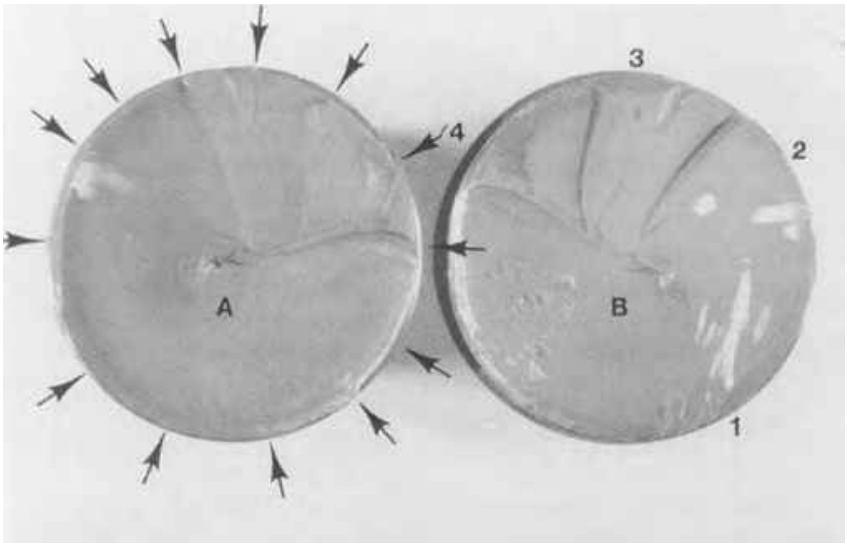


FIGURE 19 Multiple origin fatigue fracture of a pump shaft. The arrows show the locations of the many fatigue crack origins. A and B correspond to the final fracture zone of each fracture face.

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crack origins. Fatigue ratchet marks are at each of these locations. This type of fatigue fracture is referred to as *multiple origin, high cycle fatigue.*

As mentioned before, an investigator uses the relative size of each fatigue crack stage to determine the magnitude of the loads acting on the component. The identification of the crack origin is also of prime concern in conducting a failure analysis. The crack origin is important to determine if the fatigue crack initiated from a flaw in the material or a notch produced in service or during manufacturing.

Corrosion, often the primary cause of pump material damage, can increase the likelihood of fatigue cracking. *Corrosion-assisted fatigue* is the name given to this special type of cracking. Corrosion damage can change the surface texture and significantly increase the local stresses acting on the pump component. If the corrosion damage is severe enough to produce a sharp notch in a region of high cyclic loading, then fatigue cracking of the component is inevitable. In some cases, the propagation phase is also influenced by oxidation, which can mask the telltale features of the fatigue mechanism. Corrosion oxides, which form along the crack face, can produce a wedging effect, which mechanically increases the local tensile forces acting on the crack tip. This increases the crack propagation rate.

An example of corrosion-assisted fatigue at two locations in the front shroud wall of an impeller is shown in Figures 20 and 21. The evidence of corrosion pitting on the surface indicates a strong possibility that corrosion influenced the fracture mode. Further investigation shows that both shroud wall fatigue fractures were initiated at corrosion pits located in highly stressed areas of the impeller. The fluid pulsations acting on the exit vane tip result in alternating loading.

Corrosion is not the only mechanism of surface degradation that can promote this form of cracking. Surface disruptions through fretting or wear can also provide sites for fatigue crack initiation. Sharp radii and defects at the material surface such as porosity can act as stress risers. Abusive grinding can alter the microstructure at the surface and induce shallow microscopic cracks, which can propagate by a fatigue mechanism.

Once the mechanism of fatigue cracking has been identified, suitable corrective actions can be implemented. These include the following:

- **Higher strength materials** A good approximation for the endurance limit of a metal is 50% of the material's tensile strength. This is for high-cycle fatigue where no macroplastic loading is experienced. A graph published in *Deformation & Fracture of Engineering Materials*[9] shows this rule of thumb.
- **Design modification** The stress acting upon a component can be reduced with an increased section size. Reducing the stress on a component will increase its life. The design criteria for mean stress in an alternating loading environment can be determined



FIGURE 20 Overall view of a CF-3M impeller that has two corrosion-assisted fatigue fractures in the front shroud wall



FIGURE 21 A higher magnification of one of the fatigue fractures that originated at a corrosion pit at the exit vane tip and shroud intersection. Additional corrosion pitting can be seen on the impeller in this figure.

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using several analytical models. Since components are subjected to a range of loading (not a constant amplitude), a fluctuating mean stress is encountered. The anticipated load history can aid in the design process to avoid fatigue fractures. The prediction of potential component life can be based upon a fluctuating mean stress design criterion, referred to as the *Pamgren-Miner cumulative damage law*.[9]

- **Surface treatments** The introduction of compressive stresses to the surface of a part increases the fatigue life of a component. This is usually performed at crack-sensitive regions such as sharp corners or notches. If compressive stresses are introduced into the surface of a material, cyclic tensile stresses in excess of the compressive stress value are needed to cancel their effect before fatigue damage can occur. Therefore, any form of compressive stress will benefit a component with respect to fatigue cracking. Compressive stresses can be introduced by (1) cold working, (2) shot peening, or (3) a local heat treatment that introduces beneficial, compressive residual stresses (such as laser hardening or induction hardening).
- **Increased corrosion-resistant materials** The use of more highly corrosion-resistant materials is beneficial in cases where corrosion has decreased a component's life by degradation of its surface condition.

MATERIALS OF CONSTRUCTION

Impellers The pump designer needs to consider several criteria when selecting the material for the impeller:

- Corrosion resistance
- Abrasive wear resistance
- Cavitation resistance
- Casting and machining properties
- Weldability (for repair)
- Cost

For many water and other noncorrosive services, bronze satisfies these criteria and, as a result, is the most widely used impeller material for these services. Bronze impellers should not be used for pumping temperatures in excess of 250°F (120°C). This is a limitation imposed primarily because of the differential rate of expansion between the bronze impeller and the steel shaft. Above 250°F (120°C), the differential rate of expansion between bronze and steel will produce an unacceptable clearance between the impeller and the shaft. The result will be a loose impeller on the shaft.

Leaded bronzes have been used extensively in the past as impellers, especially in less demanding applications. The lead addition to bronze enhances its castability and machinability. In recent years, environmental concerns associated with lead have caused many nonferrous foundries to stop producing these alloys and pump manufacturers are increasing their use of nonleaded bronzes for impeller applications.

It should be noted that bronzes have velocity limitations above which they will suffer accelerated erosion corrosion. The maximum velocity, which will correspond with the periphery of the impeller, is higher in freshwater than in saltwater. The most resistant bronzes, able to tolerate the highest velocities, are the nickel aluminum bronzes. These alloys are often used as impellers in saltwater applications because they combine high mechanical properties, good corrosion resistance, and the capability to be weld-repaired. A nickel aluminum bronze impeller can be designed for a higher speed than any other bronze impeller alloy.

Cast-iron impellers are used to a limited extent in small, low-cost pumps. Cast iron is inferior to bronze in corrosion, erosion, and cavitation resistance. It also cannot be welded to repair damage due to wear or erosion. For these reasons, a low initial cost is usually the only justification for selecting a cast-iron impeller.

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Martensitic stainless steel impellers are widely used where bronze will not satisfy the requirements for corrosion, erosion, or cavitation resistance. The alloy most commonly used is CA-6NM. This alloy can be used for pumping temperatures above 250°F (120°C), as the thermal expansion is comparable to that of a steel shaft. Martensitic stainless steel impellers are used in a wide range of applications, including boiler feedwater, many cooling waters, and a variety of hydrocarbon applications. It does not have sufficient resistance to pitting corrosion for use in seawater.

Martensitic stainless steels are heat-treatable alloys. The specified mechanical properties are developed through a quench and temper heat treatment. Quenching can be in oil or, as is more common, in air. The cooling rate in air is sufficiently rapid that the high-temperature austenitic structure will transform to the metastable martensitic structure, which can subsequently be tempered to the desired hardness. The designer should specify that tempering be done at a minimum temperature of 1100°F (600°C) in order to assure that the casting has adequate toughness. It is also important that these alloys be heat-treated after weld repairs. This can present a problem in the case of a finish machined casting, which would suffer distortion if heat-treated. Welding techniques have been developed, however, that do not require a postweld heat treatment, but these are, in most cases, unsuitable for use on martensitic stainless impellers, particularly in higher stress areas near the periphery.

Oil and refining industry applications often involve exposure to hydrogen sulfide, which may be present as a trace contaminant in hydrocarbon fluids. Martensitic stainless steels are susceptible to a form of SCC in this environment and should be specified with a special double-temper heat treatment designed to limit hardness and thereby prevent cracking.

Austenitic stainless steels are used for impellers in applications requiring a higher level of corrosion resistance than can be obtained from the martensitic grades. A number of different alloys make up this group. The most widely used are CF-8M and CF-3M, which are the cast versions of the well-known 316 and 316L wrought materials. The cast alloys have a slightly different chemistry than the wrought grades. This difference accounts for the presence of 5 to 15% ferrite in the castings, which makes them slightly magnetic. The ferrite also enhances the resistance to SCC and hot shortness, a casting problem associated with fully austenitic cast grades. These alloys provide corrosion resistance over a wide range of pH and have reasonably good resistance to pitting and crevice corrosion in aqueous chlorides.

Higher alloyed austenitic cast grades are also available for applications requiring a greater degree of corrosion resistance. Alloy 20 contains about 30% nickel and was developed for sulfuric acid applications. The high nickel makes the alloy fully austenitic (without ferrite). Consequently, it is difficult to cast and suffers from hot shortness, which may manifest itself as fine cracking at the intersection between the vane and the shroud in an impeller. The high nickel content also makes Alloy 20 very resistant to SCC.

Austenitic grades containing 6% molybdenum have been developed for use in salt water and other high-chloride applications such as acidic brines used in oil field waterflood injection. The high level of molybdenum makes these alloys fully resistant to pitting in stagnant seawater, which will be present when a pump is not in operation. The 6% molybdenum grades are more expensive and therefore not frequently used for most applications. These alloys are usually considered only for critical, demanding applications where a high level of corrosion resistance is needed.

Austenitic stainless steels with unique properties have been developed for specific applications. A chrome-manganese alloy, discussed in the section on cavitation erosion, can be employed to mitigate or entirely eliminate cavitation damage in problem applications.

A high-strength austenitic stainless grade, CF10SMnN, can be used where the mechanical properties of CF-8M are inadequate. Some pump manufacturers also offer nitrogen-enriched austenitic grades that have corrosion resistance and mechanical properties better than CF-8M.

Duplex stainless steels offer a combination of higher mechanical properties and better corrosion resistance than the standard austenitic grades. The original duplex casting grade, CD4MCu, was developed in the 1950s. Use of this material was limited by problems with castability and weldability. Improved steelmaking technologies now enable the addition

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of precise amounts of nitrogen to duplex stainless steel. The nitrogen addition improves castability, weldability, and also corrosion resistance. Numerous duplex stainless grades have been developed in recent years, all having a specified nitrogen addition. These duplex grades all outperform the old CD4MCu grade, which did not have a nitrogen addition. Many foundries now make CD4MCu with nitrogen.

Duplex stainless impellers are extensively used in mining, flue gas desulfurization, and similar applications that require a combination of resistance to corrosion and abrasion. Duplex stainless steels also have better corrosion resistance than the standard austenitic grades and are used in a variety of applications in the chemical industry, the pulp and paper industry, and the marine industry. Duplex stainless pumps are standard for offshore high-pressure water injection pumps in the oil industry. Published corrosion data indicate that, for acceptable resistance to seawater, a duplex stainless should contain a minimum of 25% chrome, 3% molybdenum, and 0.15% nitrogen.

Casings The following criteria should be considered when selecting material for centrifugal pump casings:

- Strength
- Corrosion resistance
- Abrasive wear resistance
- Casting and machining properties
- Weldability (for repair)
- Cost

For many pumping applications, cast iron is the preferred material for pump casings when evaluated on the basis of cost. For single-stage pumps, cast iron usually has sufficient strength for the pressures developed. For corrosive or hazardous petroleum products, it may be necessary to specify cast steel or cast stainless steel. The concern with cast iron when handling hazardous fluids is that the material is inherently brittle and could fail suddenly in a catastrophic manner with no prior indication of distress.

Cast-iron casings for multistage pumps are limited to approximately 1000 lb/in^2 (6.9 MPa) discharge pressure and 350°F (177°C). For temperatures above 350°F (177°C) and pressures up to 2000 lb/in^2 (13.8 MPa) discharge pressure, a cast steel is usually specified for split-case, multistage pumps. For pressures higher than 2000 lb/in^2 (13.8 MPa), a cast or forged steel barrel-type casing is usually required.

In any evaluation of cast-iron versus steel casings, consideration should be given to the problem of casing erosion during operation. Erosion can occur either from abrasive particles in the fluid or from wire drawing across the flange of a split-case pump. Although the initial cost of a steel casing is higher than that of a cast-iron casing, a steel casing can often be salvaged by welding the eroded portions and remachining. Salvaging a cast-iron casing by welding is much more difficult, and the casting usually must be replaced.

The ductile irons are useful casing materials for pressure and temperature ratings between cast irons and steels. Although the modulus of elasticity for the ductile irons is essentially the same as that for cast iron, the tensile strength of the former is approximately double that of the latter. In any evaluation of the ductile irons as a substitute for the steels in the intermediate pressure and temperature range, it should be remembered that ductile iron casings cannot be effectively repair-welded.

Austenitic irons, commonly known by the tradename *Ni-Resist,* are used for pump casings in applications where gray and ductile irons have insufficient corrosion resistance. Austenitic irons typically contain 15 to 20% nickel. They are frequently used in brackish and saltwater applications where they are considerably more resistant to both corrosion and erosion than unalloyed gray iron. A preferred combination for this service is a Ni-Resist casing and stainless internals. The stainless steel is galvanically protected from pitting when the pump is made idle by the more anodic Ni-Resist casing.

The traditional Ni-Resist alloys have poor weldability in common with other types of cast iron. In recent years, a new and more readily weldable Ni-Resist grade has been

developed. This grade, designated *D2W,* contains a small columbium addition that enhances weldability. This new D2W grade is gaining popularity as the preferred grade for pump casings.

Bronzes are also used for pump casings in many water applications. Several bronzes are used, with the choice depending upon the specific application. Leaded bronzes, specifically leaded red brass, are used for small low-pressure pumps. This material is the least costly and easiest to cast of the bronzes. Tin bronzes, with or without lead, are used for larger centrifugal pump casings. The lead contributes to the pressure tightness of the casting. Unleaded bronzes often have to be impregnated in order to obtain adequate pressure tightness. Unleaded tin bronze can be weld-repaired, whereas the leaded version is not weldable. Nickel aluminum bronze has the highest mechanical properties and the best corrosion resistance of the bronze alloys normally considered for pump casings. It can also be repaired by welding. Nickel aluminum bronze casings are expensive and usually not competitive on a cost basis with Ni-Resist or other alternatives.

Stainless steels are selected for pump casings when required due to corrosion considerations. Martensitic stainless steels are commonly used to handle boiler feedwaters as well as many hydrocarbon applications. These materials have good mechanical properties and are suited for high-pressure applications. Their corrosion resistance is less than that of other categories of stainless steel and for this reason, they are unsuited for more aggressive waters or other fluids.

The austenitic stainless steels, particularly CF-8M and CF-3M, are frequently used for pump casings in chemical applications and other corrosive services. These materials can handle a wide range of pH. They are resistant to erosions by high velocity and can be field-weld-repaired with relative ease.

Duplex stainless steels (stainless steels having a metallurgical structure that is approximately 50% ferrite and 50% austenite) are used for pump casings in some applications that require a combination of corrosion resistance and mechanical properties superior to that of the standard austenitic grades. These materials have become the preferred choice for high-pressure, offshore injection pumps handling seawater. The higher mechanical properties permit the design of thinner wall, lighter pumps. The weight savings is an important factor in this application.

Shafts The following criteria should be considered in the selection of material for a centrifugal pump shaft:

- Endurance limit
- Corrosion resistance
- Notch sensitivity

The endurance limit is the stress below which the shaft will withstand an infinite number of stress reversals without failure. Since one stress reversal occurs for each revolution of the shaft, this means that ideally the shaft will never fail if the maximum bending stress in the shaft is less than the endurance limit of the shaft material.

In practice, however, the endurance limit is substantially reduced because of corrosion and stress raisers, such as threads, keyways, and shoulders on the shaft. In selecting the shaft material, consideration must be given to the notch sensitivity of the material, and to its resistance to corrosion in the pumped fluid. Corrosion will substantially lower the fatigue limit of the material. Fatigue cracks can initiate at corrosion pits or other surface discontinuities that act as stress risers.

In the absence of corrosion, an approximate relationship exists between fatigue endurance limits and mechanical properties. The endurance limit is equal to roughly half the tensile strength of the material. Depending upon the application, the pump designer will usually select the least expensive shaft material that will satisfy the three criteria noted previously. Carbon steel is used when corrosion resistance is not required, and relative low mechanical properties can be tolerated. A low alloy steel, often AISI 4140, is used when the mechanical properties of carbon steel are not adequate. Martensitic stainless steels, usually type 410, are a common choice when some measure of corrosion resistance,

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combined with reasonably good mechanical properties, is required. The resistance of type 410 stainless to fatigue crack initiation is related to the toughness of the material, which can vary over a wide range in commercial bar stock. The best measure of material toughness is the Charpy impact test, which is not a requirement of most relevant material specifications. In order to ensure the optimum toughness and resistance to fatigue cracking, type 410 stainless should be tempered at a temperature of 1100°F (593°C) minimum.

Stainless steels offering improved corrosion resistance and mechanical properties are also used in pump shafting. These include Nitronic 50, an austenitic grade, and 17-4PH, a precipitation hardening grade.

Several manufacturing issues involving pump shafts also need to be considered. Difficulty has frequently been experienced achieving the stringent tolerances for straightness required for long, thin shafts that are used in multistage pumps. This may necessitate intermediate stress relief during the machining process. The shaft may be stress relieved in the vertical position or, if horizontal, with supports every few feet. Type 410 pump shafting can be specially heat-treated to eliminate residual stress and maintain straightness tolerances during pump operation. Heat spot straightening of shafts is not recommended because it often adversely affects the microstructure and could lead to fatigue cracking.

Shafts can be plated or coated in specific areas for an improved resistance to wear or corrosion. Chrome plating is commonly used in this manner. The designer needs to be aware that plating reduces the fatigue endurance limit because of the fine microcracking associated with this process. An improvement in fatigue life can be achieved on plated surfaces if the substrate is shot peened. Normally, this reduction in endurance limits is not critical, because it is the flat surfaces, rather than the shoulders or keyways that are being plated. Consequently, the largest stress raisers are elsewhere in the shaft. There is also the potential for the plating process to cause hydrogen embrittlement cracks in high-strength steels. To avoid this problem, the steel should be baked at 300 to 400°F (150 to 200°C) after the plating process.

Finally, the manufacturing process needs to be carefully controlled to avoid the inadvertent introduction of stress raisers, which could shorten the life of the shaft. Abusive grinding has been identified as the root cause of some shaft fatigue failures. Heavy grinding will heat the surface and cause hardenable steels to form a thin layer of untempered martensite. This is a brittle structure and likely to develop fine cracks. These are stress raisers that can be propagated by a fatigue mechanism once the pump is placed in service. This type of problem is avoided by proper controls on the manufacturing process.

Wear Rings The following criteria should be considered in the selection of the material for the wear rings:

- Corrosion resistance
- Abrasive wear resistance
- Galling characteristics
- Casting and machining properties
- Suitability for coating

A centrifugal pump will often have both case and impeller wear rings. The impeller wear ring rotates within the bore of the stationary or case wear ring. These rings provide a close running clearance to minimize leakage from the discharge to the suction of the impeller. As the rings wear with use, leakage will gradually increase, affecting the head, capacity, and efficiency of the pump. In multistage (flexible rotor) pumps, increased wear ring clearance may also affect rotor stiffness.

To reduce the rate of wear of the wear rings, and thereby increase the life of the pump, special considerations must be given to the corrosion and abrasive wear characteristics of the ring material. Since the impeller and case rings may occasionally touch one another, the combination should also be selected to have antigalling characteristics.

Bronze is a widely used material for wear rings because it exhibits good corrosion resistance for a wide range of water services. In addition, bronze exhibits good wear characteristics in clear liquids but tends to wear rapidly when abrasive particles are present. The bronzes

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also have a relatively good resistance to galling. The leaded bronzes offer excellent galling resistance but use of these grades has been reduced due to environmental concerns associated with lead. The casting and machining properties of most grades of bronze are excellent.

In applications where bronze is not suitable because of either corrosion or abrasive wear limitations, or where pumping temperatures exceed 250°F (120°C), stainless steel rings are used. Unlike bronze, the stainless steels of the 300 and 400 series have poor galling resistance. Several options are available to minimize the possibility of galling between stainless steel rings. The clearance between the rings can be increased, serrations can be machined into one of the rings, or a minimum hardness differential of 50 to 100 Brinell points can be established between the rings, if made from a hardenable grade of stainless, such as the martensitic grades. If both rings are hardened to above 400 BHN, it is not necessary to maintain a hardness differential.

Martensitic stainless steel rings are usually hardened in a furnace. A laser hardening process is also available that involves heating only the surface of the material, which is rapidly quenched by the base metal, resulting in a precisely controlled surface hardness and a soft ductile core. Rings hardened in this manner are more resistant to cracking in some environments and provide increased wear resistance because of increased surface hardness (50 to 55 R_c). A summary of adhesive wear test results can be found in Table 3.

Increasing the clearance between rings is the least costly method for reducing the risk of galling or seizures. However, increasing the clearance will reduce the output and efficiency of the pump. In large, low-head pumps, the loss in efficiency is less than 1%, but in small, high-head pumps, the loss in efficiency can be significant. Serrated rings can be used on smaller pumps to help maintain the efficiency level, but only at an increase in manufacturing costs.

For cast-iron, bronze, hardened 11 to 13% chromium steels, and materials with similar low-galling tendencies, the recommended minimum running clearances between rings are given in API Standard 610, "Centrifugal Pumps for Petroleum, Petrochemical and Natural Gas Industries." For materials with higher galling tendencies, such as austenitic stainless steels and for all materials operating at temperatures above 500°F (260°C), it is suggested that 0.005 in (0.125 mm) be added to the recommended minimum diametrical clearances.

TABLE 3 Calculated wear factor

Material (Ring/Block)	Hardness DPH (Ring/Block)	Wear Factor K
In Distilled Water Specific Gravity 1.0		
Leaded Bronze/ASTM A48 Class 30 CI	80/205	0.17×10^{-4}
Ni-Resist/Ni-Resist	120/120	0.41×10^{-4}
Nitronic 50/Nitronic 60	195/190	0.76×10^{-4}
90-10 CuNi/ASTM A48 Class 30	87/210	1.14×10^{-4}
Stellite 12/Stellite 6	440/395	1.71×10^{-4}
Ampco 18/Ampco 18	155/155	2.40×10^{-4}
AISI 410/ASTM A743-CA6NM	300/270	2.45×10^{-4}
AISI 410/AISI 416	290/430	2.97×10^{-4}
AISI 416/AISI 416	430/360	3.57×10^{-4}
In Alcohol Specific Gravity 0.87		
Nitronic 50/Nitronic 60	195/190	0.62×10^{-4}
Leaded Bronze/ASTM A48 Class 30	80/205	1.54×10^{-4}
AISI 410/AISI 416	290/430	7.38×10^{-4}
In Iso-octane Specific Gravity 0.69		
Leaded Bronze/ASTM A48 Class 30 CI	80/205	0.41×10^{-4}
Nitronic 50/Nitronic 60	195/190	0.69×10^{-4}

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TABLE 4 Galling resistance of alloys

Metals in Contact			Threshold Galling Stress (ksi)	Threshold Galling Stress (mPa)
Silicon Bronze	BHN 200 vs. Silicon Bronze	BHN 200	4	27.6
Silicon Bronze	BHN 200 vs. Type 304	BHN 140	44	303
Waukesha 88	BHN 141 vs. Type 303	BHN 180	50+	345+
Waukesha 88	BHN 141 vs. Type 316	BHN 200	50+	345+
Waukesha 88	BHN 141 vs. S17400	BHN 405	50+	345+
Type 410	BHN 322 vs. Type 420	BHN 472	3	20.7
Type 416	BHN 342 vs. Type 316	BHN 372	13	89.6
Type 416	BHN 372 vs. Type 410	BHN 322	4	27.6
Type 440C	BHN 560 vs. Type 440C	BHN 604	11	75.8
S17400	BHN 311 vs. Type 304	BHN 140	2	13.8
S17400	BHN 435 vs. Type 304	BHN 140	2	13.8
Nitronic 50	BHN 205 vs. Nitronic 50	BHN 205	2	13.8
Nitronic 60	BHN 213 vs. S17400	BHN 313	50+	345+
Nitronic 60	BHN 205 vs. Nitronic 50	BHN 205	50+	345+
Nitronic 60	BHN 205 vs. Stellite 6B	BHN 415	50+	345+

Several antigalling material combinations have been used that do not compromise either wear or corrosion resistance. Nitronic 60 is a nonstandard austenitic grade that contains both manganese and nitrogen. This alloy has been developed for antigalling characteristics and will resist galling when mated with standard 300-series grades, Nitronic 50, and other alloys having poor galling resistance. Nitronic 60 is available in both wrought and cast forms. A cast nickel-based alloy, Waukesha 88, offers equally good antigalling characteristics and can be used in brine and other corrosive environments. A summary of galling thresholds for commonly used material combinations is shown in Tables 4 and 5.

Colmonoy- or stellite-coated rings are used in some critical applications and provide a high degree of resistance to abrasive wear, corrosion, and galling. These coatings are applied by welding, which can produce unacceptable distortion in rings, especially in larger diameters. The pump designer needs to exercise caution in specifying rings having a weld overlay because of potential manufacturing problems, which can involve cracking of the weld overlay or distortion of the ring. Some large rings for sewage pumps have performed well with tungsten carbide coatings applied using a high-velocity plasma spray process. The advantage of this process is that it does not heat the substrate, thereby avoiding distortion. Tungsten carbide is used for its high hardness and resistance to abrasion.

SELECTION OF MATERIALS OF CONSTRUCTION

The selection of materials for pumps is often not a simple and straightforward matter because of the weight assigned to various technical and economic factors by different users. A final selection may involve a compromise between the manufacturing cost and the anticipated maintenance costs. This is particularly true for fluids like seawater, for which a wide range of materials have been used, from cast-iron pumps with bronze internals to 6% molybdenum austenitic stainless steels. Despite the fact that economic factors unique to a particular application may influence the choice of materials for that application, some general guidelines can be used for some of the more common fluids.

Boiler Feedwater/Condensate High-purity water at a high temperature can be very corrosive to cast iron and carbon steel despite the fact that it usually has a low oxygen content. These materials will suffer erosion corrosion, sometimes within months, if incorrectly

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TABLE 5 Galling resistance of alloys

Metals in Contact				Threshold Galling Stress (ksi)
Class 30 Cast Iron	(BHN 159–172)	vs. UNS C93200—Leaded Tin Bronze	(RB 41–43)	14.3
ASTM A487 CA6NM	(RC 29–31)	vs. Type 416	(BHN 352–415)	5.9
ASTM A487 CA6NM	(RC 29–31)	vs. Type 410	(BHN 353–415)	2.1
ASTM A487 CA6NM	(RC 29–31)	vs. Type 420—Laser Hardened	(RC 53–55)	1.7
Laser hardened Type 420	(RC 50–51)	vs. Type 420—Laser Hardened	(RC 53–55)	15.9
Type 420	(RC 50–55)	vs. Type 420	(RC 50–55)	21.1
Type 420	(RC 48–49)	vs. As-received nodular iron	(RC 24–26)	2.5
Type 420	(RC 48–49)	vs. Nitrided nodular iron	(RC 45)	10.9
Type 420	(RC 48–49)	vs. Laser glazed nodular iron	(RC 57–60)	11.3
Type 420	(RC 48–49)	vs. Laser hardened nodular iron	(RC 57–60)	6.8
Type 420	(RC 48–49)	vs. CA15	(RC 44–45)	1.8
Type 420	(RC 48–49)	vs. Zirconia	(R45N 74–79)	25*
Type 420	(RC 48–49)	vs. TDC on AISI 4140	(RC 70–80)	5.1
Type 420	(RC 48–49)	vs. Electroless nickel on nodular iron	(RC 45–49)	5.1
Type 420	(RC 48–49)	vs. Graphalloy bronze	(45–50 Schleroscope)	4.3
Type 420F	(BHN 262–302)	vs. Type 420F	(RC 50–55)	3.8
Type 410	(BHN 262–302)	vs. Type 410	(BHN 353–415)	1.2
Type 416	(BHN 262–302)	vs. Type 416	(BHN 352–415)	2.7
Nodular iron	(RC 27)	vs. Graphalloy bronze	(45–50 Schleroscope)	3.6
Nodular iron	(RC 27)	vs. Graphalloy nickel	(45–50 Schleroscope)	
Nitrided nodular iron	(RC 45)	vs. Graphalloy bronze	(45–50 Schleroscope)	2.0
Nitrided nodular iron	(RC 45)	vs. Graphalloy nickel	(45–50 Schleroscope)	3.6
Laser glazed nodular iron	(RC 57–60)	vs. Graphalloy bronze	(45–50 Schleroscope)	3.6
Laser glazed nodular iron	(RC 57–60)	vs. Graphalloy nickel	(45–50 Schleroscope)	3.6
Laser hardened nodular iron	(RC 57–60)	vs. Graphalloy bronze	(45–50 Schleroscope)	5.0
Laser hardened nodular iron	(RC 57–60)	vs. Graphalloy nickel	(45–50 Schleroscope)	7.0
CA15	(RC 44–45)	vs. Graphalloy bronze	(45–50 Schleroscope)	2.0
CA15	(RC 44–45)	vs. Graphalloy nickel	(45–50 Schleroscope)	3.6
TDC on AISI 4140	(RC 70–80)	vs. Graphalloy bronze	(45–50 Schleroscope)	3.8

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Electroless nickel on nodular iron	(RC 45–49)	vs. Graphalloy bronze	(45–50 Schleroscope)	2.5
Graphalloy bronze	(45–50 Schleroscope)	vs. Graphalloy bronze	(45–50 Schleroscope)	3.6**
TDC on AISI 4140	(RC 70–80)	vs. Graphalloy nickel	(45–50 Schleroscope)	5.0
Electroless nickel on nodular iron	(RC 45–49)	vs. Graphalloy nickel	(45–50 Schleroscope)	3.8
Graphalloy bronze	(45–50 Schleroscope)	vs. Graphalloy nickel	(45–50 Schleroscope)	3.6**
Nodular iron	(RC 24–26)	vs. Zirconia	(R45N 74–79)	12.7*
Nitrided nodular iron	(RC 45)	vs. Zirconia	(R45N 74–79)	14.0
Laser glazed nodular iron	(RC 57–60)	vs. Zirconia	(R45N 74–79)	25.5*
Laser hardened nodular iron	(RC 57–60)	vs. Zirconia	(R45N 74–79)	25.5*
CA15	(RC 44–45)	vs. Zirconia	(R45N 74–79)	15.3*
Nodular iron	(RC 24–26)	vs. TDC on AISI 4140	(RC 70–80)	1.8
Nitrided nodular iron	(RC 45)	vs. TDC on AISI 4140	(RC 70–80)	11.5
Laser glazed nodular iron	(RC 57–60)	vs. TDC on AISI 4140	(RC 70–80)	15.3***
Laser hardened nodular iron	(RC 57–60)	vs. TDC on AISI 4140	(RC 70–80)	20*
CA15	(RC 44–45)	vs. TDC on AISI 4140	(RC 70–80)	5.1
Nodular iron	(RC 24–26)	vs. Electroless nickel on nodular iron	(RC 45–49)	3.8
Nitrided nodular iron	(RC 45)	vs. Electroless nickel on nodular iron	(RC 45–49)	2.5
Laser glazed nodular iron	(RC 57–60)	vs. Electroless nickel on nodular iron	(RC 45–49)	5.1
Laser hardened nodular iron	(RC 57–60)	vs. Electroless nickel on nodular iron	(RC 45–49)	5.1
CA15	(RC 44–45)	vs. Electroless nickel on nodular iron	(RC 45–49)	5.1

* No galling at this stress. Testing is limited by torque capabilities.

** No severe damage

*** Inconsistent results

Note: All stress values in ksi (1 ksi = 6.894759 mPa)

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applied. High-purity water is defined as that having a conductivity of 20 micromhos/cm or less, which is equal to a dissolved solids content of 10 ppm or less. At these low levels, carbon steel and cast iron are incapable of developing surface films such as magnetite that minimize erosion corrosion. The precise mechanism is not well defined, but the relationships between conductivity, oxygen content, pH, temperature, and the influence of these variables on erosion corrosion were established by pioneering work conducted at the Detroit Edison Power Plants in the 1950s.

The purity of feedwaters has increased over the years as water treatments have become more effective in removing dissolved solids to meet the demands of modern boilers. As a consequence, these waters have also become more corrosive. The early work at Detroit Edison showed that chromium additions to carbon steel dramatically enhance the resistance to erosion corrosion in high-purity water. Chrome additions of 1 to 1.25% are sufficient in most boiler feedwaters to impart acceptable corrosion resistance. Pump manufacturers initially used 5% chromium alloy steels for these applications. Difficulties in casting and welding 5% chromium steels led to the more recent use of 12 to 13% chrome steels, initially CA-15 and now CA-6NM, which is commonly used in boiler feed pumps, condensate pumps, and others that handle high-purity waters.

Water purity is directly related to conductivity, measured in units of micromhos, or microsiemens. It has been determined that, for high-temperature boiler feedwater services greater than 200°F (93°C), and with a conductivity of less than 20 micromhos, a chrome addition is usually required to avoid corrosion erosion at high velocity areas. Figure 22 shows a carbon steel shaft that was severely damaged in high-purity water.

Some applications still remain in which cast iron, bronze, and other less costly materials can be used. Materials are selected based on four critical water chemistry parameters: pH, temperature, conductivity, and dissolved oxygen. Table 6 represents an industry consensus on the recommended materials choices for the full range of high-purity waters.

Saline Water Saline waters have been defined as those that are sufficiently electrically conductive to enable an appropriate pump casing material to galvanically protect the pump internals when the pump is shut down. This corresponds to waters with more than about 1000 ppm chloride. Many pump applications involve saline waters. Among the more common saline water applications are the following:

- **Tidal river water** The chloride level here can fluctuate significantly with the season and the ingress of saltwater from a bay or estuary.
- **Groundwater** The chloride level and corrosivity can vary over a wide range. Some groundwaters, which are injected by high-pressure pumps into oil formations to enhance output, are very corrosive, due to low pH and very high chloride levels.



FIGURE 22 The corrosion erosion of a carbon steel shaft that was damaged in an ultra-pure water service after approximately 6 months

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TABLE 6 Materials for boiler feedwater services

	Feedwater Parameters				
pH range	0–14	4.5–14	6–9	6–14	9–14
Temperature ≤200°F (93°C)	S	G-S	B*-J* H-G-S	H-G-S	C*-L*-DR* H-G-S
Temperature >200°F (93°C) Conductivity <20 μmho/cm Dissolved oxygen >0.04 ppm	S	G-S			
Temperature >200°F (93°C) Conductivity <20 μmho/cm Dissolved oxygen <0.04 ppm	S	G-S		NM-G-S	
Temperature >200°F (93°C) Conductivity >20 μmho/cm Dissolved oxygen <0.04 ppm	S	G-S	B*-J* H-G-S	H-G-S	C*-L*-DR*

* Head must be less than 600 ft/stage (180 m/stage)
Materials:
B—Cast iron casing—bronze impeller—cast iron/bronze rings
C—All cast iron
DR—Carbon steel casing—cast iron impellers
G—All 12% chrome (CA-15 or CA-6NM)
H—Carbon steel case—12% chrome impellers (CA-15 or CA-6NM)
J—Cast iron case—bronze impellers—bronze rings
L—All cast iron with 12% chrome rings
NM—1- or 2-chrome barrels—12% chrome impeller (CA-15 or CA-6NM) S—All 316 stainless (CF-3M)

- **Geothermal water** This type may contain high levels of hydrogen sulfide, carbon dioxide, and other gases in addition to chlorides.
- **Oilfield brines** These are often deaerated, greatly reducing their corrosivity. Less corrosion-resistant pump materials may be used but are susceptible to corrosion during shutdown periods when oxygen cannot be effectively excluded from the water.
- **Seawater** The chemical composition of seawater is relatively uniform throughout the world. Other factors, including temperature, microbiological activity, and the presence of pollutants can alter the corrosivity of the seawater to pump materials of construction.

For each of the saline waters, a variety of materials has been used for pumps. The choice of materials for a particular application will depend on the water chemistry and other factors including the expected life of the pump, whether it will operate continuously or sit idle for long periods, and user preferences based on previous experiences. Some general considerations will influence material selections.

The materials for saline water pumps must resist erosion corrosion. Ni-Resist and copper-based alloys are frequently specified but have velocity limits, above which the protective oxide film is stripped off and accelerated corrosion occurs. Among copper-based alloys, nickel, aluminum, bronze can tolerate the highest velocity. The pump designer needs to be aware of these limitations and use bronze and Ni-Resist only for components when the velocity limits of the materials will not be exceeded.

Stainless steels develop a more tenacious oxide film than bronzes and can tolerate velocities much higher than those seen in pumps without suffering erosion corrosion. However, stainless steels are susceptible to pitting and crevice corrosion in stagnant seawater. These problems are exacerbated if marine biofouling occurs. Several methods exist for handling this problem. The stainless internals of a pump can be effectively protected by galvanic coupling with Ni-Resist. The combination of a Ni-Resist case and stainless steel internals is widely used because of this favorable galvanic relationship.

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To avoid localized corrosion during shutdown in an all-stainless pump, some form of cathodic protection is required. This can be either sacrificial anodes or an impressed current system. It is also possible to construct the pump of stainless grades that are highly alloyed and develop adequate corrosion resistance. This approach requires either 6% molybdenum austenitic grades or 25% chromium, 3% molybdenum duplex grades. These materials are considerably more expensive than standard 300-series austenitic stainlesses are frequently used in critical applications. Higher alloyed duplex grades have become the universal standard for high-pressure injection pumps, especially those used in offshore locations. The high mechanical properties enable the design of lighter, smaller pumps. The weight saving is an important factor in offshore applications.

Many large seawater pumps are constructed of cast iron with bronze internals. Provided the velocities are not too high, this combination of materials has been known to provide approximately 20 years of service in some large vertical pumps. Nickel aluminum bronze is preferred over tin bronze for the impeller because it is stronger, has better resistance to high velocity, and is more easily weld-repaired.

Monel shafting is no longer commonly specified for seawater pumping applications. This material is expensive and will develop pitting in stagnant water. Several grades of stainless steel will provide a combination of strength and corrosion resistance equivalent to Monel at a significantly lower cost. These include Nitronic 50 and several of the higher alloyed duplex grades, such as Ferralium.

When specifying materials for seawater pumps, the designer should also consider whether the water will be chlorinated, and, if so, where the chlorine is to be added. Chlorine is added to cooling waters to kill marine organisms that cause biofouling. The chlorine may be added continuously at low levels or as a shock treatment at periodic intervals. Chlorination at normal levels of up to 2 ppm does not appear to be detrimental to alloys commonly used in saline water pumps. However, an injection should be made far enough upstream of the pump intake so that the dilution occurs ahead of the pump. When an injection is made at or near the pump intake, copper alloys, stainless steels, and Ni-Resist may suffer accelerated corrosion. Recent work has shown that the corrosion rate of stainless steels will begin to increase at chlorine levels of about 5 ppm.

Galvanic considerations will also play a role in the material selection for saline water pumps. In general, the pump internals should be cathodic to the pump case. Coatings should be avoided, especially on the anodic component. Flaws or defects in a coating will expose a small area of base metal. Corrosion will then proceed at a high rate due to the extremely unfavorable area ratio. It is also inadvisable to use carbon or graphite bearings in seawater pumps. These are at the noble end of the galvanic series and are likely to cause a galvanic corrosion of stainless steels or other alloys with which they come in contact.

Table 7 indicates a number of material combinations commonly specified for seawater pumps.

Hydrocarbons Pure hydrocarbons are not corrosive, but they frequently contain small amounts of water or other substances that make them corrosive. Material selection guidelines for a variety of hydrocarbon services have been developed by the American Petroleum Institute and are reproduced in Tables 8 and 9. These tables provide general guidelines that do not necessarily apply to every situation.

Most corrosion problems in hydrocarbon production and refining applications are caused by contaminants in the fluid. These can include water, hydrogen sulfide, hydrochloric acid, hydrofluoric acid, sulfuric acid, and caustic. Some of these contaminants are present in the feedstock. Others may be process chemicals, including solvents, neutralizers, and catalysts. There are generally considered to be two types of corrosion in refineries, including low-temperature corrosion, below about 500°F, in the presence of water, and high-temperature corrosion, above 500°F, and not requiring the presence of water.

Contaminants present in crude oil are the primary source of low-temperature corrosion problems. Water is almost always present in crude oil, and is responsible for several types of corrosion. Chloride salts are often present as well. These salts may be dissolved in water that is emulsified in the crude oil, or as suspended solids. It is also possible that salts can originate from brines that are injected for secondary recovery. In either case, water hydrolyzes certain inorganic chlorides to hydrogen chloride, which is corrosive to carbon steel.

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TABLE 7 Materials for saline water pumps

	Component	Reference	Alternative	Nonferrous
Vertical pumps	Column/head	Ni-Resist[1]	316L	C614[4]
	Diffuser	Ni-Resist[1]	CF-8M/CF-3M[2]	C952[4]
	Bowl	CF-8M/CF-3M[2]	Ni-Resist[3]	C952 (4)
	Inlet ball	Ni-Resist	Ni-Resist	C952[4]
	Impeller	CF-8M/CF-3M[2]	CF-8M/CF-3M[2]	C958[4]
Centrifugal pumps	Case	Ni-Resist	CF-3M/CF-8M[5]	C952[4] G/M bronze
	Impeller	CF-3M/CF-8M[2] CD4MCu[4]	CF-3M/CF-8M[2] CD4MCu[4]	C958[4] Monel

	Component	Dearated Brine	Low pH Brine – H_2S
High pressure multistage—oilfield brines	Case	Duplex[6] C952[4]	5-6% Mo alloy[7] CF-8M
	Impeller	Duplex[6] CF-8M C952[4]	5-6% Mo alloy[7]

[1]Furnace stress relieved
[2]CF-8M should be postweld heat treated
[3]Erosion-corrosion may be high
[4]Postweld heat treat required
[5]Galvanic protection desirable to prevent crevice corrosion
[6]High-alloy grade with 25 Cr, 5-6 Ni, 3 Mo, and N
[7]20 Cr, 19-25 Ni, 5-6 Mo, and N

The water associated with crude oil is often sour, meaning that it contains hydrogen sulfide, and possibly other contaminants, including ammonia, hydrogen cyanide, phenols, and chlorides. While sour water tends not to cause general corrosion of carbon steel or cast iron (because it contains no dissolved oxygen), hydrogen sulfide is capable of inducing a form of stress corrosion cracking in hardenable steels and other alloys. Additional precautions must be taken when hydrogen sulfide is present, consisting of special heat treatments to limit hardness. Details concerning the requirements for specific materials can be found in NACE standards MR0175 (applicable to oil and gas production environments) and MR0103 (petroleum refining environments). Pumps furnished to API Standard 610 (ISO 13709) must comply with these requirements whenever the fluid contains hydrogen sulfide.

A variety of process chemicals can occasionally cause low-temperature corrosion problems in refinery pumps. These include sulfuric acid and hydrofluoric acid, both associated with different alkylation processes. Carbon steel pumps handling hydrofluoric acid have experienced leakage on occasion. Hydrofluoric acid will slowly attack nonmetallic inclusions in a carbon steel casting, eventually producing a leakage path through the casing. Enhanced nondestructive examination, usually involving a helium leak test, is usually required to assure the pressure-tight integrity of these pump casings. Monel inlays are used in high velocity areas.

Pump internals in concentrated or spent sulfuric acid service are often made of alloy 20, or similar high alloys. Hydrocarbons containing only traces of concentrated or dilute sulfuric acid have been successfully handled by carbon steel pump casings, with an aluminum bronze weld overlay.

High-temperature corrosion in refinery pumps is usually associated with sulfur compounds found in crude oil. These sulfur compounds can include polysulfides, hydrogen sulfide, mercaptans, aliphatic sulfides, and disulfides. All of these react with metal surfaces at elevated temperatures, causing uniform thinning, localized attack, or erosion-corrosion.

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TABLE 8 Material classes for centrifugal pump services (Courtesy of the American Petroleum Institute, reference 14)

CAUTION: This table is intended as a general guide for on-plot process plants and off-plot transfer and loading services. It should not be used without a knowledgeable review of the specific services involved.

Service	Temperature Range		Pressure Range	Materials Class	Ref. Note
	°C	°(F)			
Fresh water, condensate, cooling tower water	<100	<212	All	I-1 or I-2	
Boiling water and process water	<120	<250	All	I-1 or I-2	a
	120 to 175	250 to 350	All	S-5	a
	>175	>350	All	S-6, C-6	a
Boiler feedwater					
Axially split	>95	>200	All	C-6	
Double-casing (barrel)	>95	>200	All	S-6	
Boiler circulator	>95	>200	All	C-6	
Foul water, reflux drum water, water draw,	<175	<350	All	S-3 or S-6	b
and hydrocarbons containing these waters, including reflux streams	>175	>350	All	C-6	
Propane, butane, liquefied petroleum gas,	230	<450	All	S-1	
ammonia, ethylene, low-temperature services	> – 46	> – 50	All	S-1(LCB)	h
(minimum metal temperature)	> – 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;	<230	<450	All	S-1	
light, medium and heavy lubricating oils;	230 to 370	450 to 700	All	S-6	b, c
fuel oil; residuum; crude oil; asphalt; synthetic crude bottoms	>370	>700	All	C-6	b

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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
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

(*Continued*)

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TABLE 8 Material classes for centrifugal pump services (Courtesy of the American Petroleum Institute, reference 14) (*Continued*)

CAUTION: This table is intended as a general guide for on-plot process plants and off-plot transfer and loading services. It should not be used without a knowledgeable review of the specific services involved.

	Service	Temperature Range °C	Pressure Range °(F)	Materials Class	Ref. Note
Sulfuric acid concentration >85%	<38	<100	All	S-1	b
85% to <1%	<230	<450	All	A-8	b
Hydrofluoric acid concentration >96%	<38	<100	All	S-9	b

The materials for pump parts for each material class are given in Table 9.
Specific materials recommendations should be obtained for services not clearly identified by the service descriptions listed in this table.
Cast-iron casings, where recommended for chemical services, are for nonhazardous locations only. Steel casings (5.12.1.6) should be used for pumps in services located near process plants or in any location where released vapor from a failure could create a hazardous situation or where pumps could be subjected to hydraulic shock, for example, in loading services.

[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 or 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 for low-temperature services shall be selected with caution. Impact testing is recommended below –30°C (–20°F) Casting alloy ASTM A 352, Grades LCB, LC2 & LC3 is shown only for reference. Use equivalent materials for wrought alloys.

[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 9 Materials for pump parts (courtesy of the American Petroleum Institute, reference 14)

Part	Full compliance materials[b]	Material classes and abbreviations													
		I-1 CI[a] CI	I-2 CI BRZ	S-1 STL CI	S-3 STL NI-RESIST	S-4 STL STL	S-5 STL STL 12% CHR	S-6 STL 12% CHR	S-8[i] STL 316 AUS	S-9[i] STL Ni-Cu alloy	C-6 12% CHR 12% CHR	A-7 AUS AUS[c,d]	A-8 316 AUS 316 AUS[d]	D-1j Duplex Duplex	D-2[j] Super duplex Super duplex
Pressure casing	Yes	Cast iron	Cast iron	Carbon steel	Carbon steel	Carbon steel	Carbon steel	Carbon steel	Carbon steel	Carbon steel	12% CHR	AUS	316 AUS	Duplex	Super duplex
Inner case parts: (bowls, diffusers, diaphragms)	No	Cast iron	Bronze	Cast iron	Ni-Resist	Cast iron	Carbon steel	12% CHR	316 AUS	Ni-Cu alloy	12% CHR	AUS	316 AUS	Duplex	Super duplex
Impeller	Yes	Cast iron	Bronze	Cast iron	Ni-Resist	Carbon steel	Carbon steel	12% CHR	316 AUS	Ni-Cu alloy	12% CHR	AUS	316 AUS	Duplex	Super duplex
Case wear rings[k]	No	Cast iron	Bronze	Cast iron	Ni-Resist	Cast iron	12% CHR hardened	12% CHR hardened	Hard-faced 316AUS[e]	Ni-Cu alloy	12% CHR hardened	Hard-faced AUS[e]	Hard-faced 316 AUS[e]	Hard-faced duplex[e]	Hard-faced super duplex[e]
Impeller wear rings[k]	No	Cast iron	Bronze	Cast iron	Ni-Resist	Cast iron	12% CHR hardened	12% CHR hardened	Hard-faced 316AUS[e]	Ni-Cu alloy	12% CHR hardened	Hard-faced AUS[e]	Hard-faced 316 AUS[e]	Hard-faced duplex[e]	Hard-faced super duplex[e]
Shaft[d]	Yes	Carbon steel	Carbon steel	Carbon steel	Carbon steel	Carbon steel	AISI 4140	AISI 4140[f]	316 AUS	Ni-Cu alloy	12% CHR	AUS	316 AUS	Duplex	Super duplex
Throat bushings[k]	No	Cast iron	Bronze	Cast iron	Ni-Resist	Cast iron	12 % CHR hardened	12% CHR hardened	316 AUS	Ni-Cu alloy	12% CHR hardened	AUS	316 AUS	Duplex	Super duplex

(*Continued*)

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TABLE 9 Materials for pump parts (courtesy of the American Petroleum Institute, reference 14) (*Continued*)

Part	Full compliance materials[b]	Material classes and abbreviations													
		I-1 CI[a] CI	I-2 CI BRZ	S-1 STL CI	S-3 STL NI-RESIST	S-4 STL STL	S-5 STL STL 12% CHR	S-6 STL 12% CHR	S-8[i] STL 316 AUS	S-9[j] STL Ni-Cu alloy	C-6 12% CHR 12% CHR	A-7 AUS AUS[c,d]	A-8 316 AUS 316 AUS[d]	D-1j Duplex Duplex	D-2[j] Super 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]
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

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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	Filled carbon
Wetted fasteners (bolts)	Yes	Carbon steel	Carbon steel	Carbon steel	Carbon steel	Carbon steel	316 AUS[m]	316 AUS[m]	316 AUS	Ni-Cu alloy	316 AUS[m]	316 AUS	316 AUS	Duplex	Super duplex

[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 nongalling materials or nonmetallic 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 hy 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 nonwetted 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 W 40, where PRE is based on actual chemical analysis.
PRE $= \%Cr_{free} + (3{,}3 \times \%\ \text{molybdenum}) + (2 \times \%\ \text{copper}) + (2 \times \%\ \text{tungsten}) + (16 \times \%\ \text{nitrogen})$
$= \{(\%\ \text{chromium} - (14{,}5 \times \%\ \text{carbon})\} + (3{,}3 \times \%\ \text{molybdenum}) + (2 \times \%\ \text{copper}) + (2 \times \%\ \text{tungsten}) + (16 \times \%\ \text{nitrogen})$
Note that alternative materials such as "super austenitic" may also be considered.
[k]Nonmetallic wear part materials, proven to be compatible with the specified process fluid, may be proposed within the applicable limits shown in Tahle 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).
References in the notes above are to paragraphs within the API 610 standard.

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This is generally referred to as sulfidic corrosion. Chromium additions to steel reduce the corrosion rate of steel at high temperature in crude oils containing significant amounts of sulfur. The relationship between corrosion rate, temperature, and chromium content is shown in the McConomy curves. The corrosion rate is reduced to a negligible amount when the chromium content is increased to 12%. Consequently, the 12 to 13% chrome martensitic grades find considerable use in high-temperature refinery applications.

Naphthenic acids are organic acids that are found in many crude oils. They are corrosive at temperatures above about 450°F (232°C), with the corrosion rate tripling with each 100°F rise in temperature. Unlike sulfidic corrosion, no protective scale is formed, and low alloy and stainless steels containing up to 12% chromium provide no benefit over carbon steel. Damage is usually in the form of pitting and other localized attack. Austenitic stainless steels containing molybdenum are resistant to naphthenic acid corrosion. Type 316 stainless steel is adequate for applications involving naphthenic acid at lower temperature. Austenitic grades containing at least 3.5% molybdenum, such as 317, are required for more severe applications.

It should also be noted that there are several environments that are capable of inducing stress corrosion cracking in pump materials in refinery environments. In addition to aqueous chlorides, and hydrogen sulfide, which have been discussed previously, aqueous amines, and polythionic acid can cause cracking. Amines are used to remove hydrogen sulfide and carbon dioxide from refinery streams. Cracking has been found at welds in carbon steel components. It is thought to be similar to caustic stress corrosion cracking. Postweld heat treatment of carbon steel welds has been found to be an effective method of preventing cracking, and should be required for all fabrication welds, and repair welds, in carbon steel pumps that will handle amines.

Polythionic acids are formed by the reaction of oxygen and water with the iron/chromium scale that covers the surface of austenitic stainless steel components as a result of high-temperature sulfidic corrosion. Since oxygen and water are not present during normal operation in situations where austenitic stainless steels would be used, conditions suitable for polythionic acid to form occur only during shutdowns. Procedures to prevent cracking, including nitrogen purging, have been developed, and this form of cracking, which affects only sensitized materials, is relatively uncommon.

MATERIALS FOR SLURRY AND ABRASIVE SERVICES

The construction materials for pumps that handle high concentrations of suspended solids are often based upon high bulk hardness. In many applications, coatings, hard liners, and weld overlays are used to specifically increase the surface hardness of the internal wetted portions within the pump. However, many of the slurry applications use nonmetallics that do not have high bulk hardness because of their unique qualities.

Nonmetallics Contrary to "harder is better," a good number of slurry pumps use nonmetallic materials, such as rubber, that absorb the kinetic energy of the solid particles through a large elastic deformation of the surface. Natural rubber is the most commonly used material since it provides good wear resistance with abrasive particles less than approximately 0.25 to 0.38 in (6 to 9 mm). Rubber linings pose a problem in the bonding of this outer shell to a metallic substrate. This is particularly true for cutwater areas of a casing and, of course, attachment to metal skeletons of an impeller. Care must be taken also in considering the liquid phase of the slurry and the temperature of the application, both of which can degrade the rubber.

Other factors to consider when using elastomeric liners include pressure in the flow passage versus the pressure between the liner and the wall, and temperature, since rubber softens around 240°F (115°C). It is also important to consider the size of the particles greater than 0.2 in (5 mm) if they are dull or greater than 0.08 in (2 mm) if they are sharp, and the head per stage.

Metals In mildly abrasive services, carburized steels are sometimes used to increase the wear life of components. Carbon is diffused into the surface of carbon steel, which, after

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a hardening heat treatment, can achieve a surface hardness of 60 R_c. This gas diffusion heat treatment can produce high hardness layers that penetrate the outside surface of the pump component to a depth of approximately 0.080 to 0.090 in (2.0 to 2.3 mm). However, after carburization, the materials are impossible to weld-repair without cracking. Using a special process, usually a vacuum furnace, carburizing has been employed in the surface hardening of the 12% chromium stainless steels, such as CA15, for abrasive services where mild corrosion is expected.

The most commonly used materials for severe slurry services are the abrasion-resistant cast irons found in ASTM A532. Essentially, three main classes and several types of alloys are covered in this specification. The most widely employed material in slurry applications is the Class III hard irons. A brief description of this class of abrasion-resistant iron is as follows.

All three classes contain a martensitic matrix with secondary hard phases of chrome and iron carbides that increase the wear resistance. The molybdenum in Class II increases the material's hardenability for thicker cross sections.

It should be noted that machining and welding these three classes of material is impossible. Another important consideration is the role of carbon content on corrosion, erosion, and fracture resistance. High carbon contents reduce corrosion resistance because any chromium tied up as chrome carbide is no longer available to form a protective chrome oxide layer. Although beneficial with regard to erosion and abrasion resistance, high carbon content increases the susceptibility to breakage by thermal and mechanical shocks.

To counteract this problem, a number of precautionary measures must be adopted to enhance the serviceability of this class of material. First, slow warm-up cycles must be instituted, typically around 100 to 150°F per hour.[12] Another strategy is to lower the hardness from about 600 to 400 Brinell by a partial anneal. This measure reduces brittleness, but at the expense of erosion resistance.

Linings, Inserts, and Coatings The low ductility and toughness of A532 cast irons do not permit their usage with primary pressure boundaries according to ASME code and API regulations. This restricts the use of hard irons to internal wetted parts. Therefore, it is necessary to use steel pressure casings with hard materials as liners. A common slurry pump consists of ASTM A532, Class III, Type A (HC-250) impellers and replaceable HC-250 wear liners for the volute and for both the inlet and outlet ends of the pump casing.

Since pump erosion is often quite localized, in some instances it is more practical to install replaceable, mechanically attached inserts at high wear areas, such as the cutwater. These are typically made of sintered tungsten carbide or some other hard material. Newer materials, such as ceramic composites and toughened ceramics, should perform better than the "cermets" used in the past.

Several problems, however, can occur with mechanically attached inserts. One problem is protecting the fastening device against erosive wear. Another problem is the insert's tendency to act as a turbulence riser due to an imperfect fit or erosion-induced crevices and offsets. Limited success has been achieved with weld-applied overlays of stellite and other hard-facing materials. Weld overlays are extensively utilized, but several problems may occur. These include a propensity for cracking, debonding resulting from preferential corrosion of the bond line, dilution of the hard-facing material with the substrate, and potential uneven thickness after machining.

ASTM A532 Grade	Composition (Cr, Ni, and Mo)	Hardness (BHN)
Class I: Type A (Ni-Hard)	1 to 11% Cr, 3 to 7% Ni	500 to 600
Class II: Type A, B, C, D, E	11 to 23% Cr, 0.5–3.5% Mo	450 (annealed for machining) 600 (hardened)
Class III: Type A (26% chrome iron, original trade name of HC-250)	23 to 28% Cr	400 to 600 depending on desired properties

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Thermal spray coatings as well as diffusion surface treatments have been used in pump applications for fluids containing high concentrations of suspended solids. Spray coatings are restricted to areas within the pump, accessible by the line of sight. Diffusion-produced coatings are not limited by this constraint. A disadvantage of diffusion processes is that they are performed at high temperatures that can negatively influence the base material properties. Diffusion coatings can range from traditional gas carburizing to the diffusion of high chromium alloys. These coatings increase the surface hardness of the component and, depending upon the process, can increase the material's surface hardness to values in excess of 60 R_c. Diffusion layers can be produced to a depth of approximately 0.100 in (2.54 mm). One item of caution: these coatings usually render the material unweldable after application. For this reason, steps must be taken to protect areas of anticipated welding, such as attachment piping. Future weld repairs are not possible unless the coating is completely worn off or removed.

Through the years, developments in thermal spray equipment have enhanced the acceptability of this surface modification process. Thermal spray processes employ the transfer of a material onto another by raising the temperature of the hard-facing material, usually in powder form, and projecting it against the component that requires the additional erosion resistance. The bond strength between the hard-faced material and the substrate material is directly influenced by the maximum velocity that the particles of molten material achieve in a given thermal spray process. The greatest bond strength is achieved by the highest velocity process. The typical thermally sprayed materials used in pumps to resist solid particle erosion damage are as follows:

- Nickel chromium boride coatings
- Cobalt-based hard-facing coatings
- Tungsten carbide coatings
- Solid particle tungsten carbide loaded (1) or (2)

The processes used to apply the above hard-facing materials are as follows:[13]

Process	Typical Particle Velocity
Flame spray	100 ft/s (30.5 m/s)
Plasma spray	800 ft/s (244 m/s)
D-Gun (Union Carbide trade name for detonation gun process)	800 ft/s (244 m/s)
HVOF (high velocity oxy-fuel)	3000 ft/s (915 m/s)

The severity of the service usually dictates the process. In the past, thermal spray coatings, for the most part, were tungsten carbide and the diffusion coatings were high in borides. It was found that for spray coatings an increased performance could be achieved by applying them over erosion-resistant substrates. This is a challenge because the high-chromium, carbon, abrasion-resistant materials are thermal crack sensitive. Overlay coatings, if applied several times, are thicker but are more prone to cracking, chipping, and spalling. An alternative is to use carburized carbon steel or 12% chromium stainless steel centrifugal pump components for mildly corrosive environments. Coatings are frequently used to increase the life of plungers in reciprocating pumps for slurry services.

Another process for applying hard-faced materials is by laser consolidation. This process can be accomplished in two different ways. The first case is one in which a laser beam is used to melt a thermally sprayed coating applied upon a substrate. The other process is to simultaneously melt the substrate while applying a hard-faced material. In either case, the principle is to use the hard-faced material as a consumable in a laser-welding operation. Since laser welding is a rapid process, very little dilution of the hard-facing material is produced. This allows for much thinner coatings that are less prone to thermally induced cracking during an operation. In addition, since there is little dilution, the hardness and chemistry of the coating are very consistent. This provides for uniform erosion resistance throughout the entire coating thickness.

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SUMMARY

It should be clear from the discussion of materials for saline waters and hydrocarbon applications that the selection of materials for a pump is a complicated exercise, requiring knowledge of the engineering properties of the material, its fabrication characteristics, and corrosion and erosion resistances. An in-depth discussion of the considerations governing material selections for other pump applications, such as the chemical industry, mining industry, and others, is beyond the scope of this section, but a number of references are included to provide additional guidance. Past experience is often helpful, but it is recommended that the materials selections be reviewed by a metallurgist or corrosion engineer with experience in this area, especially for critical applications or those applications with which you are not familiar.

REFERENCES

1. Peterson, M. B. "Classification of Wear Processes." *Wear Control Handbook.* 1980, pp. 9–15.
2. Sparkar, A. D. "Fretting." *Wear of Metals.* 1976, pp. 116–121.
3. "Corrosion Data Survey," Metals Section, 5th ed., National Association of Corrosion Engineers, 1974.
4. Miller, J. E. "Miller Number." *Chemical Engineering.* July 1974, pp. 103–106.
5. "Wear of Pumps." *Metals Handbook,* Vol. 18. ASM International: Metals Park, OH, 1992.
6. Chen, J. H., and Hu, Z. W. "Main Causes of Slurry Wear of Various Materials Under Field and Laboratory Conditions." *Wear of Metals.* 1989, pp. 9–13.
7. "Coal Slurry Feedpump for Coal Liquefaction." Final Report EPRI AF-853. Study Conducted Under EPRI Project Manager H. Gilman by the Rocketdyne Division of Rockwell International. 1978, pp. 8–67.
8. Ives, L. K., and Ruff, A. W. "Electron Microscopy Study of Erosion Damage in Copper." *Erosion: Prevention and Useful Applications.* ASTM STP 664. 1977, pp. 5–32.
9. Hertzberg, R. W. "Cyclic Stress and Strain." *Deformation & Fracture Mechanics of Engineering Materials.* 1976, pp. 415–462.
10. Kesnil, M., and Lukas, P. "Fatigue of Metals." *Materials Science Monographs,* 7. 1980, pp. 9–16.
11. LaQue, F. L. "Marine Corrosion Causes and Prevention." 1975, pp. 84.
12. Wong, G. S., and Ackerman, R. E. "Coal Slurry Pump Development." Rockwell International Report RI/RD-38-217. 1984, pp. 83–217.
13. Bushan, B., and Gupta, B. K. "Coating Deposition by Hard Facing." *Handbook of Tribology.* 1991, pp. 8.1–8.26.
14. American Petroleum Institute Standard 610. "Centrifugal Pumps for Refinery, Petrochemical and Gas Industries," 10th ed., 2004.

FURTHER READING

Archard, J. F. "Contact and Rubbing of Flat Surfaces." *Journal of Applied Physics,* Vol. 24, 1953, pp. 981–988.

Bitter, J. G. A. *Wear,* Vol. 6, 1961, pp. 5–21, 169–190.

Butler, G., and Ison, H. *Corrosion and Its Prevention in Waters.* Robert E. Krieger Publishing: Huntington, NY, 1978.

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"Corrosion."*Metals Handbook.* 9th ed., Vol. 13. ASM International: Metals Park, OH.
Dillon, C. P. *Corrosion Control in the Chemical Process Industries.* McGraw-Hill: New York, 1986.

Hokkirigawa, K., and Kato, K. "Theoretical Estimation of Abrasive Wear Resistant Base on Microscopic Wear." *Wear of Metals.* 1989, pp. 1–8.

Holm, R. "Theory of Hardness & Measurements Applicable to Contact Problems." *Journal of Applied Physics,* Vol. 20, 1949, pp. 319–327.

Kotecki, D. J., and Ogborn, J. S. "Abrasion Resistance of Iron-Based Hardfacing Alloys." *Welding Research Supplement.* pp. 269-s–278-s.

LaQue, F. L. *Marine Corrosion—Causes and Prevention.* Electrochemical Society: Princeton, NJ, 1975.

LaQue, F. L., and Copson, H. R. *Corrosion Resistance of Metals and Alloys,* 2nd ed. Reinhold Publishing: NJ, 1963.

McCaul, C. "Evaluation of Corrosion Failures in Pumps." *ASTM STP 1000.* ASTM: Philadelphia, PA, 1990.

Rabonowicz, E. "Wear Coefficients—Metals." *Wear Control Handbook.* 1980, pp. 475–506.

Sparkar, A. D. "Abrasive Wear." *Wear of Metals.* 1976, pp. 69–73.

Speller, F. N. *Corrosion—Causes and Preventions,* 3rd ed. McGraw-Hill: New York, 1951.

Uhlig, H. H. *Corrosion and Corrosion Control.* John Wiley and Sons: New York, 1967.

SECTION 8.2
MATERIALS OF CONSTRUCTION FOR NONMETALLIC (COMPOSITE) PUMPS

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The use of composite materials is playing an increasing role in the manufacture of industrial pumps due to the cost savings they offer in manufacturing, installation, and operation. Composite parts can be molded to near "net shape." This eliminates the cost of secondary machining to attain the final part. Because one composite material can replace two or three different grades of metal part, the combining of several parts into one assembly is possible. This part's integration reduces assembly time, reduces inventory, and ultimately reduces manufacturing costs.

Because composite pumps weigh less than metal pumps, they are easier to handle during installation and maintenance. Further, because composites naturally dampen vibration, a composite pump operates more quietly than a comparable metallic pump.

The corrosion resistance of composites is superior to metals. For this reason, a composite pump's life can be significantly greater than a metallic pump. When the amortization costs of both composite and metallic pumps are calculated, the composite pump has the cost advantage.

FACTORS TO CONSIDER

Factors to consider in using composite materials for pump parts are mechanical properties, abrasion and corrosion resistance, temperature, costs, weight and insulating properties, exposure to sun, fire resistance, and chemical resistance.

Mechanical Properties The composite pump or parts must be designed to withstand the loads/stresses of the operating conditions. If the existing metal part design were

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simply replicated in a composite, the composite part could fail. Excessive stresses for a composite pump at the nozzle area or at the bearing housing can cause the pump housing to creep. The result is part distortion or bending and shaft misalignment.

A solution to this problem is to calculate the stress levels in the composite part and design within the composite's elastic limit. These complex stress calculations are now done on a routine basis with the aid of a computer and nonlinear FEA (finite element analysis) programs.

Abrasion and Corrosion Resistance The abrasive wear on 316 stainless steel is caused by the continuing removal of the metal's protective oxidation layer on the metal's surface. Composites do not need a protective layer and therefore have better resistance to mild abrasives. However, some composites are subject to corrosion by caustic liquids.

As background, metallic materials deteriorate from any one or combination of the following electrochemical reactions: galvanic action, pitting, corrosive attack, crevice corrosion, intergranular corrosion, and stress corrosion. The temperature, pH, formation, removal of an oxide corrosion barrier, or the velocity of the solution affects the rate of corrosion in a metal's surface.

Temperature Most composite pump materials are limited to operating temperatures below 300°F (150°C) for noncorrosive liquids and under 250°F (120°C) for corrosive liquids. Like metals, there is a reduction of mechanical properties for composites with increasing temperature.

Costs There are three areas of cost to consider when evaluating a composite part: development costs, initial capital costs, and cost savings.

Developing costs would include the following:

- Designing the part for composites
- Evaluating and testing alternative composite materials
- Prototype molding the parts

Initial capital costs are in the cost of the molds to produce the parts. Cost savings in manufacture and use include the following:

- Little or no corrosion increases the time between maintenance and extends the useful life of the pump.
- Reduced secondary machining cost at manufacture (the parts are near "net shape").
- Integrating several parts into one assembly results in a less expensive part, lower inventories, and less assembly time.
- Quality control cost is reduced as part tolerance is primarily determined by the mold.

The initial cost of a composite pump versus a metallic pump is typically as follows:

- Equal or slightly more expensive than an all 316 stainless steel pump
- 80% the cost of an Alloy 20 pump
- 50% the cost of a Hastelloy C pump

Weight and Insulation Properties With a significant reduction in weight compared to metal, the composite pump is easier to handle during installation and maintenance removal. As an example, an immersion sump pump that is usually 3 to 10 ft long weighs only 10 to 30% of a comparable metal pump.

Added insulation on the pump to conserve high temperature process heat is usually not needed on a composite pump, as the composite is a natural insulator when compared to metal. This saves the cost to fabricate an insulation cover over the pump and provides easier access for maintenance.

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Exposure to the Sun Some composites are susceptible to the UV (ultraviolet) rays of the sun. Two percent carbon black in the resin is an effective UV blocker. Hindered amine light stabilizers (HALS) can also be added to the composite to effectively prevent damage by the UV rays of the sun.

Fire Resistance In a severe fire, thermoplastic composites will melt and burn and thermoset composites will char and ash. If the composite reaches 400 to 500°F (204 to 260°C), there will be distortion of the pump and the pump will be lost. Because of the insulating properties of a composite pump housing, however, the liquids inside the pump will be cooler than if the pump were metal.

Fire retardant additives can be used with composite materials, but they may affect optimum mechanical properties. Most thermoset composites are self-extinguishing in a fire, but the pump manufacturer should be consulted to determine if there is a potential for toxic fumes if the composite is burned.

CHEMICAL RESISTANCE OF COMPOSITES

Composites do not corrode because they are nonconductive. Therefore, an electrochemical reaction does not take place. (An exception to this may be when electrically conductive long carbon fibers are used in the wrapping of a pressure vessel or a pump housing.) Their lack of resistance to different chemical fluids can degrade composite materials. This degrading can be in the resin, the reinforcing fiber, or in the coupling agent between the resin and fiber. If a composite material is going to be degraded, it will usually be immediate. Fluid temperature and concentrations are vital when selecting the appropriate composite material. With some composites, such as nylon, the pH can be a factor when selecting the best composite material for the application.

As an initial test, a sample coupon of the composite can be immersed in the process liquid and the results noted. A more definitive method is to review the composite manufacturer's informational database on chemical resistance. Many composite manufacturers have run extensive chemical resistance studies of their materials in a variety of liquids at set temperatures over time. The composite's physical properties are measured before and after the test period and the percentage deterioration versus time is documented in the manufacturer's literature.

The Hydraulic Institute, discussed later, has a polymer material selection guide that recommends specific composites for various liquids.

TYPES OF COMPOSITES

Composites can be divided into two general groups of organic compounds: thermoplastic and thermoset.

Thermoplastic Thermoplastic materials can be repeatedly melted and solidified. They do not go through an irreversible chemical process when heated. Amorphous thermoplastics resins have a broad softening range and are suitable for processes that require good melting strength like thermoforming. They can be transparent in color but do have limited chemical resistance.

Crystalline thermoplastic resins have a sharp melt point and are better suited to molding processes like injection molding where melt strength is not required. They have excellent chemical resistance. Their physical properties can be modified and improved with the addition of glass or carbon fiber, tougheners, or mica. Chart 1 shows the effect of various reinforcement materials on selected polymers.

Some of the thermoplastic processes that are used to form pump parts are injection molding, vacuum forming, extrusion, and blow molding. Thermoplastic resins suitable for pump applications include floropolymers, acrylics, polyethylene, polypropylene, polyvinyl chloride, and others. The allowable temperature range for these materials is from 130 to 300°F (54 to 150°C), depending on the material. Figure 1 shows the elastic modulus versus service temperature for various polymers.

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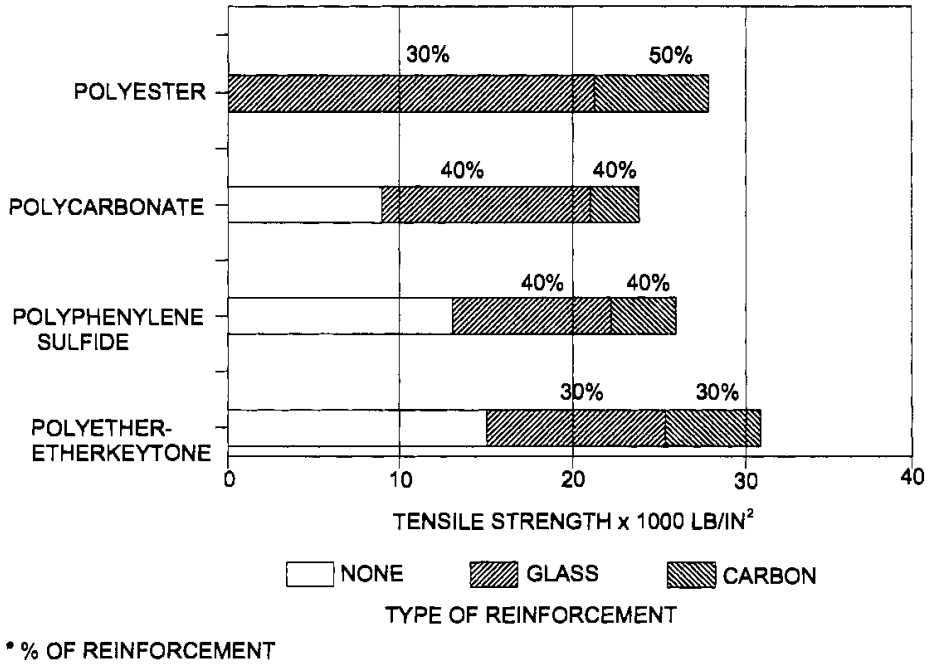


CHART 1 Thermoplastics strength versus reinforcement

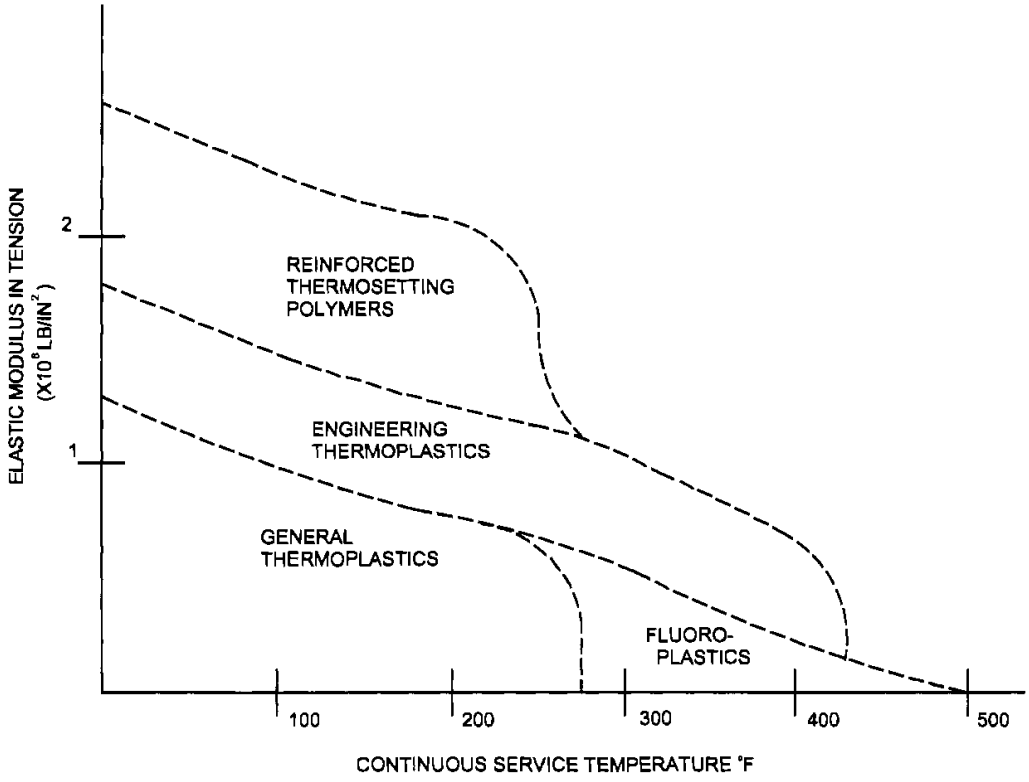


FIGURE 1 The effect of temperature on modules of elasticity

TABLE 1 Differences between thermoset and thermoplastic

Process	Thermoset	Thermoplastic
Average range of molded thickness	0.030 to 2.00 in (0.75 to 50 mm)	0.06 to 0.50 in (1.5 to 13 mm)
Weight range of piece	1 to 500 lb (0.5 to 225 kg)	0.5 to 5.0 lb (0.22 to 2.26 kg)
Glass content by volume	50 to 60%	30 to 40%
Length of glass fiber	0.25 in (6 mm) to several inches (mm)	0.06 in (1.5 mm)
Strength	Not uniform throughout	Uniform
Minimum annual quantities	Less than 1000	10,000
Obtain strength from ribs	Not necessarily	Yes
Tooling	Depends on complexity and size	Generally 20 to 30% higher than compression molding
Process	Compression or resin transfer	Injection, cannot use compression molding

Thermoset Thermoset materials, when heated, go through an irreversible chemical change with the result that the material will not soften when reheated. Thermoset resins are generally reinforced with glass fiber, linen, carbon fiber, graphite, minerals, and so on.

They have an allowable application temperature range of 180 to 250°F (82 to 120°C), depending on the material. Examples of thermosets are epoxies and vinyl esters. They can be stronger than thermoplastics but are usually more brittle. Thermosets are processed by compression molding or transfer molding. They have good chemical resistance to acids, bases, and oxidizing agents.

Table 1 outlines the differences between thermoset and thermoplastic polymers.

LINED PUMPS

To provide chemical resistance for a metallic pump, it is possible to line inside the pump housing. Some of the more popular liner approaches are flexible, rigid, sprayed, or molded. With a flexible-lined pump, a metal casing is made to the design shape and pressure. Then a loose liner is fitted inside the pump housing and bonded to the metal casing with an adhesive. This type of liner can come loose under vacuum conditions. There also can be problems at the termination of the liner to the mating pipe flanges. Depending on the liner material and design of the pump, liners can be molded into a pump housing. Two of the more popular methods are rotomolding, in which the liner material becomes bonded to the metallic armor, and injection molding, in which the liner is locked to the armor by strategically placed locking grooves.

A resin used in some lined pumps is PTFE (polytetrafluoroethylene). By itself, PTFE does not have the physical properties needed for a pump housing. When it is supported by a metal housing, however, it provides an excellent lining material due to its outstanding range of chemical resistance at high temperatures.

Rigid pump liners are machined out of plastic stock shapes or molded into a near net shape. The liner is then mechanically compressed between bolted steel plates to provide the support. Such pumps are usually small in size.

A sprayed-on composite lining is made by taking a metal casing of the required pressure rating, heating it, and then spraying the composite material inside the housing and flanges. At 0.030 in (0.76 mm) thickness, the lining will not have pinholes and is suitable for pump applications. This lining process is usually limited to pumps that require up to 100 horsepower (75 kW) drives.

Table 2 gives an overview of the comparative advantages and disadvantages of the different styles of pumps.

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TABLE 2 Comparative advantages and disadvantages of the different styles of pumps

Feature	Thermoset	Thermo-plastic	Flex-lined	Sprayed-lined	Rigid-lined
Cut impellers available	Yes	Yes	Yes & No	Yes	Yes
Potential for damage to liner in operation or repair	Low	Low	High	High	Low
Integral casing flanges	Yes	Yes	Yes	Yes	No
Efficiency	High	Med	Med	Yes	Med/Low
CPI hydraulic coverage	Yes	No	Yes	Yes	No
Conforms to ASME/ANSI B73.1M dimensions	Yes	No	Yes	Yes	No
Cost	Med	Low	Yes	Yes	Med/High
Pressure capability	Med	Low	High	High	High
Vacuum capability	Good	Fair	Low	Good	Good
Temperature capability	Med	Med/Low	High	High	High

TABLE 3 Mechanical properties of some unreinforced composites

Property/Material	Steel	Glass	Vinyl Ester	PVC	PVDF	PTFE	PPS	PP	PEEK	Epoxy
Modules of elasticity 1×10^6 lb/in^2	29	5	3.5	0.4	0.16	0.8	0.6	0.2	0.6	3.5
Tensile strength 1×10^3 lb/in^2	60	300	11	7	7	4	14	6	16	16
Specific gravity	7.8	2.6	1.2	1.3	1.8	2.2	1.3	0.9	1.31	2.2
Coefficient of thermal expansion 1×10^5 in/in/°F	0.6	0.3	2.0	2.8	8.5	10.0	2.8	8.0	4.0	2.2

Lb/in^2 × 6.894757 = kPa
In/in/°F × 1.8 = cm/cm/°C

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CERAMIC AND CARBON PUMPS

Solid ceramic or carbon pumps are used for very hostile liquids such as concentrated hydrofluoric acid. Ceramic and carbon pumps are much more expensive than nonmetallic pump materials, but do have the chemical resistance required for extremely aggressive chemicals.

TECHNIQUES FOR MOLDING PUMP COMPONENTS

Injection molding is a process for high volume forming of high quality thermoplastic parts. The part cost is low, but the initial mold costs can be high. Reinforcing materials can be added to an injection molded material. Typical additives are short-strand fibers (0.06 to 0.50 in or 1.5 to 13 mm long) and are glass, graphite, mica, and so on.

Compression or transfer molding uses thermoset resins and can make small quantities of larger parts. The mold costs are less, but the part price is usually more due to trimming and secondary machining steps. The reinforcing materials for these processes can be continuous or chopped mat strands, as well as woven mats and blankets. These long-strand fibers give exceptional physical properties to the final part.

MECHANICAL PROPERTIES OF COMPOSITES

Composites in general have far lower physical properties when compared to steel and lower physical properties than aluminum or brass. Composites can, however, be designed to withstand the operating pressures of a pump by increasing the section modulus of the part; that is, increasing the wall thickness. Table 3 shows some unreinforced composite properties versus steel. Figure 2 illustrates where the various composites and metallic are positioned and applied on a grid of developed pressure versus flow.

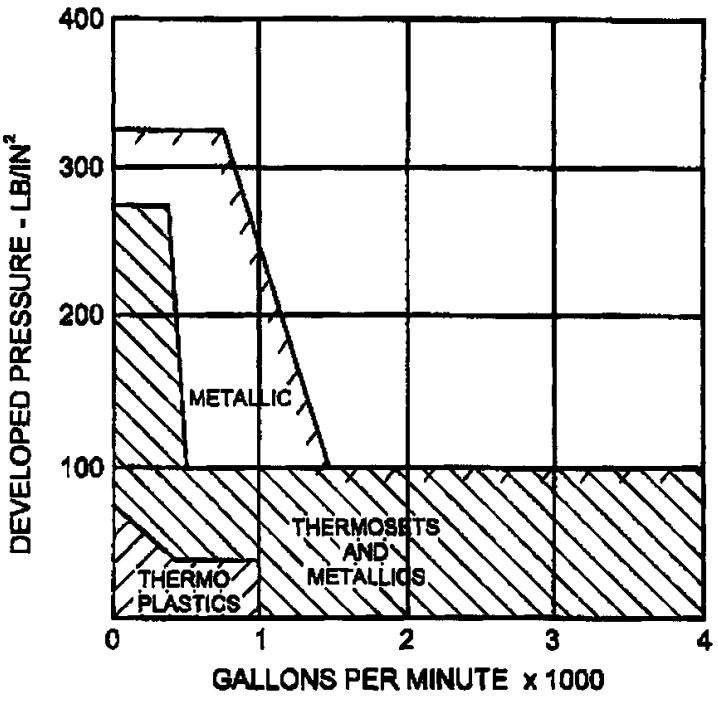


FIGURE 2 Developed pressure versus flow for composite pumps (lb/in² × 0.0689 = bar; gpm × 0.227 = m³/h)

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TABLE 4 Thermoplastics used for pump parts

Chemical Name	Common Reference	Temperature Limit °F	Temperature Limit °C
Polycarbonate	Lexon®	250	120
Phenylene oxide	Noryl®	194	90
Polyphenylene	Ryton®-PPS	250	120
Polyphenylene	PP	150–180	65–82
Chlorinated polyvinyl chloride	CPVC	230	110
Polyvinylidene chloride	PVDC	160	70
Polyvinyl chloride	PVC	140	60
Polyetherether keytone	PEEK	250	120
Polytetrafluoroethylene	Teflon®-PTFE	460	238
Chlorotrifluoroethylene	Teflon®	500	260
Polyvinylidenefluoride	Kynar®-PVDF	300	150

Thermoplastics PVC and CPVC pumps are low cost and suitable for the lower temperature ranges. PVC can be used up to 140°F (60°C) and CPVC, with greater abrasion resistance, is suitable for temperatures up to 210°F (100°C). PVC and CPVC can be solvent-bonded together to attain the best attributes of each resin in a single pump.

Polypropylene (PP) has excellent corrosion resistance and is used in hydrocarbon service up to 185°F (85°C). It is not suitable for strong acids or chlorinated hydrocarbons. The material can be ultrasonic, vibration, and spin welded in assembly.

PVDF (Kynar) is very corrosion and abrasive resistant and suitable for temperatures up to 300°F (150°C). It is also relatively expensive.

PTFE (Teflon) is extremely corrosion resistant, but its low physical properties require it to have a metal reinforcement backing to withstand high pump pressures. Table 4 is a partial list of thermoplastics used for pump parts.

Thermosets Vinyl esters and epoxies are the strongest of the thermoset composites and usually do not require external metal reinforcement. Vinyl esters are resistant to corrosion, whereas the epoxy composites are resistant to solvents. Table 5 is a list of thermosets used for pump parts.

CANDIDATE FLUIDS FOR COMPOSITE PUMPS

This section discusses some of the more general liquids that are suitable for use in composite pumps.

Chlorine and Caustics Chlorine is used in the production of organic chemicals such as vinyl chloride, chlorinated solvents, pesticides, and fluorocarbons. Chlorine is also used in

TABLE 5 Thermostats used for pump parts

Name	Common Reference	Temperature °F	Temperature °C
Vinyl ester GL	VE-Gl	250	120
Vinyl ester C	VE-C	250	120
Epoxy	Epoxy	250	120

GL—glass reinforced (should not be applied to hydrochloric acid or caustics)
C—carbon reinforced

the pulp and paper industry, municipal water purification, sewage treatment plants, and in the electrolysis of sodium chloride to produce sodium hydroxide. Additional fluids in this category are as follows:

- Potassium hydroxide
- Sodium chloride solutions
- Seawater, brine
- Sulfuric acid diluted up to 70%
- Hydrochloric acid
- Sodium hydrochloride
- Nitric acid up to 5%

Figure 3 shows that a composite pump can handle up to a 40% concentration of hydrochloric acid at 175°F (80°C).

Sulfuric Acid Figure 4 shows the operating range of a composite pump, pumping sulfuric acid.

Ferrous and Ferric Chloride Ferric chloride is used as an etching reagent in the production of printed circuit boards. It is also used as a coagulant in wastewater treatment. In metal pickling operations, ferric is produced when hydrochloric acid reacts with iron and steel. In all of these applications, composite pumps have replaced the more costly titanium pumps.

Ethylene and Propylene Candidate applications for composites of ethylene and propylene are:

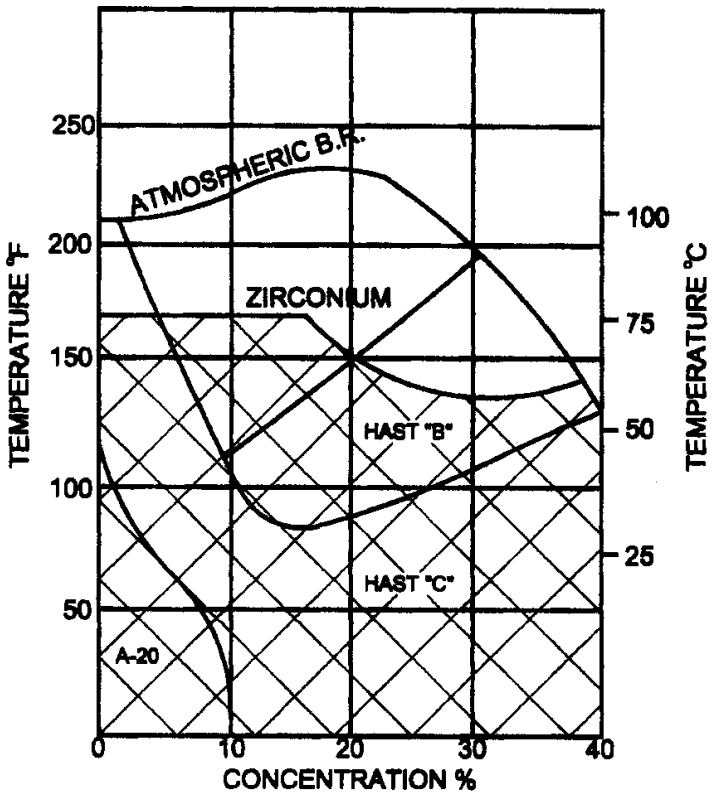


FIGURE 3 Material selection for hydrochloric acid

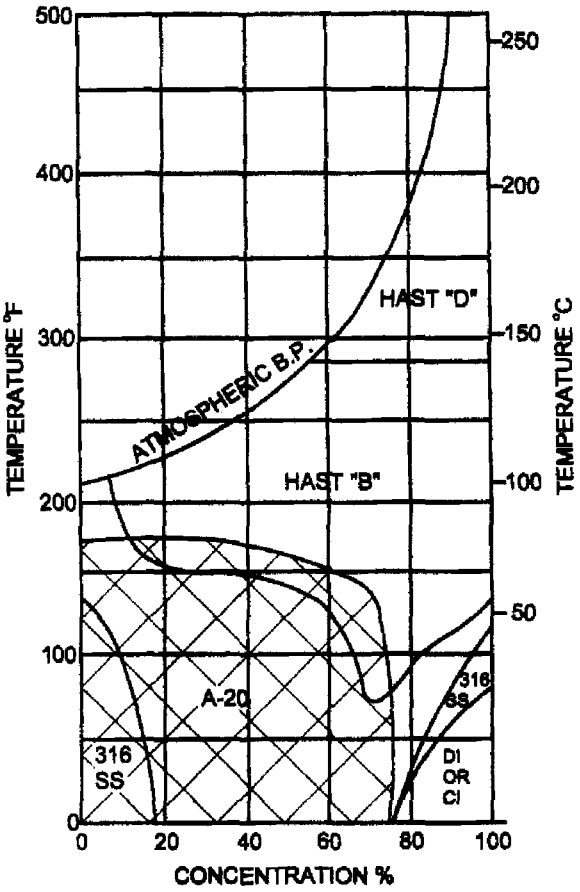


FIGURE 4 Material selection for sulfuric acid

- Propylene glycol
- Ethylene glycol
- Diethylene glycol
- Proplene chlorhydrin

Dyes Acid, sulfur, and Diazo dyes can be used in composite pumps.

Agriculture Chemicals Pesticides comprised of insecticides, herbicides, fungicides, and fertilizers can be severely corrosive to a composite pump. The chemical supplier should be contacted for their recommendations of materials suitable for use with their products.

Table 6 lists the maximum temperature and concentration of several liquids for various polymers.

INDUSTRIES USING COMPOSITE PUMPS

Pulp and Paper Composite pumps are used in bleaching and cooling of liquor preparation. The pumps can handle white, black, and green liquors. Other liquids suitable for composite pumps associated with this industry include:

- Sulfuric acid
- Sodium chlorate

- Sodium hypochlorite
- Sulfur dioxide
- Chlorine
- Methanol
- Peroxide bleach
- Sodium chloride
- Chlorine dioxide
- Sodium hydroxide
- Hydrochloric acid

Metal Finishing Composite pumps are an excellent choice for use in this industry for two reasons:

- The chemicals used in electroplating, steel pickling, etching, anodizing, galvanizing, and plating do not chemically attack composites.
- Metal pumps can generate stray currents and could effect the plating process.

Composite pumps are natural electrical insulators, which makes them well suited for this industry.

Desalinization and Water Purification Composites can replace 316 stainless steel, duplex stainless steel, and Alloy 20 on sea water distillation plants. On reverse osmosis equipment, composite pumps can be used for backwash, membrane blowdown, and intake screen wash.

Aqua Culture Composite pumps can be used for waste removal, seawater transfer, washdown, and filtering without adversely affecting sea life. This adverse reaction can occur with metal pumps. Metal pumps can have a chemical reaction with seawater and stray electrical currents generated by the pump can affect the pump's mechanical seal.

STANDARDS FOR COMPOSITE PUMPS

There are many composite pump designs for low-pressure and low-capacity use in the general-purpose market. For composite pumps in the "industrial chemical market," however, a separate standard ASME B 73.5M has been written. This standard is similar to B73.1M for metal pumps in that it requires the same dimensional interchangeability, shaft deflection, and seal chamber requirements. The composite pump standard differs from the metallic pump standard in these areas:

- Basic working pressure is from 100 to 275 lb/in^2 (6.9 to 19 bar) depending on pump size.
- Nozzle flanges are Class 150 dimensions but not Class 150 rated.
- Hydrostatic pressure factor above working pressure will depend on size, rpm, and manufacturing process.
- Pressure-temperature limit will be based on a manufacturer-user agreement for the liquid and its concentration.
- The standard applies to both thermoplastic and thermoset composites.
- Casing, casing cover, and gland have a minimum corrosion allowance of two years.

Hydraulic Institute Material Selection Guide A material selection guide has been written with recommendations of what composite should be used with what type of

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liquid. This guide lists 150 liquids at various concentrations and temperatures and recommends suitable compositites for applications. See References and Further Reading at the end of this section.

PUMP CONSTRUCTION

The pump section views in Figures 5 to 8 illustrate the construction of some composite pumps.

Components Figure 9 shows various pump components made from both thermoplastic and thermoset polymers. The enclosed impellers are made of thermoplastic resins because the front and back shroud can be ultrasonic- or vibration-welded together. Semiopen impellers can be made of either type of composite. On sizes greater than 6 in (152 mm) diameter, thermosets are usually used due to their superior strength over thermoplastics. When thermoplastics are used in larger impellers, a metal reinforcing skeleton is used for reinforcement. Casing and casing covers are made of either composite depending on the pump size and pressure rating.

The seal between the casing and cover is usually a shaped "O-ring" held by a groove running around the perimeter of the casing. The "O-ring" is compressed between the casing and cover with a ring of bolts around the casing's perimeter.

Shafts are made of AISI 4140 or 316 stainless steel with a thermoplastic shaft sleeve for corrosion protection. Mechanical seals are of "outside" construction to prevent liquid from contacting the metal portion (springs) of the seal.

Bedplates Composite materials have several advantages over metals and other materials when used for bedplates. Composites do not rust, rot, or deteriorate in adverse environments. The composite bedplate maintains its shape over time. This assures that the pump shafts will stay in alignment, and the pump's nozzle will not be stressed due to a sagging bedplate. The designed in features in a composite bedplate include an integral drip lip, slopping surface to collect drips, and ringed grout holes.

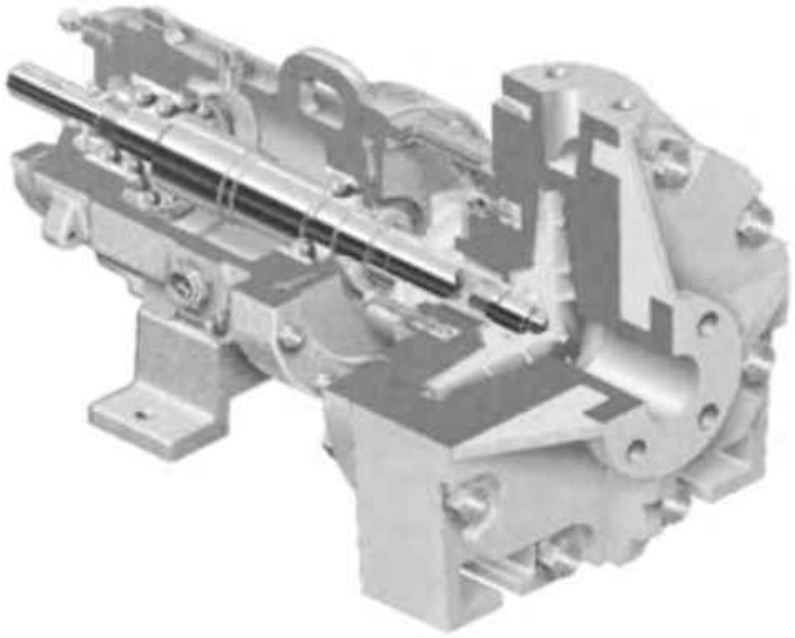


FIGURE 5 Thermoset ASME B73.5M pump (Flowserve Corporation)

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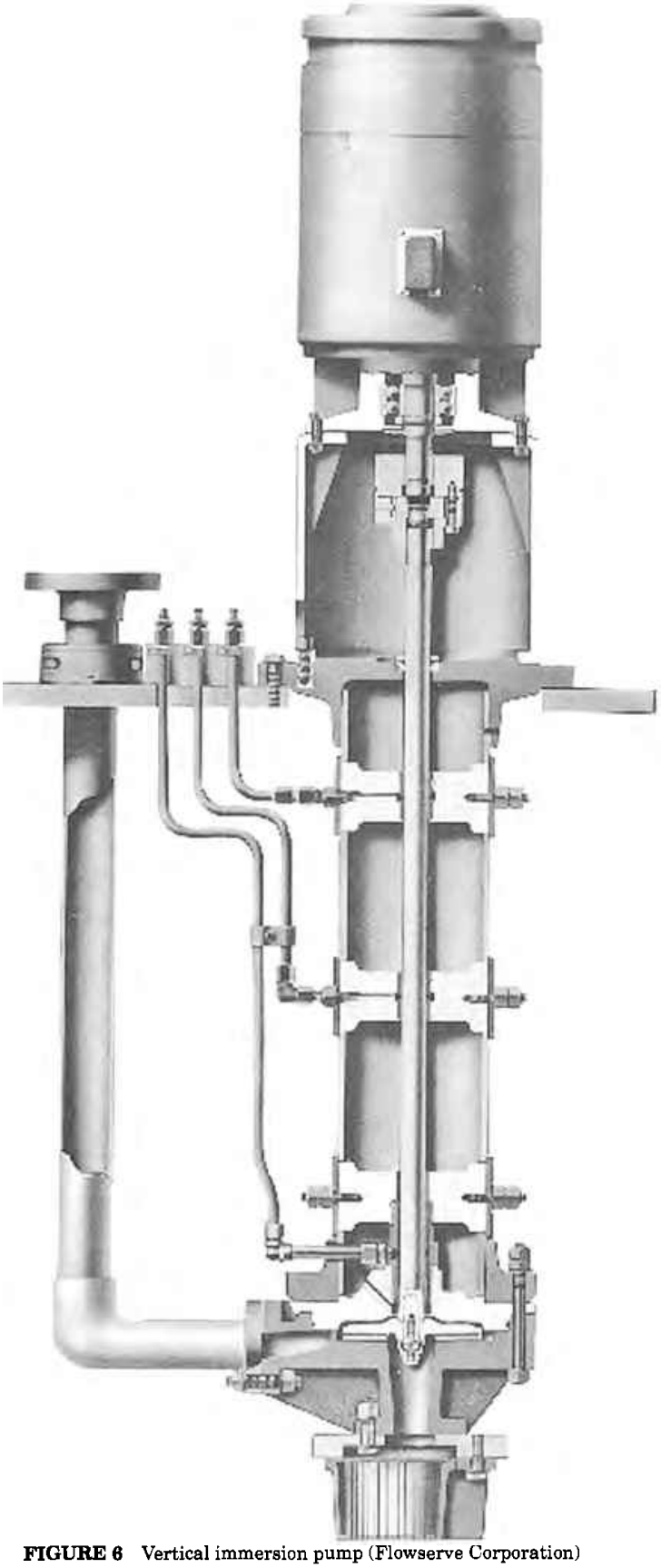


FIGURE 6 Vertical immersion pump (Flowserve Corporation)

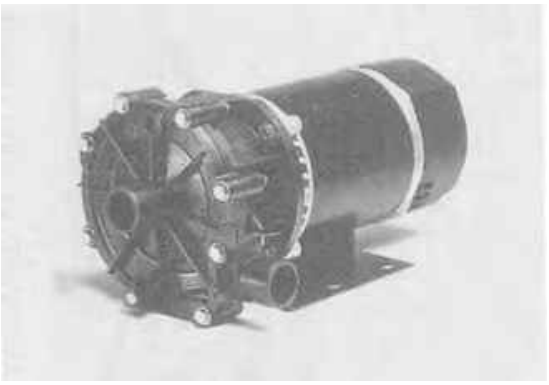


FIGURE 7 Thermoplastic coolant pump (Flowserve Corporation)

Disclaimer The data and notes included in this section are only for general guidance. No recommendations are intended as a guarantee by the author or other sources. Pump manufacturers and material suppliers can supply specific application information for their products when used with specific liquids in identified services and environments.

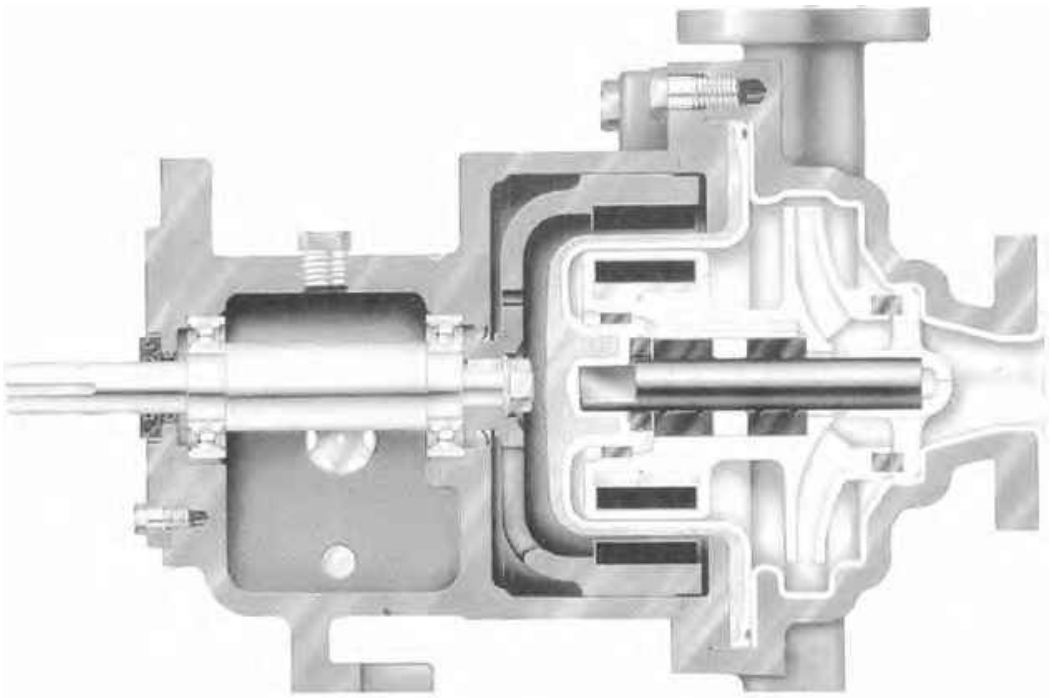


FIGURE 8 Fluoropolymer lined magnetically coupled sealless pump (Goulds Pumps)

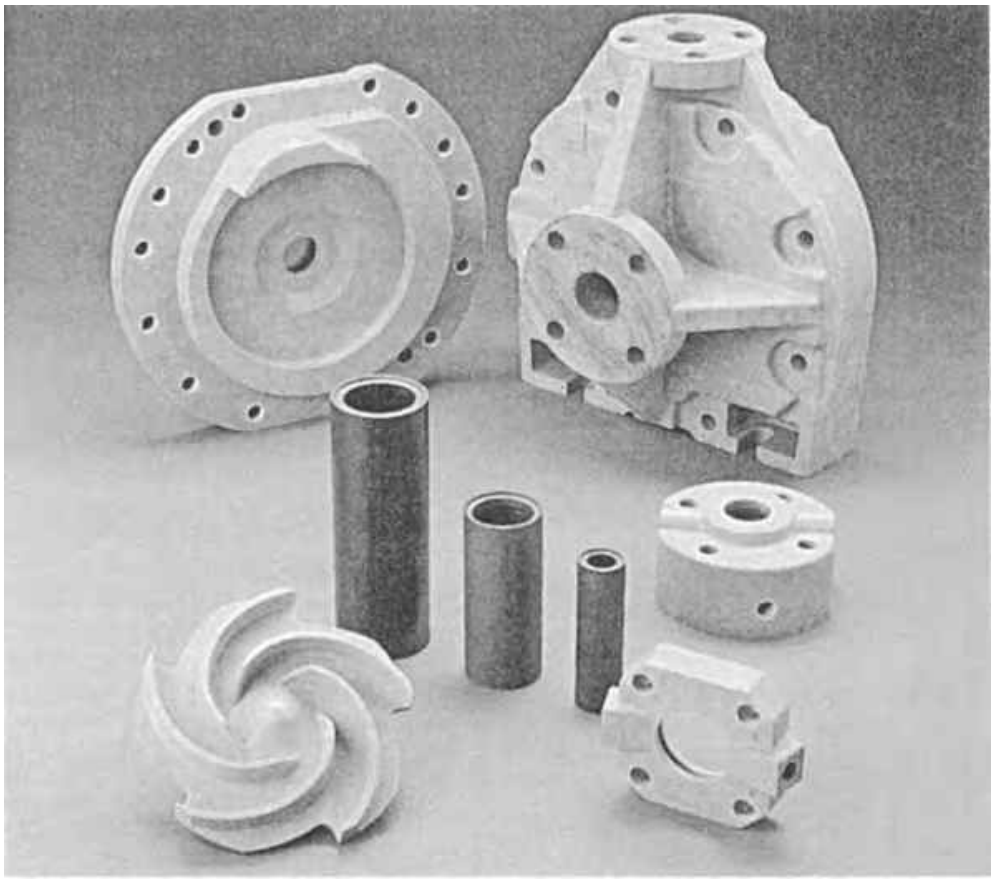


FIGURE 9 Components made of thermoset (Flowserve Corporation)

REFERENCES AND FURTHER READING

American Society of Mechanical Engineers. "Specification for Thermoplastic and Thermoset Polymer Material Horizontal End Suction Centrifugal Pumps for Chemical Process." ASME B73.5M-1995, New York.

American National Standard for Pumps—General Guidelines for Types, Definitions, Application, Sound Measurement, and Decontamination, ANSI/HI 9.1-9.5-2000, Section 9.3.3, Common Polymer Materials of Construction for Various Liquids, Hydraulic Institute, Parsippany, NJ. www.pumps.org.

Besic, D. "Spotlight on Plastics." *Pump and Systems Magazine,* Oct., 1994, Fort Collins, CO.

Buse, F. W. "Tutorial on Composite Pumps." *Proceedings of the 11th International Pump Users Symposium*. March, 1994. Texas A&M University, College Station, TX.

Fusco, J. "Plastic Developments Advance Pipe Performance and Safety." *Chemical Processing,* Dec., 1990.

Henning Kausch, H. "Advanced Thermoplastic Composites." Hanser Gardner Publications, 6915 Valley Ave., Cincinnati, OH, 1992.

Lin, S., and Pearce, E. M. "High Performance Thermosets," Hanser Gardner Publications, 6915 Valley Ave., Cincinnati, OH, 1993.

Margus, E. "Choosing Thermoplastics Pumps." *Chemical Engineering,* July, 1991. McGraw-Hill, New York.

Modern Plastics Encyclopedia '99. The McGraw-Hill Companies, Inc., Highstown, NJ, 1998.

LIBERTY EXHIBIT 1017, Page 827

Plastics Technology Manufacturing Handbook. Bill Communicators Inc., 355 Park Ave., South New York, NY, 1998.

Rosato, D. V., DiMattia, D. P. "Designing with Plastics and Composites—A Handbook." Van Nostrand Reinhold, New York, 1991.

Internet Resources

Injection Molding Magazine's listing of books: http://www.immbookclub.com

Plastic technology consulting source: http://www.rapra.net/intro.htm

Listing of plastic resources and issues: http://www.polysort.com

C • H • A • P • T • E • R • 9

PUMP DRIVERS AND POWER TRANSMISSION

SECTION 9.1
PUMP DRIVERS

9.1.1
ELECTRIC MOTORS AND MOTOR CONTROLS

A. A. DIVONA
A. J. DOLAN
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The two most common types of electric motors are *alternating current (ac) induction* and *direct current (dc) commutated shunt- or series-wound*. All electric motor configurations except one type (reluctance brushless) convert electrical energy to mechanical energy from the magnetic flux linkage of their two magnetic circuits. One of these circuits is in the stator and the other is in the bearing-mounted rotor. This flux linkage between the two magnetic circuits produces a moment of force at the rotor radius that results in a torque on the motor shaft causing shaft rotation. The speed of the rotation times the torque equals the output power at the motor shaft. This is, of course, the power used to drive a pump.

The basic difference between these two types of electric motors has to do with their electrical power source. The first type has been historically powered by 60 cycle (Hz) alternating voltages direct from the public utility power grid (50 Hz in most of Europe and some other parts of the world). For this type of motor, the speed is determined by the number of magnetic poles designed in the motor and the alternating sinusoidal frequency of the voltage from the power grid. Methods have been developed to alter the speed of an ac motor with a fixed number of poles and a fixed line frequency (it is also possible to wind such a motor with several poles for other speeds). However, the squirrel-cage induction motor remains without question the most common type of motor used to drive pumps. The reasons for this are worldwide availability, excellent reliability, excellent performance characteristics, and ease of replacement.

Because of its adaptability to variable speed using ac inverters and vector drives, the ac induction motor will continue to be the number one prime mover for all types of pumps as the pump industry continues to adopt variable speed motor/drives.

The other principle motor type used to drive pumps is the dc motor. The dc electrical machine also contains two magnet fields, one in each of the stator and rotor assemblies of the motor. Dc current creates the rotor field provided through commutator brushes. These brushes have a finite life and are the principal maintenance/failure point in this type of machine. However, in some applications, particularly those where only dc (battery)

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power is available, these machines are common. The dc motor, powered by the voltage from storage batteries or from a dc generator, can be speed-adjusted by varying the voltage with a power supply. In a limited number of instances dc motors are used where adjustable speed is required. Its speed relationship to the voltage is linear and very useful for some pump applications such as constant displacement types, which require speed adjustment to set flow. [See Section 9.2.2.] Most dc motor applications are in the smaller sizes such as automotive and off road equipment. The overall use of dc motors for pump drives is predicted to decline.

A third type of pump prime mover is the permanent magnet machine, sometimes called a brushless dc motor. Unlike dc motors permanent magnet motors have no brushes; unlike the induction machine there is no squirrel cage with induced electric current either; the field is generated by permanent magnets, offering a more reliable prime mover for a broad range of applications, with superior performance and long-life benefits. Permanent magnet motors require a motor drive (inverter, variable speed or frequency drive) to operate; very simple so-called "sensoreless" drives can be used for pumping applications.

There are other types of electric motors considered for driving pumps. The reasons for this include the dramatic recent advances in power electronics and microprocessors, advances in motor materials such as permanent magnets, and advances in the pumps themselves. Besides the active interest in adjustable speed pumps, another reason for the interest in some newer motor types for pump applications is because of recent U.S. government regulations enacted to improve energy conservation by implementing motor efficiency mandates along with a time schedule. For example, the *U.S. Energy Policy and Conservation Act* of September 10, 1992 (EPACT) stipulates that all covered electric motor products must meet the efficiency levels per NEMA MG1 1993. The requirement covers all electric motors from 1–200 horsepower (1–150 kW) manufactured after October 24, 1997 that operate from 230/460 VAC power at 60 Hz line frequency. The U.S. Department of Energy approved test method to meet the new efficiency levels is per IEEE-112B. This regulation was originally intended for ac induction motors.

For those pump applications that require the motor torque to increase as the square of speed, these new, more efficient motors will run at higher speeds because their slip is less. This could cause an appreciable increase in overload within the motor. Overload could also result in other mechanical parts of the system. There are other ramifications resulting from these new requirements that must be carefully analyzed when selecting an electric motor, starter, or inverter. The resulting analysis might very well cause the selection to be some other electric motor type, such as a permanent magnet brushless dc, permanent magnet ac, synchronous (sine wave driven version of the brushless dc), or even a switched reluctance brushless dc motor. Each of these types must be powered with an inverter and a controller. However, the result can most likely offer adjustable speed with a very high efficiency over a wide speed range. This use of an inverter is required for each of those other types of motors mentioned. With the availability of the new vector controlled ac inverters (Section 9.2.2), the high-efficiency ac induction motor can also be applied for variable speed. This will eliminate the need for a soft starter frequently required with line fed motors. The elimination of the starter helps somewhat to offset the additional cost of the inverter.

TYPES OF MOTORS

Alternating-Current Motors

SQUIRREL-CAGE INDUCTION MOTOR By far the most common motor used to drive pumps is the squirrel-cage induction motor (Figure 1). This motor consists of a conventional stator wound with a specific number of poles and phases, and a rotor that has either cast bars or brazed bars imbedded in it.

The squirrel-cage induction motor operates at a speed below synchronous speed by a specific slip or revolutions per minute. The synchronous speed is defined as

$$N = \frac{f \times 60 \times 2}{p}$$

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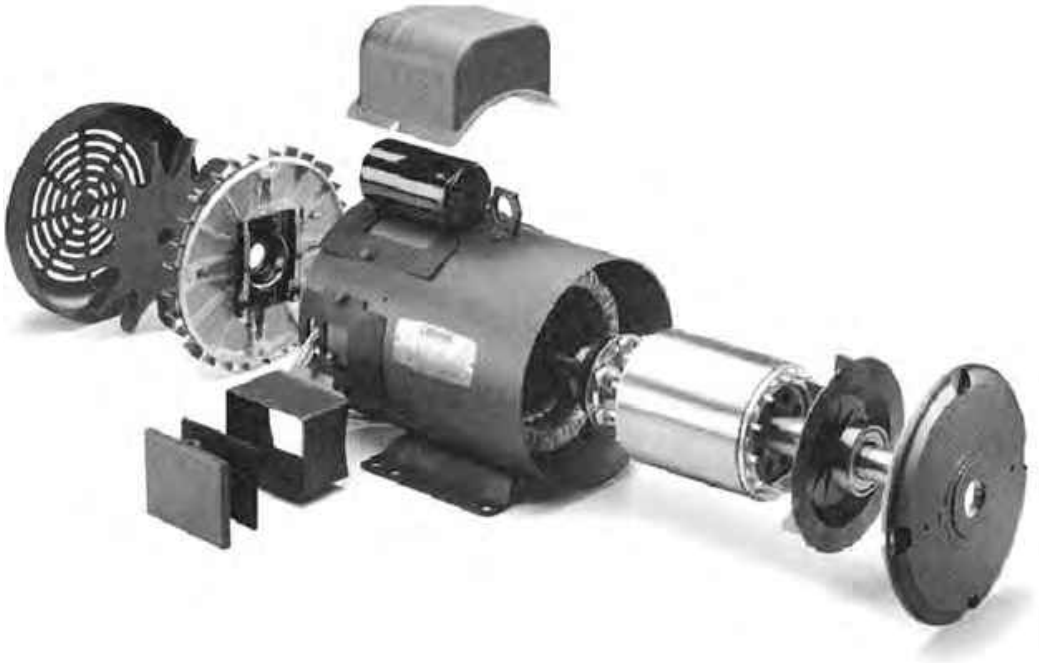


FIGURE 1 Exploded view of a typical squirrel-cage induction motor (Courtesy of Leeson Electric Corp.)

where N = speed, rpm
f = line-power frequency, Hz
p = number of poles

The percent slip is defined as

$$\% \text{ slip} = \frac{(N - s) \times 100}{N}$$

where s = slip, rpm

When the stator winding of a squirrel-cage induction motor is connected to a suitable source of power, a magnetic flux is generated in the air gap between the stator and rotor of the motor. This flux revolves around the perimeter of the air gap and induces a voltage in the rotor bars. Because the rotor bars are short-circuited to each other at their ends (end rings), a current circulates in the rotor bars. This current and the air-gap flux interact, causing the motor to produce a torque.

The squirrel-cage induction motor exhibits a characteristic speed-torque relationship that is determined by the resistance of the rotor bars. Thus, the desired speed-torque characteristics are obtained by selecting a metal of suitable resistance when designing the rotor bars. The slot shape and size for the bars in the rotor can be selected to achieve a certain rotor resistance.

Figure 2 suggests several typical speed-torque characteristics that have been standardized by NEMA (National Electrical Manufacturers Association), covering motor frames 143T through 449T. Motors larger than 449T may not have these same values, but generally have the same characteristic curves. Also, single-phase motors may not exhibit these characteristics and are defined specifically by NEMA with different values.

Most pumps are driven by NEMA B characteristic motors when operated from three-phase power sources.

WOUND ROTOR INDUCTION MOTOR The wound-rotor induction motor is in every respect similar to the squirrel-cage version except that the rotor is wound with insulated wire turns and this winding is terminated at a set of slip rings on the rotor shaft. Connections are made

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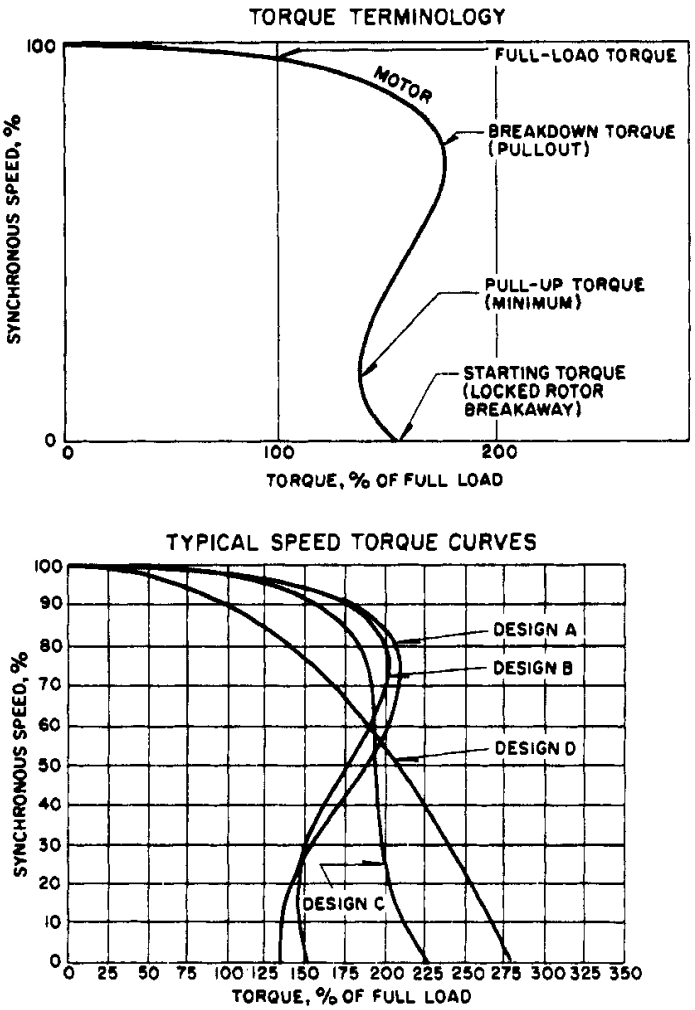


FIGURE 2 These curves are characteristic of NEMA frame size squirrel-cage induction motors through size 449T (typically through 300 hp–224 kW)

to the slip rings through brushes and in turn to an external resistor, which can be adjusted in ohmic value to cause the motor speed-torque characteristics to be changed. These types of motors have been used in some pump applications in the past, but due to the availability of inverter-fed ac squirrel-cage induction motor drives, they are no longer very practical.

Figure 3 demonstrates the speed-torque characteristics of a wound-rotor induction motor for several resistor values. It will be noticed that increasing the external resistance of the control will cause the peak torque of the motor to be developed at lower speeds until the peak torque occurs at zero speed. Increasing the resistance beyond this value will cause the motor to have a limited torque as, for example, curves 4, 5, and 6. This motor can be used where torque control is required or where variable speed is necessary. In the variable-speed application, the rotor resistance is adjusted to produce a motor torque that matches the load torque at the specific speed desired. This system is not as useful and cost-effective as an inverter-fed ac induction motor due to the recent developments of power electronics and microprocessors.

SYNCHRONOUS MOTOR The synchronous motor is also similar to the squirrel-cage induction motor except that it operates at synchronous speed and its rotor is constructed with definite salient poles on which a field coil is wound and connected to a source of direct current for excitation. The most common synchronous motor is constructed with slip rings on the rotor shaft to connect the dc excitation to the field coils.

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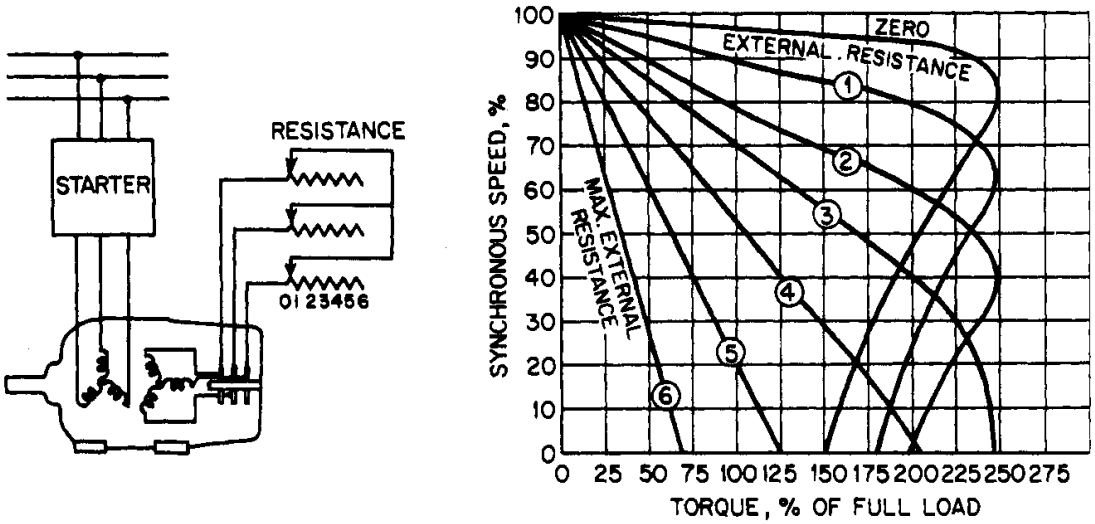


FIGURE 3 Typical speed-torque characteristics of a wound-rotor induction motor (Westinghouse Electric)

There are various means of providing the dc power to the slip rings:

1. **Static excitation** The power to be connected to the slip ring brushes on the motor shaft is obtained from a transformer and rectifier package external to the motor.
2. **Direct-connected exciter** This arrangement has a dc generator directly connected to the synchronous motor shaft (Figure 4). The dc power from this generator is connected to the brushes of the synchronous motor slip rings.
3. **Motor-generated exciter** The dc power for exciting the synchronous motor is generated by means of a remote motor-generator set operating from normal ac power, and the dc voltage from this motor-generator set is connected to the brushes of slip rings of the synchronous motor.

Another form of synchronous motor is known as the *brushless synchronous motor* (Figure 5). As the name implies, this motor has its rotating field excited without the use of slip rings for connecting the external direct current to the motor field. The construction

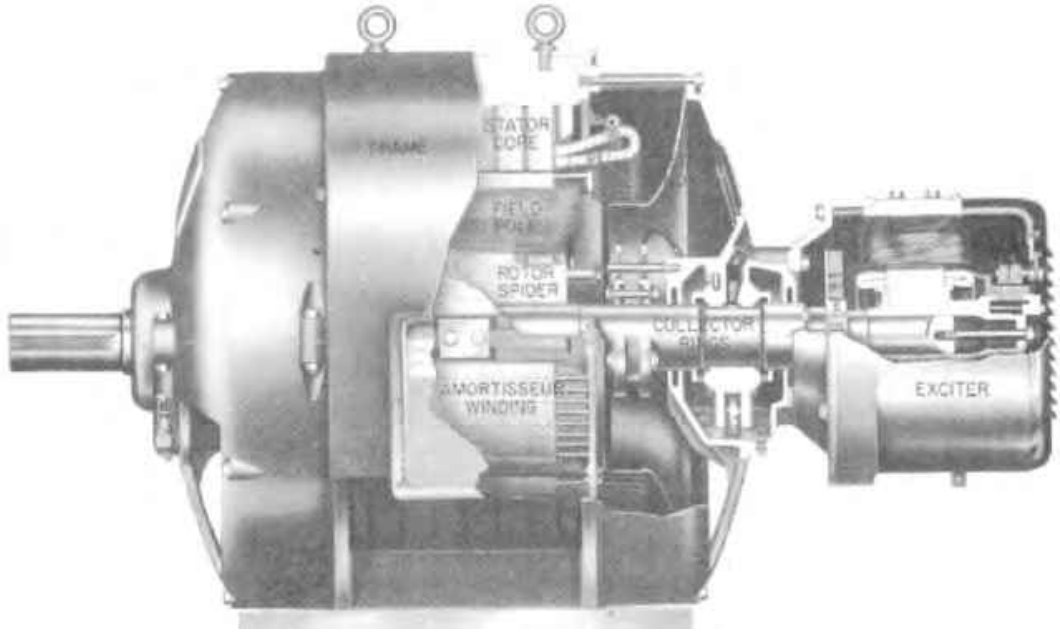


FIGURE 4 Synchronous motor with direct-connected exciter (Electric Machinery Manufacturing)

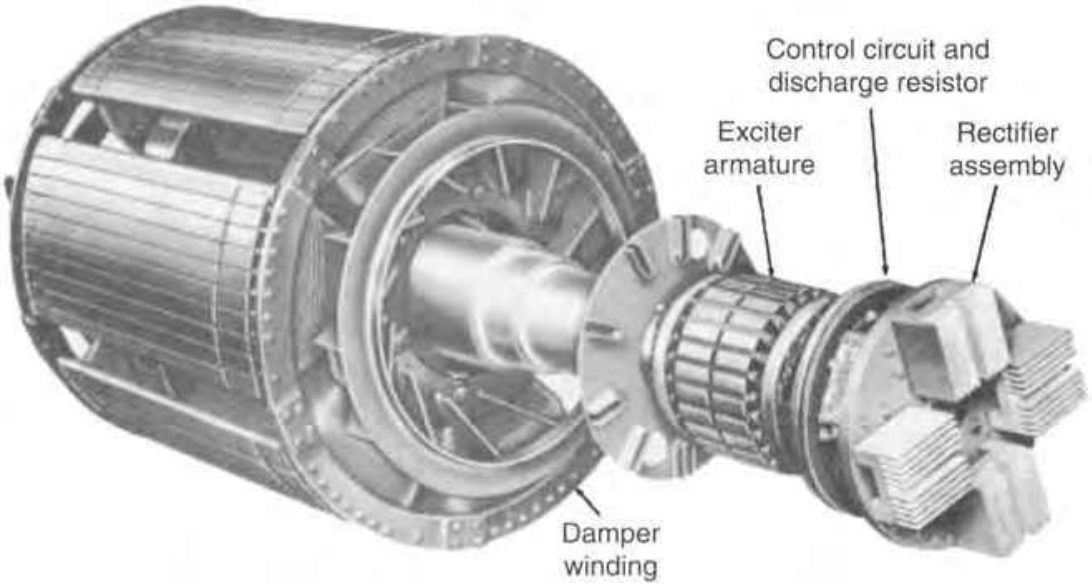


FIGURE 5 Brushless synchronous motor with ac generator mounted on the shaft with rectifier and control devices (Electric Machinery Manufacturing)

of this motor incorporates a shaft-connected ac generator. The field of the ac generator is physically stationary and connected to a source of dc voltage. The rotor of this ac generator is connected through a solid-state controlled rectifier mounted on the synchronous motor rotor and in turn connected to the synchronous motor field. This arrangement facilitates a connection between external excitation power and the rotating field of the synchronous motor through the air gap of the shaft-connected ac generator. The brushless synchronous motor has many advantages over the conventional slip-ring synchronous motor. Among these are the elimination of brushes and slip rings, which are high-maintenance items; the elimination of sparking devices, which are not permissible in certain atmospheres; and the use of static devices for field control, which are more reliable than conventional electromagnetic controls.

Synchronous motors are used for pump applications requiring larger horsepower ratings at lower speed conditions, as illustrated in Figure 6. Also, they are used on applications where a high power factor or a power-factor-correction capability is desired. Of less importance is the characteristic of the synchronous motor that it will always operate at synchronous speed (does not have a slip) regardless of load. Synchronous motors are started on their damper windings (the same as squirrel-cage induction motors), and when they have accelerated to within 5% of synchronous speed, the field is applied and the motor accelerates to synchronous speed (Figure 7). Typical characteristic curves are shown in Figure 8.

Direct-Current Motors Dc motors are only occasionally used to drive pumps. Most of the current applications for dc pump drives are in some form of automobile or off-the-road equipment due to the dc power from storage batteries. There are some other situations that might call for dc motor pump drives such as shipboard duty, railway applications, aircraft, mining installations, and some other emergency battery operations.

There are three types of dc motors available (Figure 9): shunt, series, and compound-connected.

Larger horsepower ratings of shunt-wound dc motors are frequently qualified as "stabilized shuntwound" motors and incorporate a series field similar to that of a compound wound motor. This is necessary to adjust the regulation of the shunt motor so as not to exhibit a rising speed-torque characteristic. It is important to be aware of the speed at which a dc motor will operate on pump applications because of pump performance guarantees.

The windings in the stators of the wound types are connected to the armature windings through brushes and a commutator three different ways to achieve different performance

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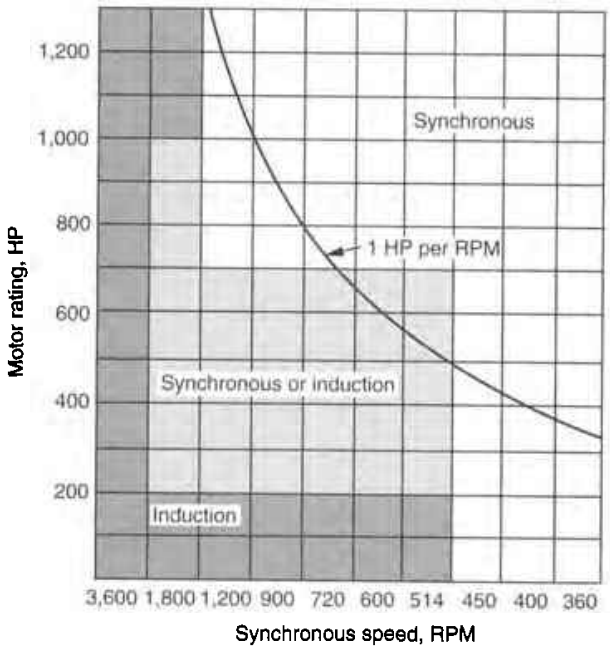


FIGURE 6 Below 514 rpm, or powers greater than approximately 1 hp/rpm (0.746 kW/rpm), synchronous motors are a better selection than squirrel-cage induction motors because higher cost can generally be offset by higher power factor and efficiency (from *Power* special report, "Motors," June 1969)

characteristics (see Figure 3 for details). With the use of pulse width modulation (PWM) or silicon-controlled rectifiers (SCR), variable dc power supplies the armature voltage, which can be controlled along with a separate control for the field current (see Section 9.2.2). Using this control scheme, the dc motor can be used to control speed over a very wide range in a smooth manner. For example, constant voltage can be produced from zero rpm up to base design speed of the motor by armature control and constant power above base speed using flux weakening of the armature field. When operating from a constant voltage, dc motors are available to provide up to a 4 to 1 speed range with an adjustable dc field power supply. With this separate control scheme for both field and armature, a 100 to 1 speed range is easily achievable. Figure 9 summarizes the various

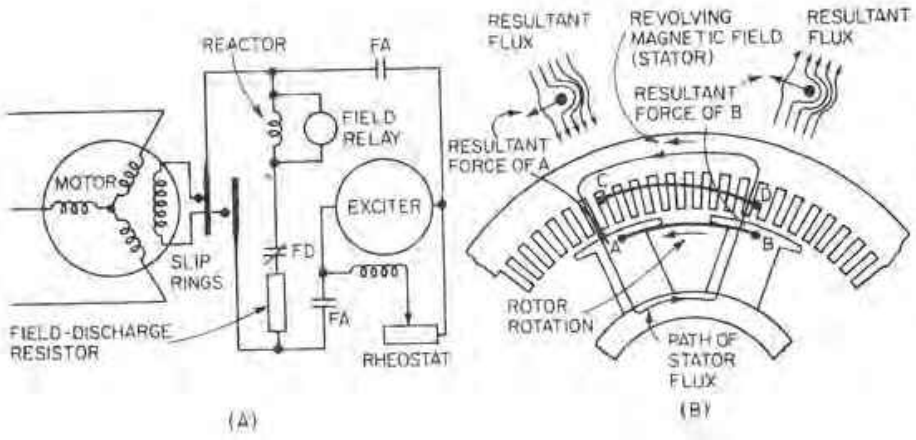


FIGURE 7 Method of starting synchronous motor. (A) Typical field control (brush type) energizes the dc field to pull the rotor into synchronism as it comes up to speed. Field relay senses change in induced frequency as motor speeds up. (B) Damper winding is similar to a squirrel-cage rotor winding. It produces most of the starting torque, and the synchronous motor starts with essentially induction-motor characteristics (from *Power* special report, "Motors," June 1969).

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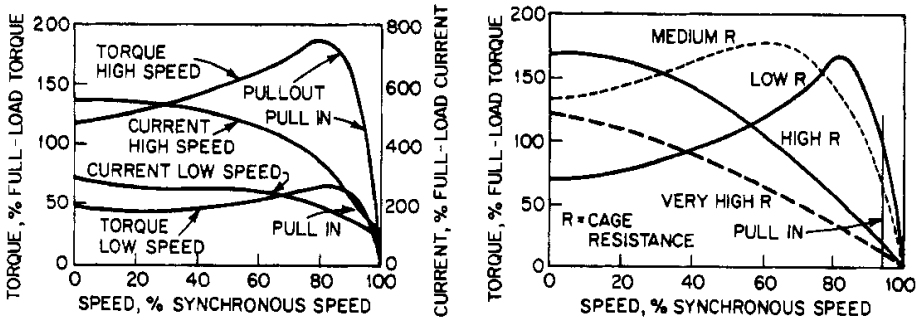


FIGURE 8 Characteristics of synchronous motors depend on rotor design. Torque and current relations are influenced by synchronous speed. High-resistance cage produces high starting torque but low pull-in torque (from *Power* special report, "Motors," June 1969).

field connection schemes for dual wound dc machines with mechanical commutation systems. Figure 10 illustrates typical characteristics of the dc motors discussed. Figure 11 shows the equivalent circuit of a permanent magnet type dc motor.

Permanent Magnet (PM) Brushless Motors The permanent magnet brushless dc motor has been in existence for about three decades and is finally being widely used for many applications[1,5,6]. Its being brushless has frequently been mentioned as justification for its cost, higher than most other motor types. However, the PM brushless motor has two other features that are arguably more important than the fact that it contains no mechanical brushes and commutator for commutation of the phase windings to the power source. First, it produces the highest continuous output power per unit volume of any motor yet invented. The other important virtue of a PM brushless motor is that it produces its output power with the least input power. For a given size and output performance envelope, it has the most efficient motor of any electric motor yet invented. With the emphasis on reducing power consumption, this type of motor will be used in many pump applications. In addition, and perhaps most importantly for the application of pumps, the PM brushless motor possesses one other very important feature or use of the permanent magnets contained within the rotor assembly. Although the magnets are there to supply a magnetic field from the rotor to pass through the air gap of the motor into the stator, these same magnets can serve as a magnetic coupling. If a sleeve-shaped liquid barrier made of a suitable material to be impervious to the liquid is fitted between the rotor and stator, the PM brushless can be used as the best hermetically sealed wet motor known. There are, of course, many possible configurations of this PM brushless hermetic motor concept. These PM brushless hermetic motors can be configured as axial gap[2],

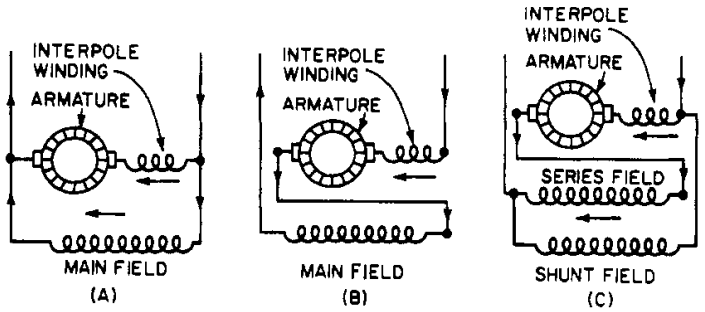


FIGURE 9A through C Types of dc motors. (A) Shunt motor has field winding of many turns of fine wire connected in parallel with the armature circuit. The interpole winding aids commutation. (B) Series motor has field in series with the armature. Field has a few turns of heavy wire carrying full-motor current flowing in the armature. (C) Compound motor has both a shunt and a series field to combine characteristics of both shunt- and series-type motors in the same machine (from *Power* special report, "Motors," June 1969).

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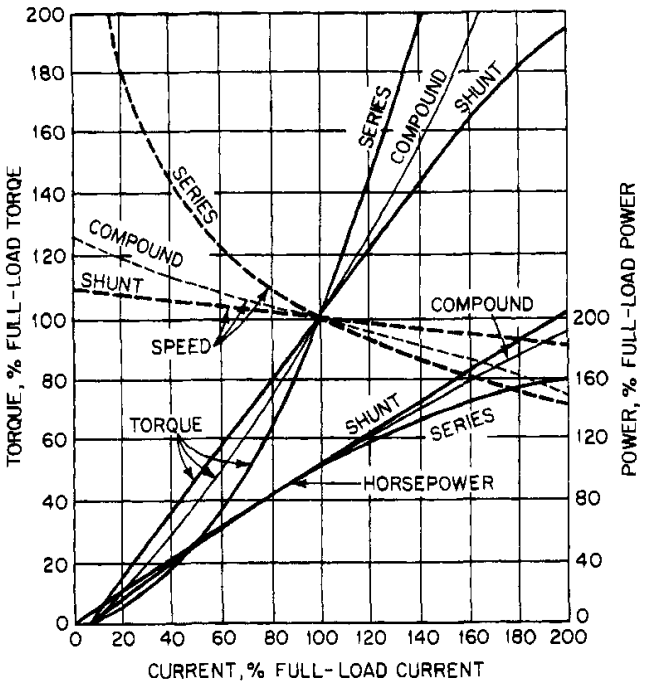
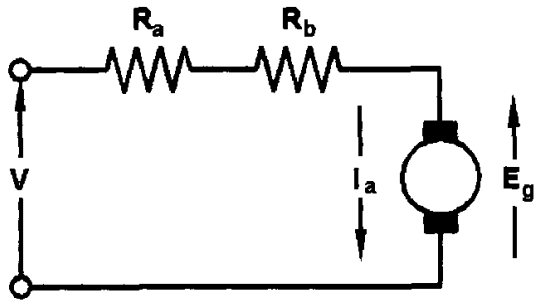


FIGURE 10 Speed, torque, and power characteristics of dc motors (from *Power* special report, "Motors," June 1969)

radial gap inside rotor, or radial gap outside rotor motors (Figure 12). The choice depends upon the best way to design the pump integration.

All PM brushless motors must be powered with an electronic inverter and controller, which can be used for controlling pump performance in controlled loop systems of all types. The electronic drive can be a simple square wave, a six-step trapezoid type, or a sinusoidal drive. The PM motor is a synchronous machine and the controller needs to know the shaft position; in some applications a resolver or encoder is used. However, in many pumping applications a 'sensorless' controller can be used. The choice of controller topology is application specific, but not difficult.

Three permanent magnet choices can be used for these motors. The lowest priced magnet material is known as ferrite or ceramic with a magnetic flux output of about one-third



V = Terminal voltage, volts
R_a = Armature resistance, ohms
R_b = Resistance of the brushes, ohms
I_a = Amature current, amperes
E_g = Back-EMF, volts

FIGURE 11 The equivalent circuit of a dc motor with a permanent magnet stator

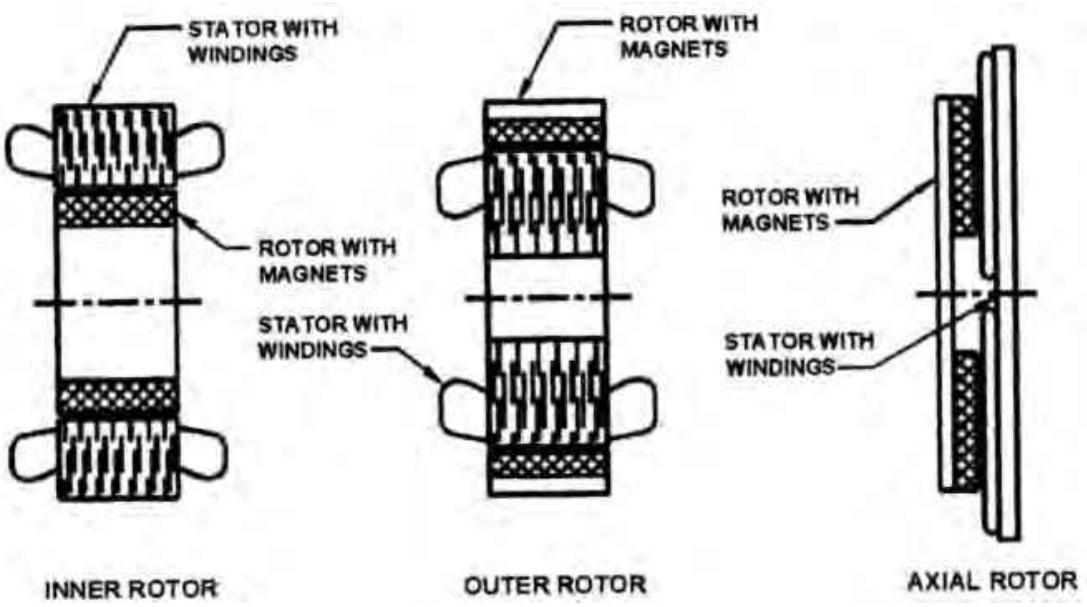


FIGURE 12 Inner, outer, and axial rotor PM brushless motor choices for pump drives

to one-half of the other two magnet choices. By careful design, this magnet material can be used for 50 hp (37 kW) and up in PM brushless machines to keep costs competitive with legacy motors. The two high energy or rare earth magnet materials, Samarium Cobalt and Neodynium Iron Boron, can be used from 1 to 50,000 hp (36 MW). The more expensive of the two, Samarium Cobalt, is used for high temperature and other hostile environment pump motor applications. The newer rare earth magnet material, Neodymium Iron Boron, is much lower in cost, with the highest useful magnetic flux at moderate operating temperatures. Permanent magnets can be assembled onto the surface of the rotor or they can be imbedded within the rotor structure. The imbedded magnet design can be used to direct drive high hp (kW) pumps to several thousand rpm. In either design, the rotor can be made to be quite robust and yield a very long useful operating life. In fact, the only failure modes for PM brushless motors have to do with bearings or winding insulation. Figure 13 shows examples of large PM brushless motors.

The speed can be controlled to greater than 100 to 1 if required with constant power over the highest speed range of any other motor type. The efficiency remains very high over the entire speed range. There are many design possibilities including axial gap and outside rotor configurations. For example, because of the high magnetic strength of the permanent magnets, the rotor of a PM brushless motor can also serve as the impeller of a pump that is integrated with the motor in a hermetically sealed package with no shaft seals. Utilizing the axial-gap rotor configuration of Figure 12, the concept is illustrated in Figure 14. Commercially significant impeller torque levels can be generated by the PM brushless motor in this configuration[3]. This eliminates the need for a separate magnetic coupling or larger-size canned induction motor (see the discussion at the end of this section and in Section 2.2.3).

The PM brushless motor can be driven with either square wave currents over a 120 degrees electrical commutation angle or with sinusoidal currents over 180° electrical commutation angle. The latter scheme generally yields lower torque fluctuations, which is sometimes important for pump applications. Either drive requires shaft angle feedback data to tell the phases when to be powered. This feedback data are also a requirement for vector driven ac induction motors. The sensor used for this feedback can be as simple as *Hall* switches mounted in the motor that send out a pulse to the controller each time a rotor magnetic pole changes polarity as the motor rotates. An optical encoder or a *resolver* can also be used for this purpose. Several sensorless or remote sensors have been developed that capture the rotor angle location from the stator phase windings so a shaft-mounted sensor is not required.

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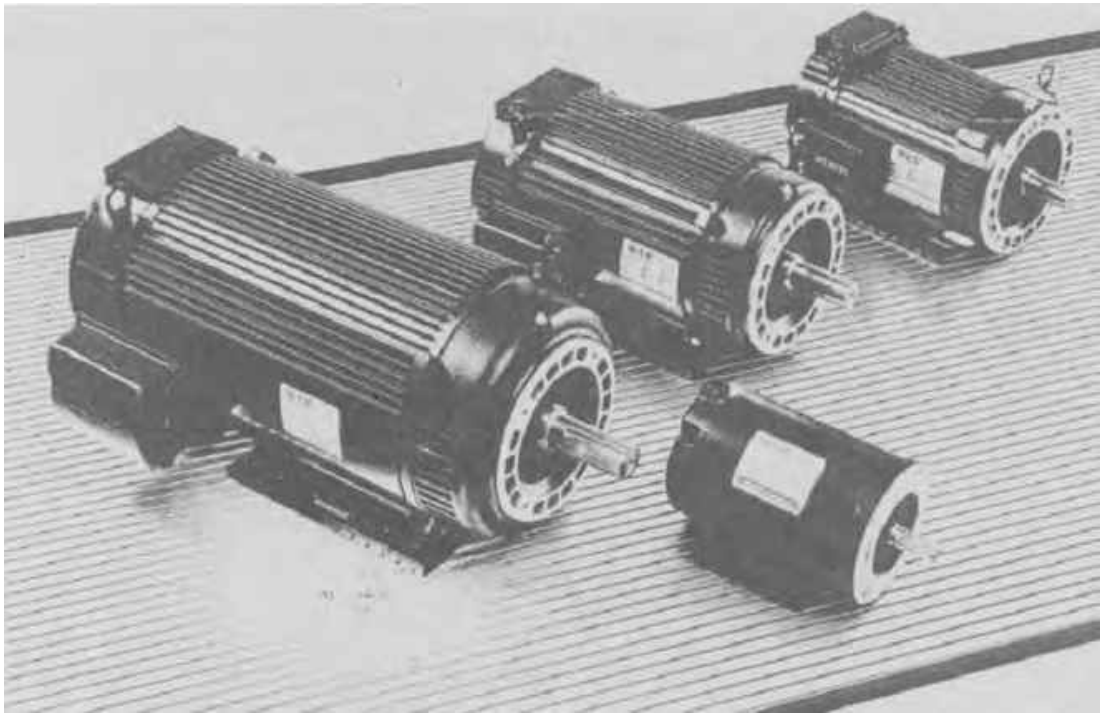


FIGURE 13 Large horsepower (kW) high-performance permanent magnet inside rotor brushless motors (courtesy of Pacific Scientific)

Switched Reluctance Brushless Motor The SR brushless dc motor is one of the oldest motors known, but it has not been used much until recently. Its main feature is that it is a true brushless motor with most of the virtues of its PM cousin, but it does not require permanent magnets[4]. Proponents of switched reluctance motors point out its inherent robustness. The rotor of the SR motor is so simple and rugged it can survive very at high speeds as well as in a variety of other unusual environmental conditions.

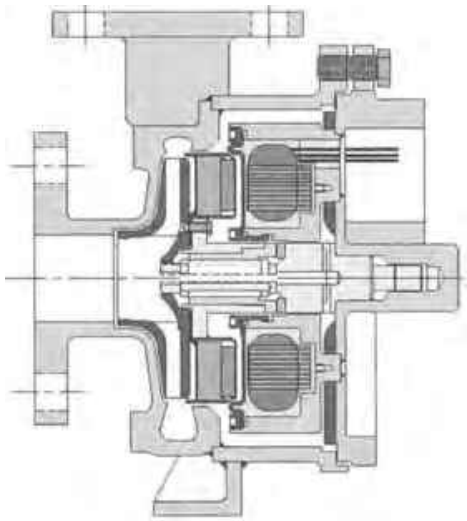


FIGURE 14 Permanent magnet brushless motor integrated into a sealless pump. Permanent magnets are mounted on the impeller, which serves as the rotor of the motor. (Courtesy of Flowserve Corporation)[3]

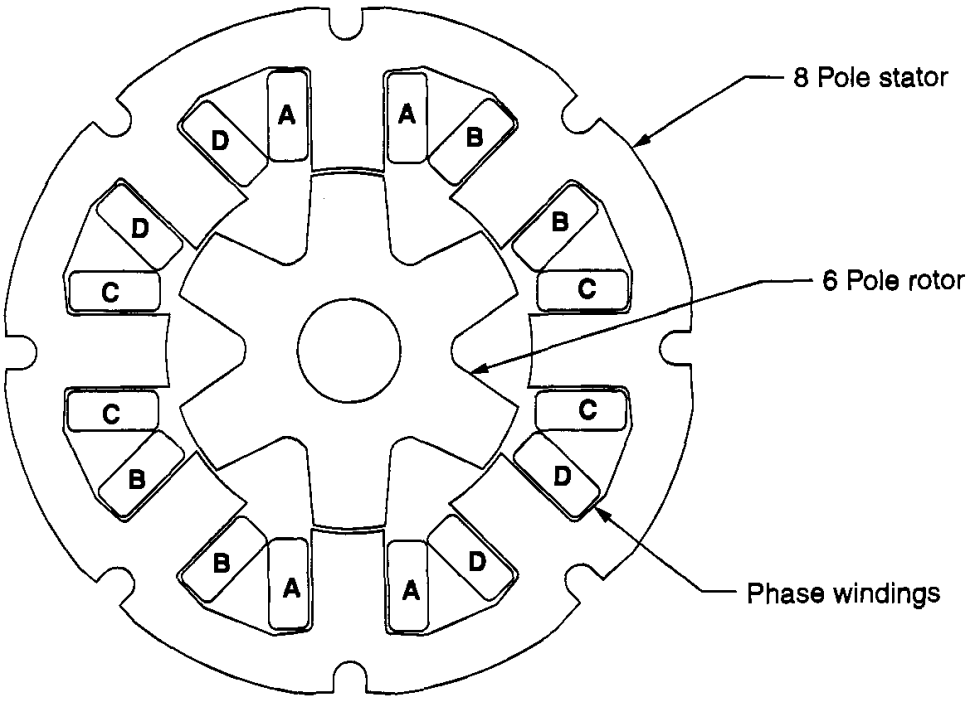


FIGURE 15 Switched reluctance brushless dc motor, (6) rotor poles, (2) stator poles per phase

Figure 15 shows the typical cross section of a switched reluctance motor, illustrating how simple this motor is with phase coils placed around the stator poles. The rotor consists of a set of magnetic steel gear-shaped laminations taken from the bore of the stator laminations. They are stacked and retained on the motor shaft. No magnets or windings of any kind are required to produce torque. The SR motor is said to be a doubly salient pole machine. The torque is produced by the magnetic attraction of the closest rotor poles to those stator poles which are magnetized by the phase coils. There is an abundance of technical information regarding the performance details of this technology. The important point is that the SR motor is an excellent choice for high-speed high-powered centrifugal pumps.

There is a considerable difference in the inverter topology for the SR motor as compared to either the PM brushless or the ac induction motor. The SR machine must be driven as a unipolar machine rather than a bipolar motor like the other two. This means the standard ac inverter cannot be used for the SR motor. The phases are each connected to the dc power in parallel rather in the standard bridge fashion with all of the phases connected together at a center tap. Figure 16 shows the normal power circuit for the (4) phase SR motor shown in Figure 15. The same circuit is applicable for both two phase or three phase. It is essentially an independent half bridge circuit for each phase. Certain two-phase designs are very cost effective for high speed pump drives that are required to rotate in only one direction without the need for reversing.

MOTOR ENCLOSURES

Electric motors are manufactured with a variety of mechanical enclosure features to provide protection to the working parts for specific environmental conditions. Although these special enclosures are classified, they are not, for reasons of economy, available for all size motors. The electric motor industry has incorporated a number of specific enclosure classifications on standard designs of small- and medium-size motors.

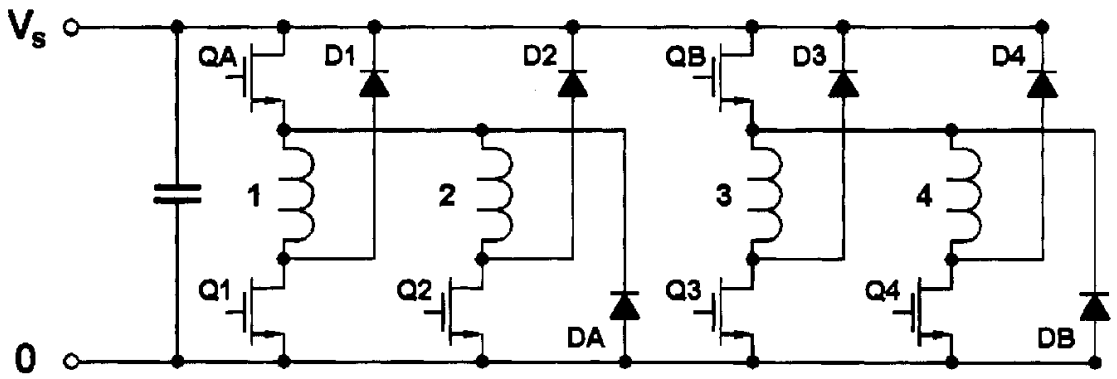


FIGURE 16 Power control circuit for (4) phase SR motor, (6) transistors, (6) diodes, (3) current sensors, (1) capacitor, and (6) motor lead connections

The following list describes the motor enclosure specifications available for polyphase ac induction motors used with pumps. This availability depends on demand. Furthermore, the PM brushless and SR brushless motors might not be available in some of the enclosures listed. In general, totally enclosed motors are required for most integral horsepower (kilowatt) pump applications regardless of the motor type. In some instances, such as high-speed centrifugal pumps with integral motors, the motor is furnished to the pump manufacturer as a rotor and stator parts kit. The motor is then included in the pump enclosure.

OPEN This enclosure permits passage of external cooling air over and around the windings and rotor of a motor and normally includes no special restrictions to ventilation other than those that are inherent in the mechanical parts of the motor. Most designs include openings at each end of the motor frame either in the bearing end frame or openings around each end of the frame.

OPEN DRIPPROOF This is an open machine with ventilating openings designed to permit satisfactory operation when liquids or solids fall on the machine at any angle up to 15° from the vertical. Openings are normally in the bottom portions of the frame assembly. This construction includes mechanical baffling to prevent materials from entering the machine within these limits.

SPLASHPROOF This is an open machine with ventilating openings designed to permit satisfactory operation when liquids or solids fall *directly* on the machine or come toward the machine in a straight line at any angle up to 100° from the vertical.

GUARDED ENCLOSURE This provision limits the size of the ventilating opening to prevent accidental contact with the operating parts of the motor other than the shaft.

SEMIGUARDED This is an open construction where some of the ventilating openings are guarded and the remaining openings are left open.

OPEN, EXTERNALLY VENTILATED This motor is ventilated by a separate motor-driven blower mounted on the motor enclosure (piggyback construction).

OPEN, PIPE-VENTILATED This motor is equipped to accommodate an air-inlet duct or pipe for accepting cooling air from a location remote from the motor. Air is circulated in the motor either by its own internal blower parts or by an external blower, in which case the motor is said to be *forced-ventilated.*

WEATHER-PROTECTED TYPE I This is an open-construction motor with ventilating openings to minimize the entrance of rain, snow, and airborne particles to the motor's electrical parts, and with the openings arranged to prevent the passage of a $\frac{3}{4}$-in (19-mm) round rod.

WEATHER-PROTECTED TYPE II This construction has the same features as the type I machine, but in addition the intake and discharge ventilating passages are designed so high-velocity air and airborne particles blown into the machine by storms can be discharged without entering the internal ventilating passages of the motor leading to the electrical parts. The ventilating passages leading to the electrical parts of the motor are provided with baffles or other features to allow at least three abrupt changes of at least 90° for the ventilating air. The intake air path and openings are proportioned to maintain a maximum of 600 ft/mm (3 m/s) velocity of the entering air.

TOTALLY ENCLOSED This motor is designed without air openings, so there is no free exchange of air between the inside and outside of the motor frame; the construction is not liquid or airtight. Normally the rotor has die-cast air circulating fins at each end to circulate the trapped inside air and improve convection cooling to the frame.

TOTALLY ENCLOSED, FAN-COOLED This totally enclosed motor is equipped with an external fan operating on the motor shaft to circulate external air over the outside of the motor. Most motors have cast fins the length of the frame to increase surface area and take advantage of the fan cooling.

EXPLOSIONPROOF This totally enclosed motor is designed to withstand an internal explosion of gas or vapor and constructed to prevent ignition by the internal explosion of gases or vapors outside the motor.

TOTALLY ENCLOSED, PIPE-VENTILATED This motor enclosure is similar to the open, pipe-ventilated motor except that it is equipped to accept outlet ducts or pipes in addition to inlet ducts or pipes.

TOTALLY ENCLOSED, WATER-COOLED This is a totally enclosed motor cooled by water passages or conductors internal to the motor frame.

TOTALLY ENCLOSED, WATER-TO-AIR HEAT EXCHANGE This is a totally enclosed motor equipped with a water-to-air heat exchanger in a closed, recirculating air loop through the motor. Air is circulated through the heat exchanger and motor by integral fans or fans separate from the rotor shaft and powered by a separate motor.

TOTALLY ENCLOSED, AIR-TO-AIR HEAT EXCHANGE This motor is similar to the water-to-air heat exchanger motor, except external air is used to remove the heat from the heat exchanger instead of water.

SUBMERSIBLE This totally enclosed motor is equipped with sealing features to permit operation while submerged in a specified medium at a specified depth.

Environmental Factors The environmental conditions of the pump application dictate the type of motor enclosures to be used. A brief set of rules for selecting motor enclosures follows.

DRIPPROOF For installation in nonhazardous, reasonably clean surroundings free of any abrasive or conducting dust and chemical fumes. Moderate amounts of moisture or dust and falling particles or liquids can be tolerated.

MILL AND CHEMICAL MOTOR (TO NEMA FRAME 449T) For installation in nonhazardous, high-humidity, or chemical applications free of clogging materials, metal dust, or chips, or where hosing down or severe splashing is encountered.

TOTALLY ENCLOSED, NONVENTILATED OR FAN-COOLED For installation in nonhazardous atmospheres containing abrasive or conducting dusts, high concentrations of chemical or oil vapors, where hosing down or severe splashing is encountered.

TOTALLY ENCLOSED, EXPLOSIONPROOF For installation in hazardous atmospheres containing:

Class I, Group D	Acetone, acrylonitrile, alcohol, ammonia, benzine, benzol, butane, dichloride, ethylene, gasoline, hexane, lacquer-solvent vapors, naphtha, natural gas, propane, propylene, styrene, vinyl acetate, vinyl chloride, or xylenes
Class II Group G	Flour, starch, or grain dust
Class II, Group E	Metal dust including magnesium and aluminum or their commercial alloys
Class II, Group F	Carbon black, coal, or coke dust

NOTE: Under Class 1 only, there are two divisions that allow some latitude on motor selection. Generally, Class 1, Division 1 locations are those in which the atmosphere is or may be hazardous under normal operating conditions, including locations which can become hazardous during normal maintenance. An explosionproof motor is mandatory for Division 1 locations. Class 1, Division 2 refers to locations where the atmosphere may become hazardous only under abnormal or unusual conditions (breaking of a pipe, for example). In general, a motor in a standard enclosure can be installed in Division 2 locations if the motor has no normally sparking parts. Thus, open or standard totally enclosed squirrel-cage motors are acceptable, but motors with open slip rings or commutators (wound rotor, synchronous or dc) are not allowed unless the commutators or slip rings are in an explosionproof enclosure.

BEARINGS AND LUBRICATION

Very large horsepower (kilowatt) motors are generally supplied with oil-lubricated sleeve bearings with oil supplied from a reservoir. In some cases, pressurized oil lubrication systems are installed by the pump manufacturer along with hydrodynamic thrust bearings. All NEMA frame induction motors are available with ball bearings. These standard ball bearings are normally permanently grease lubricated. The bearings used in a motor must be sealed to keep the lubricant inside the bearings and keep contaminates from getting into the bearings. Double-sealed bearings are common for many pump applications.

Ball bearings are subject to early failure when used in electric motors driven by PWM inverters. This very common problem must be addressed. It is caused by the high carrier frequency used in the inverter to generate the sinusoidal currents for each phase. This results in generation of high common-mode voltages inside the phase windings of the stator. Because there is an excellent electrostatic coupling between the stator/frame and the rotor from the windings, a voltage is induced in the shaft. The ball bearings represent the least-resistant path for a short circuit to the stator. However, the balls seldom actually contact the races because of the film of grease or oil in between. When the voltage builds up in the shaft until it is greater than the insulating capability of the film of lubricant, the voltages arc across the lubrication gap and a flashover current goes through the bearing. In a relatively short amount of time, the bearing races will become grooved, causing the bearings to become noisy. Metal particulate will then egress from the bearing surfaces as the process continues, causing catastrophic bearing failures after a few months.

Therefore, all electric motors that are driven by PWM drives must have a shaft grounding system to provide a low resistance path between the shaft and the motor

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frame. There are other solutions to this potential problem, which can be discussed with the motor supplier.

Sleeve Bearings Motors that use oil-impregnated porous sleeve bearings are lubricated with an oil-soaked wick. These bearings are available in motors up to approximately 1 hp (0.75 kW). Sleeve-bearing motors larger than 1 hp (0.75 kW) are ring-lubricated. Lubricating oil is drawn up from the bearing sump to the bearing by a ring that rolls over the top of the motor shaft as the shaft rotates. Larger motors, having bearing heat losses that cannot be dissipated directly, may require the use of a pressurized lubrication system wherein oil is pumped into the bearings and allowed to recirculate through a heat exchanger. The oil delivered to each bearing is metered to provide only the required amount. A lubrication system composed of heat exchanger, sump, and pump is normally common to a number of bearings, rather than having a single lubricating pump for each bearing. Other types of bearings must be used in place of, or in addition to, sleeve bearings when thrust loads are present. Smaller sleeve bearings are in the form of a cylindrical shell and are usually made of bronze or steel-backed babbitt metal. Larger sleeve bearings are usually split on a horizontal centerline, allowing easy assembly and disassembly for inspection and replacement. The bearing housing is also split on the horizontal centerline and held together with bolts between the top and bottom halves.

Rolling Element Bearings All new electric motors manufactured to NEMA standards use grease-lubricated ball bearings with high radial and thrust load capacities. They are axially pre-loaded to eliminate any radial or axial play for quiet operation and long life. Most motor end frames include an outer race locking plate on the shaft end bearing to prevent race rotation due to output shaft loads. These bearing mounting features are required for high performance, long life, and high efficiency operation. Most motor manufacturers provide their larger frame sizes with grease fittings for relubrication during the lifetime of the motor. The smaller frames use bearings that are grease-packed and sealed for life at the factory and cannot be relubricated.

NEMA motors subject to very high loads and operating temperature in the larger frames may require oil lubrication to the rolling element bearings. This can be by either oil circulation within the bearing frame or from a pressurized lubrication system similar to that used with sleeve bearings. In addition, for motors required to carry very high thrust loads, quite often in only one direction, taper or spherical roller thrust bearings may be used.

Hydrodynamic Thrust Bearings Certain types of pump applications, such as very large vertical pumps, exhibit very high thrust loads that cannot be accommodated by rolling element bearings. Hydrodynamic bearings, usually with tilting, self-leveling thrust pads, are used for very high thrust loads. These bearings are sometimes referred to as "Kingsbury-type" in recognition of the original manufacturer of this bearing type. This type of bearing is oil-lubricated from a self-contained oil sump or an external pressurized pumping system, depending on size and rating.

MOTOR INSULATION

Insulation systems must be used in a motor to electrically insulate the windings from the mechanical parts of the motor as well as to insulate the phase winding conductors from one another. Extra insulation is also required between any adjacent phase windings for 230 or higher voltage motors. In general, the higher the operating voltage, the better the insulation system must be.

All NEMA-designed electric motors are manufactured with cord laced end turns to assure that they are positioned so they cannot touch any mechanical parts. In addition, the phase windings are nearly always impregnated with a varnish by either dipping or trickling into the heated end turns. The purpose of this is to secure the conductors to prevent them from vibrating between one another, which would wear through the wire insulation

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and cause turn-to-turn shorts. This stator varnish also provides some additional insulation protection to the system. For example, this coating makes the windings resistant to ambient conditions such as moisture. Consequently, a motor insulation system is very complex, utilizing several different materials, parts, and processes to effectively insulate the windings. The parts considered in an insulation system include slot cells, phase barriers, conductor insulation, slot wedges, end turn supports, tie material, and winding impregnation material.

Since the introduction of PWM inverters for use with all types of variable speed drives, such as ac induction, PM brushless and switched reluctance brushless, a new insulation system failure mode has emerged. This is the most significant problem in installations where the inverter is located 50 ft (15 m) or more away from the motor. The long motor lead cables used in these applications cause very high voltage spikes not present at the inverter end of the lead cables. These voltage spikes across the first turns of each phase winding result in the degradation of the wire insulation due to the corona insulation failure. Special magnet wire insulation and motor manufacturing methods are available to overcome this problem. Therefore, if an inverter is to be used to drive a pump motor with long connection cables, an "inverter duty" motor should be specified.

There are four basic classes of insulating materials currently recognized by the motor industry. Each differs according to its physical properties and can withstand a certain maximum operating temperature (frequently termed total *temperature* or *hot-spot temperature)* and provide a practical and useful insulation life. The insulation classes and their maximum operating temperatures are

Class A	90°C
Class B	130°C
Class F	155°C
Class H	180°C

Those factors that contribute to the maximum operating temperature of a motor insulation system are the ambient temperature, the temperature rise in the motor winding caused by motor losses, and any overload allowance designed into the motor (service factor).

The current standard for motors within the range of NEMA ratings (frames 140T to 449T) requires nameplate marking for the maximum allowance ambient temperature, the power rating, the associated line current needed to develop this power, the class of insulation used, and the service factor provided. Motors larger and smaller than NEMA frame sizes have nameplate marking for maximum allowable ambient temperature, temperature rise in degrees Celsius either by thermometer or resistance measurement, power rating, line current needed to develop this rating, and any service factor provided.

In addition to ambient temperature, there are several additional environmental conditions that must be considered when applying an electric motor. In applications where chemical fume or moisture levels are abnormal and can cause decomposition of an insulation system, standard insulation will be inadequate. These applications require a motor with a premium insulation system that will incorporate highly resistant components and may include special impregnation techniques. Chemical fumes and moisture can also be destructive to the mechanical parts of a motor, and special protective treatment should be provided to these parts. NEMA frame-size motors have a special motor for chemical industry applications that has standard features to resist these environmental factors. For example, these motors usually contain shafts made from stainless steel.

Applications with excessive vibration can destroy a winding and damage the mechanical parts of a motor. In such cases, it is advisable to provide (1) extra treatment for the winding to ensure that it is rigid and will not vibrate and chafe the insulating materials and (2) a mechanical construction that will have the strength to withstand the above.

If abrasive dust is present, the motor insulation should be protected with a resilient surface coating to withstand the impact of the abrasive particles.

Because all insulation systems employ components that can in some degree support fungus growth in tropical locations, motors applied in such areas should incorporate fungus-proofing treatment on the insulation.

Obviously, applications exhibiting a combination of any or all of the environmental conditions discussed should have special protection for each condition. Applications in unusual environmental conditions may require special protection and should be discussed with the motor manufacturer or distributor.

COUPLING METHODS FOR PUMP APPLICATIONS

Direct Coupling Pumps are frequently directly coupled to motors, and where the pump is not close-coupled, it is usually coupled by means of a flexible coupling. The use of a flexible coupling permits minor misalignment (angular and parallel) between motor and pump shafts. The use of some older style flexible or solid couplings could cause severe radial and axial loads on the motor bearings. Since the development of the flexible disk couplings, the earlier coupling designs have largely disappeared, as have the severe loads resulting from misalignment. The flexible disk coupling is capable of transmitting very high torque for its size, with minimum radial and axial forces on the shafts resulting from misalignment. [See Section 9.3]

Close-Coupling Close-coupled pumps have become very popular for certain applications. In this arrangement, no coupling is provided between the pump and motor shafts and the pump housing is flange-mounted between close-tolerance fits on both the motor and the pump flanges. The pump impeller is mounted directly on the motor shaft. Care must be taken in this arrangement to ensure that the motor shaft runout or axial movement plus machine tolerances do not cause interference between the pump housing and its rotor. This is usually not a problem if properly fitted ball bearings are used in the motor. The motor shaft material must be compatible with the fluid being pumped, and if the pump impeller is held in place by a nut, the threat must respect the rotation of the motor. High-pressure close-coupled pumps of a nonbalanced design can cause excessive shaft thrust, which may be incorporated in the motor bearing capacity. It is always good practice with close-coupled pumps to provide some form of flinger on the motor shaft to prevent liquids that leak past the pump seal from entering the motor bearing.

Flanged Motors Flanged motors allow an easy means of aligning pump housings with motors. This construction is usually in the form of a vertical mounting in which the motor is set on top of the pump and the pump supports the motor weight. The pump and motor shafts are normally coupled, and those comments made under the subject of direct-coupling methods are applicable. Also, as in the case of coupled pumps, this construction permits thrust forces that must be considered when selecting a motor if the pump does not have a thrust bearing.

A further extension of flange motors includes the vertical hollow-shaft motor. With this design, a variable length of shafting connects the pump and motor. The pump shaft passes through the center of the motor bore, and the motor torque is imparted to the pump shaft by a suitable coupling at the top of the motor. The weight of the shaft and the pump impeller and the force of the hydraulic thrust are assumed by the motor bearings.

The coupling on the motor can be made a *self-release coupling* (Figure 17) to prevent the motor from delivering torque to the shaft in the event the motor is started in the wrong direction and to prevent reversed motor rotation from unscrewing the threaded joints between lengths of pump shafting.

Another modification to a coupling is a nonreverse *ratchet* (Figure 18), which prevents the remaining head of liquid in a pump from rotating the pump in the reverse direction when the pump is stopped. This prevents possible overspeeding of the pump and motor when the pump is connected to a large reservoir and most of the total pump head is static. This also prevents a pump with a long discharge column pipe from running in reverse with no liquid in the upper portion of the pipe to lubricate the line shaft bearings. Starting a pump capable of back spinning is also prevented.

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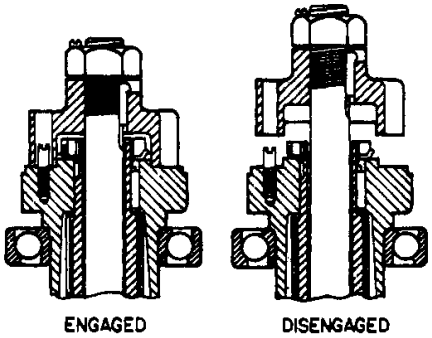


FIGURE 17 Self-release coupling connecting pump head shaft to hollow shaft of vertical motor disengages as a result of pump shaft couplings unscrewing (U.S. Electrical Motors)

PERFORMANCE

Ac induction motors are designed to produce their rated power at a certain speed at a specific line voltage, line frequency, and ambient temperature. These motors will also operate at a specific efficiency and power factor when all of these conditions are met. The normal operating conditions of a motor are stipulated on its nameplate with values for power,

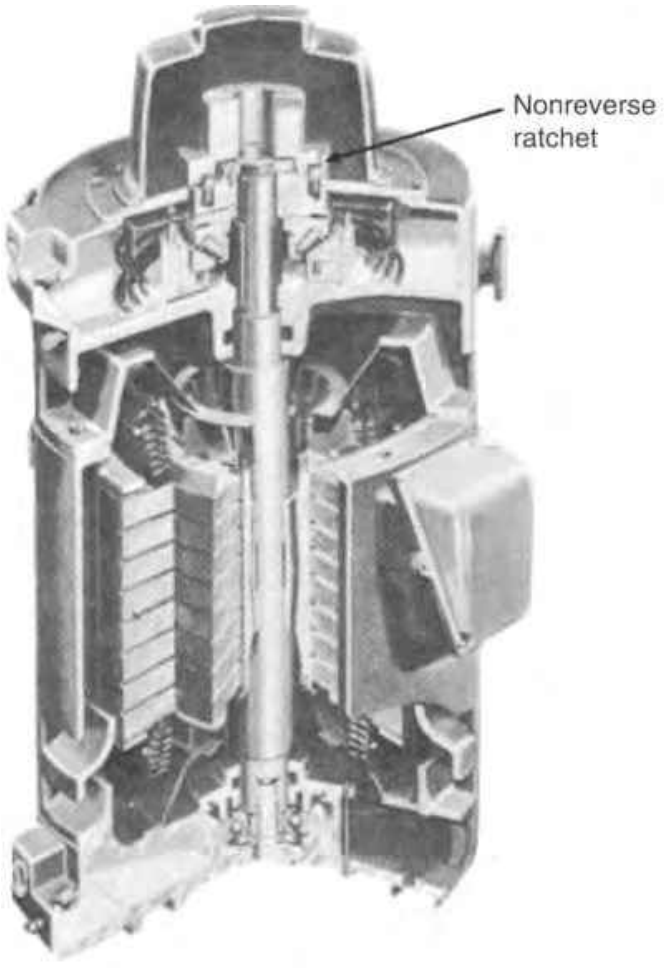


FIGURE 18 Section of vertical hollow shaft motor showing nonreverse ratchet. Spring-loaded pins ride on ratchet plate in one direction only (General Electric)

speed, ambient temperature, and frequency. If the operating conditions are different from the nameplate ratings, the motor performance will be altered. Many of the newer PM and SR brushless motors are designed to produce their rated output power over a wide speed range. A high efficiency ac induction motor can also perform similarly if a Flux Vector Inverter is used for control.

Voltage Ac motors are designed to operate satisfactorily at their nameplate voltage rating with a ±10% variation from nameplate voltage when operating at rated nameplate frequency. This dictates that the motor will develop rated power and speed to a pump and will operate at a safe insulation temperature over the range of voltage. The motor torque will vary directly with the square of the applied voltage divided by the nameplate voltage. This affects the peak torque of the motor and will cause the motor speed-torque curve as shown in Figure 2 to be altered. Within the ±10% voltage band, a motor can be expected to accelerate and operate a pump safely and continuously. A motor should never be expected to operate continuously beyond the ±10% band. If the voltage varies more than ±10%, the pump and motor may not operate satisfactorily.

For example, assume a pump is operated by a NEMA design B motor (refer to Figure 2) that will produce 200% pull-out torque at rated voltage. If the line voltage were to fall to 70% of the rated name-plate voltage, the motor would produce only 49% of its peak torque value. The pull-out torque of the motor would then become $0.49 \times 200 = 98\%$ of rated torque. It then becomes very doubtful whether the motor will be able to sustain the pump load, and the motor can be expected to lose speed, stall, or become overloaded. Certainly, it will also heat up, which will shorten the expected life of the bearing lubrication and winding insulation system.

In a similar sense, a motor may be unable to accelerate a pump if low line voltage exists. In the example previously discussed, this same motor develops 150% of rated torque when started at zero speed and rated voltage. If the line voltage is again 70% of nameplate voltage, the motor will develop $0.49 \times 150 = 73\%$ of rated torque. This may be a problem with certain types of pumps, such as a constant displacement pump. It is conceivable that this would not be a problem in starting a centrifugal pump because of its square-law speed-torque characteristics. If the motor voltage never increased beyond 70%, the centrifugal pump would not reach normal operating speed. An exception to this rule is the commutating ac motor, for which a ±6% voltage variation is allowable.

Varying motor voltage from nameplate rated voltage will also affect the motor operating speed, power factor, and efficiency established for rated voltage and load. Most polyphase ac induction motors will operate at several rpm faster than nameplate speed at 10% over voltage and several rpm below nameplate speed at 10% under voltage. The speed of a synchronous motor is determined by the line frequency of the alternating voltage not the voltage level. Therefore, voltage variation has no effect on the speed of a synchronous motor. However, voltage variation does effect maximum torque output of a synchronous motor.

Dc motors can also be operated over a ±10% voltage range from nameplate rated voltage. However, it should be recognized that different types of dc motors will have different speed and torque characteristics over the voltage range. This should be taken into account when meeting pump performance requirements.

One of the most common applications for dc motors has been for metering pumps because of their ease of setting the speed to precisely meter the fluid being pumped. Although a voltage variation would change the pump speed, it is customary to include a speed control loop with the dc supply to maintain constant speed in spite of line voltage variations.

Frequency Ac induction motors will operate satisfactorily at rated load and voltage with a frequency variation up to ±5% from rated nameplate frequency. However, the speed of the motor will vary almost directly with the line frequency. The speed of a synchronous motor will vary directly with applied frequency. The combined variation of voltage and frequency must not be more than ±10% from rated nameplate voltage and frequency, provided the frequency variation does not exceed ±5% from rated nameplate frequency. The reason is that frequency variation from the nameplate frequency will cause motors to

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operate at a power factor and efficiency other than those established for rated frequency. Any electric motor such as PM or SR brushless and even ac induction, which is powered from an inverter, is unaffected by frequency variations.

Speed and Speed Range Synchronous and induction motors were originally expected to be operated at one specific speed when powered from line voltage and frequency. However, with the availability of inverters, this is no longer the case. In fact, the use of inverters for driving induction motors is expanding at a fast pace as the merits and benefits of adjustable speed are being realized. The use of speed controlled PM and SR brushless motors along with their inverters and controllers allow the pump application to take advantage of the usefulness of adjustable speed. Using the inverter with any of these types of motors eliminates any difficulties from speed variations from the past. This is because most inverters have a speed control loop that will automatically maintain the set speed.

Acceleration A motor must be capable of accelerating as well as driving a pump at rated speed and power. The acceleration can be analyzed by examining a typical pump-motor combination involving a centrifugal pump driven by a six-pole squirrel-cage induction motor rated 10 hp (7.46 kW) with NEMA design B torque characteristics. The curve in Figure 19 demonstrates this combination when the pump is loaded and the motor is operating at nameplate frequency and voltage. It will be noted that the torque produced by the motor at any speed must be greater than the torque required by the pump when up to speed. The excess torque at any speed is available to accelerate the motor, coupling, and pump rotating parts.

NEMA has provided a handbook for motion control that outlines the fundamental equations for the required torque calculations to accelerate any inertial loads to a predetermined speed. For the use of ac induction motors that are line-fed with constant voltage and frequency, the formulas are used to calculate the time to the motor slip from synchronous speed for a given total rotating inertia. For the use of inverter-fed ac induction, PM or SR brushless motors, the formulas are used to determine the torque required to accelerate the total rotating inertial load to the desired speed in the desired time.

The fundamental equations for motion, which include torque, inertia, time and acceleration relationships, are given below. They can be used for motor selection analysis for pumps. For example, if the torque available from an induction motor is known, the equation can be solved for the acceleration time of the system inertia (from NEMA "Programmable Motion Control Handbook").

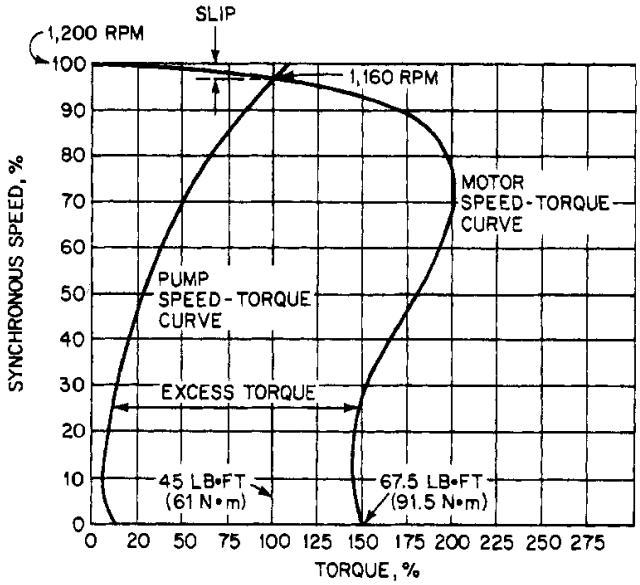


FIGURE 19 Typical speed-torque characteristic curves for a centrifugal pump and a squirrel-cage induction, NEMA design B motor. Excess torque accelerates pump.

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$T_{TOTAL} = T_A + T_C$: Total torque = Acceleration torque + constant torque

$T_A = J_{\text{TOTAL}} \times \alpha$: Acceleration torque = Inertia × Angular acceleration

$\alpha = \dfrac{\omega_{\max}}{t_a} \times 2\pi$: Angular acceleration = Max RPM/acceleration time

T_C= Torque from all other constant forces, friction, windage, preload, and so on.

NOTE: Use consistent units depending on whether U.S. Customary or SI.
The time to accelerate the pump is equal to

in USCS units $$t = \frac{WK^2 \times \Delta\text{rpm}}{308T}$$

in SI units $$t = \frac{MK^2 \times \Delta\text{rpm}}{9.55T}$$

where t = time, s
WK^2 = total weight (force) moment of inertia, lb · ft²
Δrpm = change in speed
T = torque, lb · ft ($N \cdot m$)
MK^2 = total mass moment of inertia, kg · m

Because the difference in torque is not uniform over the speed range, the curve (Figure 19) can be analyzed by assuming discrete changes in speed and an average torque over this change in speed. A time period can be calculated for each discrete speed change, and all time values can be totaled to obtain the complete acceleration time.

Increasing the inertia of the pump, the operating speed of the pump and motor, or the torque required by the pump at any speed will result in a longer acceleration time, which may not be possible for the motor. Each motor can operate at a reduced speed and at a torque in excess of rated torque for a given time. Beyond this time, the windings and/or rotor can be damaged. The use of an inverter-driven ac induction SR or PM brushless motor eliminates all of these problems associated with the use of line-fed ac induction motors. This is due to the torque control capability of the inverter/motor drive system using current control. The limit on the ability of the adjustable speed drive to control acceleration is a function of the current capability of the inverter (as long as the thermal limit of the motor has not been exceeded).

When an application is analyzed and found to present an acceleration-time problem, consideration should be given to

1. Unloading the pump during acceleration
2. Reducing the inertia of the rotating parts
3. Selecting a larger motor
4. Considering an adjustable speed motor and drive

When very large pumps are started, the line voltage will sometimes drop because of the high starting current, which causes the motor torque to be reduced by the square of the voltage ratio. Naturally, the acceleration time is greater because of the reduced torque produced by the motor. This will not hamper the ability of the motor to accelerate the pump as long as the motor develops more torque than is required to drive the pump at any speed over the accelerating range. The use of SR or PM brushless induction motors eliminates this problem because these motors do not draw high inrush currents. In fact, the starting current is controlled very precisely.

A similar analysis can be made for synchronous motors after the accelerating speed-torque curve for the motor is known. It should be recognized that a synchronous motor operates as a squirrel-cage induction motor up to the moment of synchronization. At that time, the synchronous motor must have an additional capability of synchronizing torque,

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frequently called *pull-in torque,* to accelerate the motor from subsynchronous to synchronous speed. If the motor cannot develop enough torque at its synchronous speed, it will pull out or stall. When this type of motor is used for a pump, it is advisable to unload the pump during acceleration operation to reduce the torque required from the motor for acceleration.

When using a synchronous motor, attention should always be given to the breakaway torque that is required to start the pump from zero speed. This is particularly important with constant displacement pumps, where the pump will operate at a constant torque over the entire accelerating speed range. Again, a better selection might be an inverter-driven ac induction motor, SR or a PM brushless motor.

Service Factor Motors are available with service factor ratings that range from 1.0 to as high as 1.5. A service factor implies that a motor has a built-in thermal capacity to operate at the nameplate power times the service factor stamped on the nameplate. It should be noted, however, that when the motor is operated at the service factor power, the motor will operate at what is termed a *safe temperature.* This means that the motor will operate at a total temperature that is greater than the temperature for a motor designed for the same power with a 1.0 service factor. Consequently, it is not advisable to apply a motor with a service factor larger than 1.0 where the continuous power requirements will be greater than the normal power.

A service factor rating on a motor is to provide an increased power capacity beyond nominal nameplate capacity for occasional overload conditions. Also, the speed-torque characteristics are related to the nominal power rating and not the service factor power. When adjustable speed motor/drive systems are selected, the old service factor ratings are not useful. The supplier of the drive system will assist in selecting a thermally rated system that will satisfy the pumping operation. All inverters are equipped with thermal safety protection for both the motor and the drive so the system cannot be overloaded to minimize overheating. This is a very useful feature for many pump applications.

Efficiency Motors are designed to operate with an efficiency expressed in percent at rated voltage, frequency, and power. Efficiency is defined as

$$\text{Efficiency } \% = \frac{\text{shaft output power} \times 100}{\text{electrical input power}}$$

The efficiency of a given motor design will vary slightly from unit to unit because of manufacturing tolerances and variations in materials. For this reason, guaranteed efficiencies are usually lower than actual efficiencies. When motors are operated at reduced powers, the tendency is for the efficiency to decrease because the losses tend to be fixed.

Several other factors have an effect on motor efficiency. Increasing the applied voltage and operating a motor at its rated power will increase efficiency very slightly, whereas decreasing applied voltage will decrease the efficiency noticeably. Also, increasing the frequency will cause a very slight increase in efficiency, and decreasing the applied frequency will cause a slight decrease in efficiency.

Inverter-fed ac induction and PM or SR brushless motors are not subject to changes in their efficiencies due to changes in line voltage or frequency because the ac line voltage is rectified into a dc voltage before the regeneration of the power to the motor. The dc voltage is always reasonably constant and unaffected by changes in the line power. This of course assumes that there is sufficient voltage and power headroom in the design.

Dynamometers are used to determine efficiency for small motor ratings—up to approximately 500 hp (373 kW)—and standard methods and formulas are used for calculating efficiency for large motors. For the latter, efficiency values can vary, depending upon which "standard" procedure is used. Several suppliers market torque and power analyzers for induction motors that base their output on these formulas using current data measured from the motor. These methods are accurate enough to be acceptable. They have been reasonably verified by applying the method to smaller motors that have been tested on a

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dynamometer. NEMA has established one method, but some suppliers use different methods, depending upon their national standards or established practices.

Higher efficiencies have been designed into most new motors by selecting materials and proportions that reduce losses, but such design choices usually make the motor more costly. This also includes filling the stator slots with more copper of a larger wire gage to reduce ohmic losses. These new motors designed for higher efficiencies offer improvements from 2% to as high as 7% depending upon the size and manufacturer.

Motor operating efficiency in a typical application often may not be that described by the manufacturer. Quoted efficiencies are always at rated power output, rated frequency, and rated voltage. However, it is almost universal in application that a motor is oversized for its applied load and frequently operates at other-than-normal voltage. Both of these variables can greatly reduce the efficiency of a motor. However, when a motor is inverter-driven, the efficiency of the system can be much better controlled.

Power Factor The power factor of a motor is expressed as

$$PF = 100 \cos \theta$$

where θ is the angle between voltage and current at motor terminals (leading or lagging).

The power factor at which a motor operates is dependent on the design of the motor and is established at rated voltage, frequency, and power output.

For induction motors, the power factor can never be 100% leading. A number of factors will influence the power factor of an induction motor:

Condition	Effect on power factor
Increase applied voltage	Decrease
Decrease applied voltage	Increase
Increase load	Increase
Decrease load	Decrease
Increase applied frequency	Slight increase
Decrease applied frequency	Slight increase

In synchronous motors, it is usual to use two varieties of motors, the 100 (unity) and 80% leading motors. The power factor of a synchronous motor operated at rated voltage and frequency is fixed by its field excitation and its power output. At a given power output, the power factor can be adjusted over a range by adjusting the field excitation. Increasing the field excitation will cause the motor to operate at a more leading power factor, and, conversely, reducing the field excitation will make the power factor lag.

Varying the power output of a synchronous motor with a constant field excitation will vary the operating power factor. A decrease in power output will cause a more leading power factor; conversely, increasing the power output will induce operation at a lagging power factor. Consequently, to operate the motor at rated power factor with a varying output power, it is necessary to adjust the field excitation. However, this is not normally done because a synchronous motor is frequently used for improving the power factor and the more leading power factor is used to accommodate power factor improvement. When a synchronous motor is overloaded and operates at a more lagging power factor, it is not usual to increase the excitation beyond its rated excitation because of the extra heating this will develop in the motor. In this case, the more lagging power factor is simply accepted.

The power factor is not of much concern for SR and PM brushless motors driven by inverters because of the control capabilities for these systems. The PM machine exhibits the best power factor of any known motor under nearly all operating speeds and power levels except if excessive field weakening or phase advance is used.

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TYPES OF CONTROLS

In order to use an electric motor to drive a pump, a means of starting and stopping the motor is required. The devices used for this purpose can be called controls. Only small—perhaps up to two or three hp (kW)—motors can be controlled with a simple on-off switch (starting from full rated line voltage). This is because when a motor is first started it produces no back EMF and will draw very high starting currents. If the load inertia is very high, the motor can overheat during the long acceleration time. In addition, the high starting current peaks can cause heavy dips in the line voltage. Therefore, for larger motors, these line voltage dips must be minimized by the use of starters. There are four basic ways to start ac induction motors:

1. Direct online starting
2. Low voltage starting
3. Rotor resistance starting
4. Low frequency starting

Other devices used to control motors such as *soft starters, adjustable frequency controls,* and *vector drives* are discussed along with the inverters and controls for other motor types such as dc motors, SR and PM brushless in Section 9.2.2.

Alternating-Current Motor Starters

MANUAL STARTERS Manual motor starters are designed to provide positive overload protection and start and stop control of single-phase and polyphase motors up to about 3 HP (2.5 kW). A single manual toggle switch or a push button switch is used for "on," "off," and "tripped" states. These types of starters use full line voltage applied to the motor.

MAGNETIC STARTERS Magnetic motor starters cost more and are more reliable than manual starters. They are designed to control a motor by incorporating a magnetically operated contactor to apply power to the motor terminals rather than allowing full current to pass through the contacts.

An overload relay is incorporated to protect the motor from overloading. Magnetic starters are available for reversing and nonreversing service and are also made as non-combination or combination types. The non-combination starter combines only the motor contactor and overload relay. The combination starter combines these parts along with either a circuit breaker or a fused switch to provide shortcircuit protection.

REDUCED-VOLTAGE STARTERS Reduced-voltage starters are available in several types and are basically magnetic starters with additional features to provide reduced voltage, which in turn provides for reduced motor-starting current or torque. These starters include the types discussed in the following paragraphs.

PRIMARY-RESISTOR STARTERS Primary-resistor starters, sometimes known as *cushion-type* starters, will reduce the motor torque and starting inrush current to produce a smooth, cushioned acceleration with closed transition. Although not as efficient as other methods of reduced-voltage starting, primary-resistor starters are ideally suited to applications where reduction of starting torque is of prime consideration. A typical diagram for this type of starter is shown in Figure 20.

AUTOTRANSFORMER STARTERS Autotransformer starters are widely used reduced-voltage starters because of their efficiency and flexibility. All power taken from the line, except

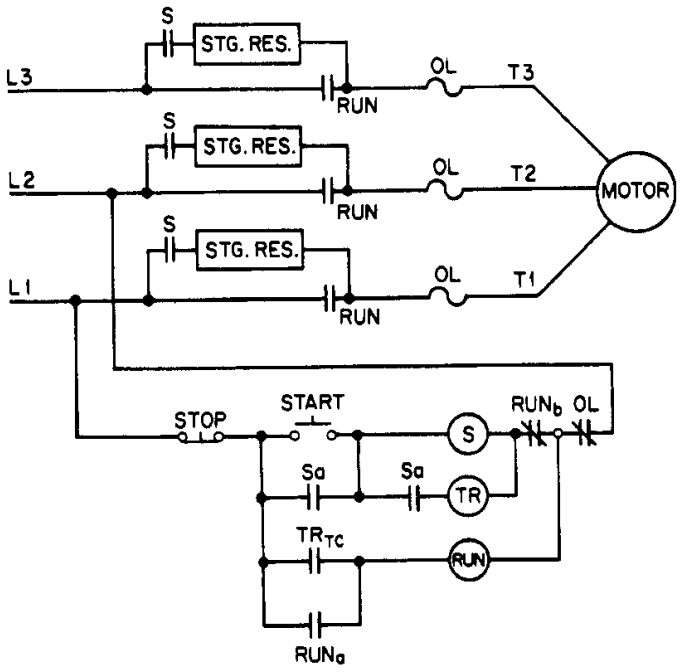


FIGURE 20 Typical primary-resistor, reduced-voltage starter wiring diagram (Westinghouse Electric)

transformer losses, is transmitted to the motor to accelerate the load. Taps on the transformer allow adjustment of the starting torque and inrush to meet the requirements of most applications. The following characteristics are produced by the three voltage taps:

Tap, %	Starting torque, % locked torque	Line inrush, % locked current
50	25	28
65	42	45
80	64	67

A typical diagram for this type of starter is shown in Figure 21.

Part-winding starting provides convenient, economical, one-step acceleration at reduced current where the power company specifies a maximum or limits the increments of current drawn from the line. These starters can be used with standard dual-voltage motors on the lower voltage and with special part-winding motors designed for any voltage. When used with standard dual-voltage motors, it should be established that the torque produced by the first half-winding will accelerate the load sufficiently so as not to produce a second undesirable inrush when the second half-winding is connected to the line. Most motors will produce a starting torque equal to between one-half and two-thirds of NEMA standard values with half the winding energized and draw about two-thirds of normal line-current inrush. A typical diagram is shown in Figure 22.

Star-delta starters have been applied extensively to starting motors driving high-inertia loads with resulting long acceleration times. They are not, however, limited to this application. When 6 or 12 lead delta-connected motors are started star-connected, approximately 58% of full-line voltage is applied to each winding and the motor develops 33% of full-voltage starting torque and draws 33% of normal locked rotor current from the line.

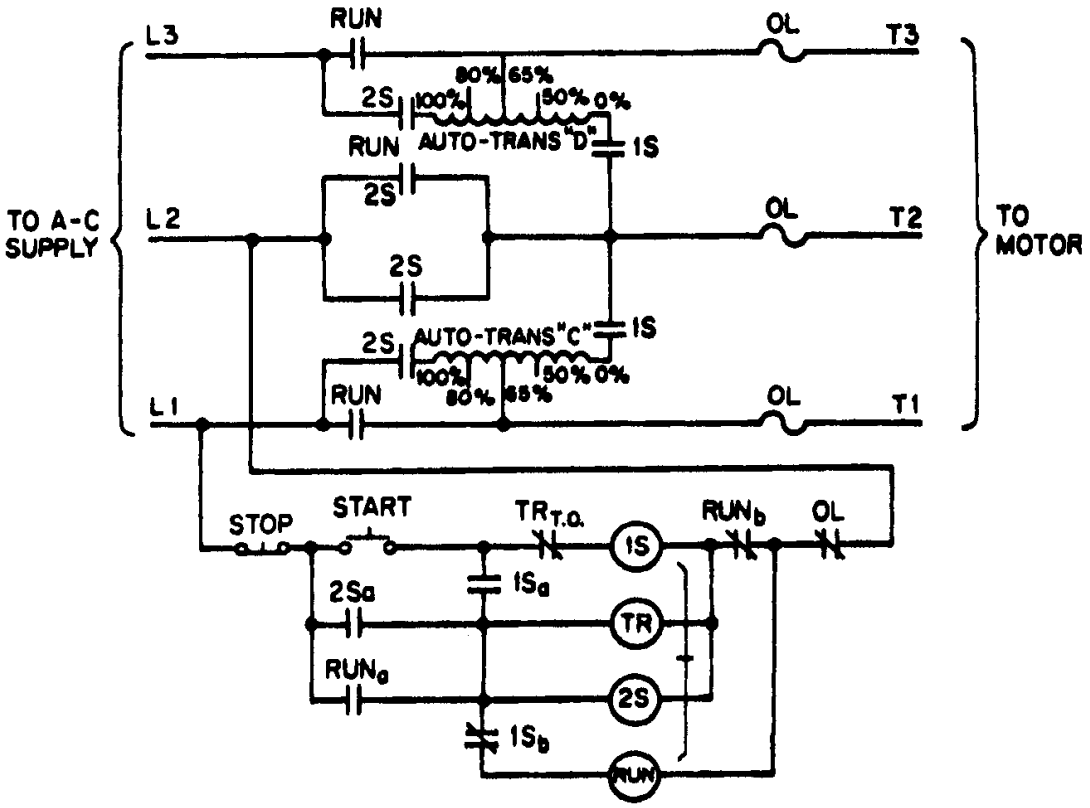


FIGURE 21 Typical autotransformer, reduced voltage starter wiring diagram (Westinghouse Electric)

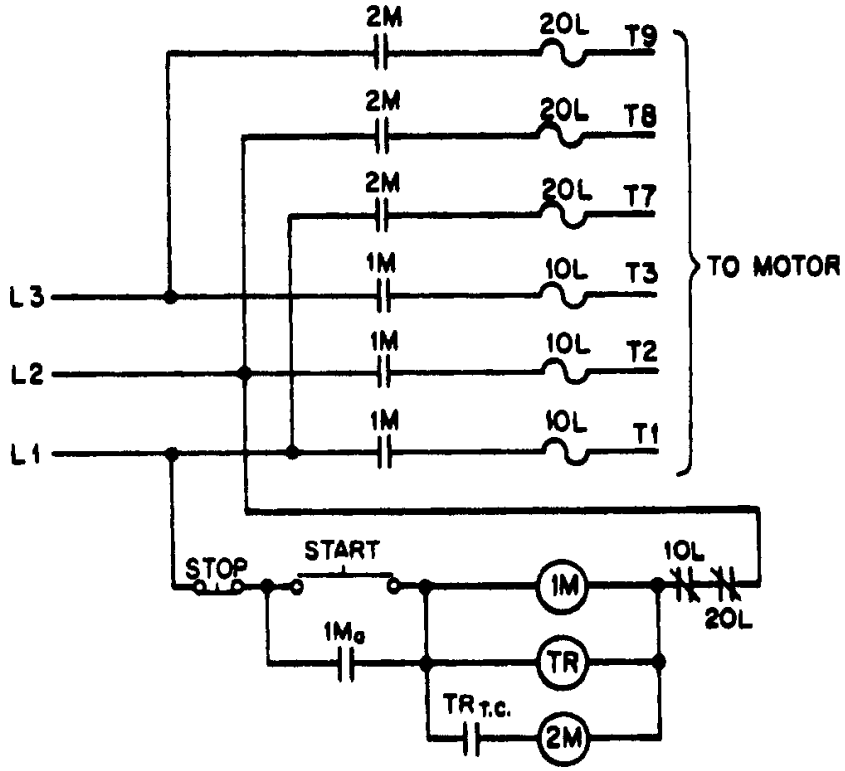


FIGURE 22 Typical part-winding starter wiring diagram (Westinghouse Electric)

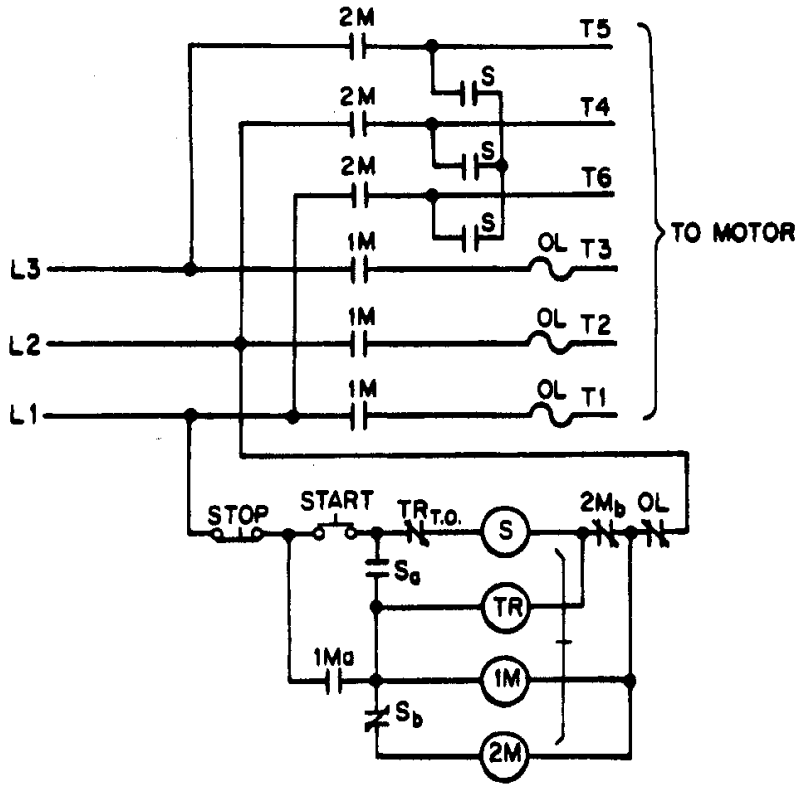


FIGURE 23 Typical star-delta, reduced-voltage starter wiring diagram (Westinghouse Electric)

When the motor has accelerated, it is reconnected for normal delta operation. A typical diagram is shown in Figure 23.

WOUND ROTOR MOTOR STARTERS These magnetic motor starters are used for starting, accelerating, and controlling the speed of wound-rotor motors. The primary control includes overload protection and low-voltage protection or low-voltage release, depending on the type of pilot device. Disconnect switches, circuit breakers, and reversing can be added to the primary circuit when required. Reversing starters are not designed for plugging.

The secondary circuit contains the NEMA recommended number of accelerating or running contactors and resistors to allow approximately 150% of motor full-load torque on first point of acceleration. Additional accelerating points can be added for high-inertia loads or exceptionally smooth starts. Adjustable timing relays permit field adjustment. Standard starting duty NEMA 135 resistors allow 10 s starting out of every 80 s. A typical diagram is shown in Figure 24.

SYNCHRONOUS-MOTOR STARTER Synchronous, magnetic, full-voltage starters provide reliable automatic starting of synchronous motors. They can be used whenever full-voltage starting is permissible. Automatic synchronization is provided by field relay, which assures application of the field at the proper motor speed and at a favorable angular position of stator and rotor poles. As a result, line disturbance resulting from synchronization is reduced and effective motor pull-in torque is increased. A typical diagram is shown in Figure 25.

Brushless synchronous-motor starters require special consideration inasmuch as all brushless synchronous motors are not constructed in the same way. The usual starter incorporates a low-power adjustable dc excitation source to energize and control the output of an integral shaft-connected exciter.

In addition, a pull-out relay and a timing relay are incorporated to initiate synchronization and stop the motor in event of pull-out.

Direct-Current Motor Starters Direct-current motor starters are designed to apply normal voltage to the motor field and, by means of a resistor, reduce voltage to the armature.

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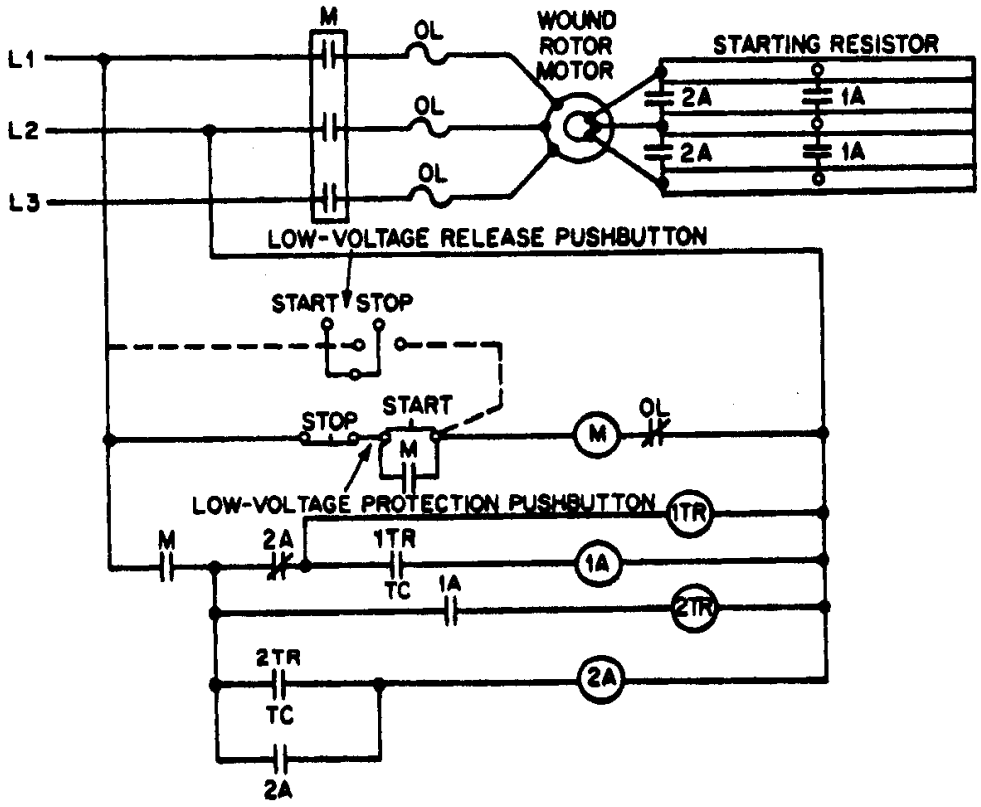


FIGURE 24 Typical wound-rotor motor starter wiring diagram (Westinghouse Electric)

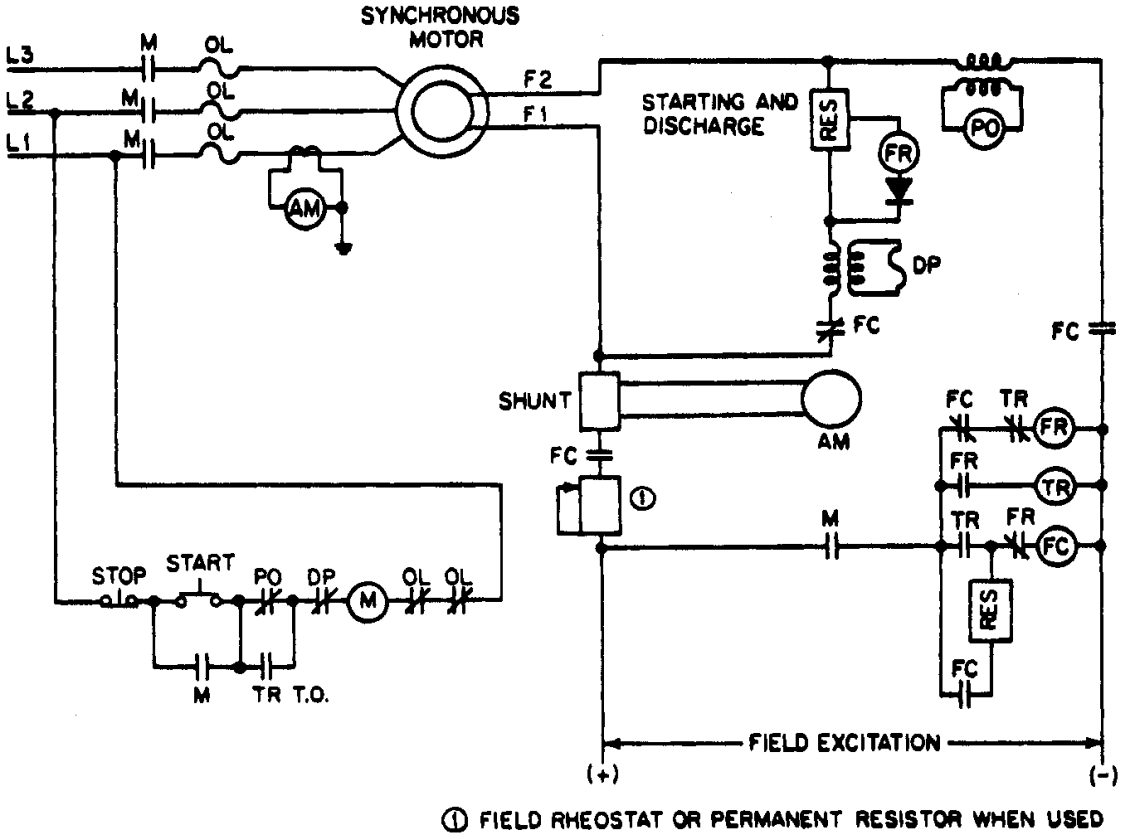


FIGURE 25 Typical synchronous-motor starter wiring diagram (Westinghouse Electric)

Timing relays and contractors progressively short out the resistor until full voltage is on the armature.

If the motor is equipped with a field rheostat for field-range speed adjustment, the starter will apply the preset field voltage as adjusted by the field rheostat after full voltage is applied to the armature.

These starters incorporate a field failure relay to deenergize in the event of field failure and overload relays to protect the motor against overspeed.

The use of dc drives or SCR voltage controllers includes the facilities for all required types of starting conditions for using dc motors for pump applications. Some of these issues are covered in Section 9.2.2.

SEALLESS PUMP MOTORS

Various centrifugal pump designs are available that require no shaft sealing; that is, no packing or mechanical seal. These pumps are completely leakproof, and some are submersible. Such pumps are used when leakage cannot be tolerated or when pumping conditions such as pressure and/or temperature make conventional sealing difficult, if not impossible.

The shaft seal is eliminated by joining the pump and motor housings together to create a single leakproof unit. The impeller and motor rotor are mounted on a single shaft. Most designs permit the pumped liquid to circulate through the motor rotor and motor bearings. However, there are designs where either circulation is also through the stator or the stator is sealed and filled with dialectric oil. See Section 2.2.3.2.

Another popular method used for smaller pumps is to separate the pump shaft completely from the motor shaft. A permanent magnet coupling is provided to transmit the motor torque to the pump shaft through a nonmagnetic shell or section of the pump housing. The pump and motor can be integrated in these designs. See Section 2.2.3.1.

Figure 26 classifies various motor designs and references other places in the handbook where the special features and applications of these units are described in more detail.

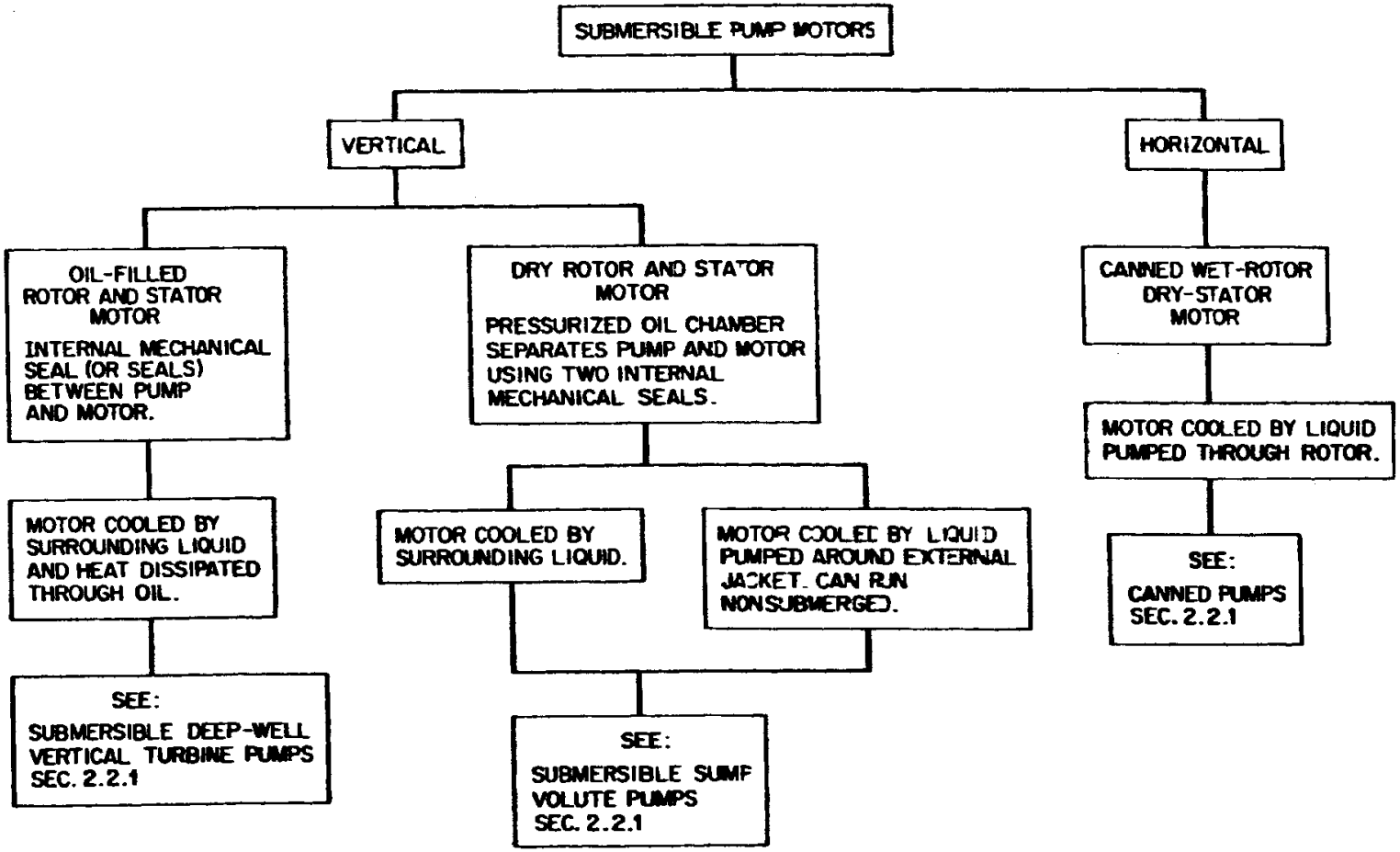


FIGURE 26A Pump motors for sealless centrifugal pumps: Submersible pumps

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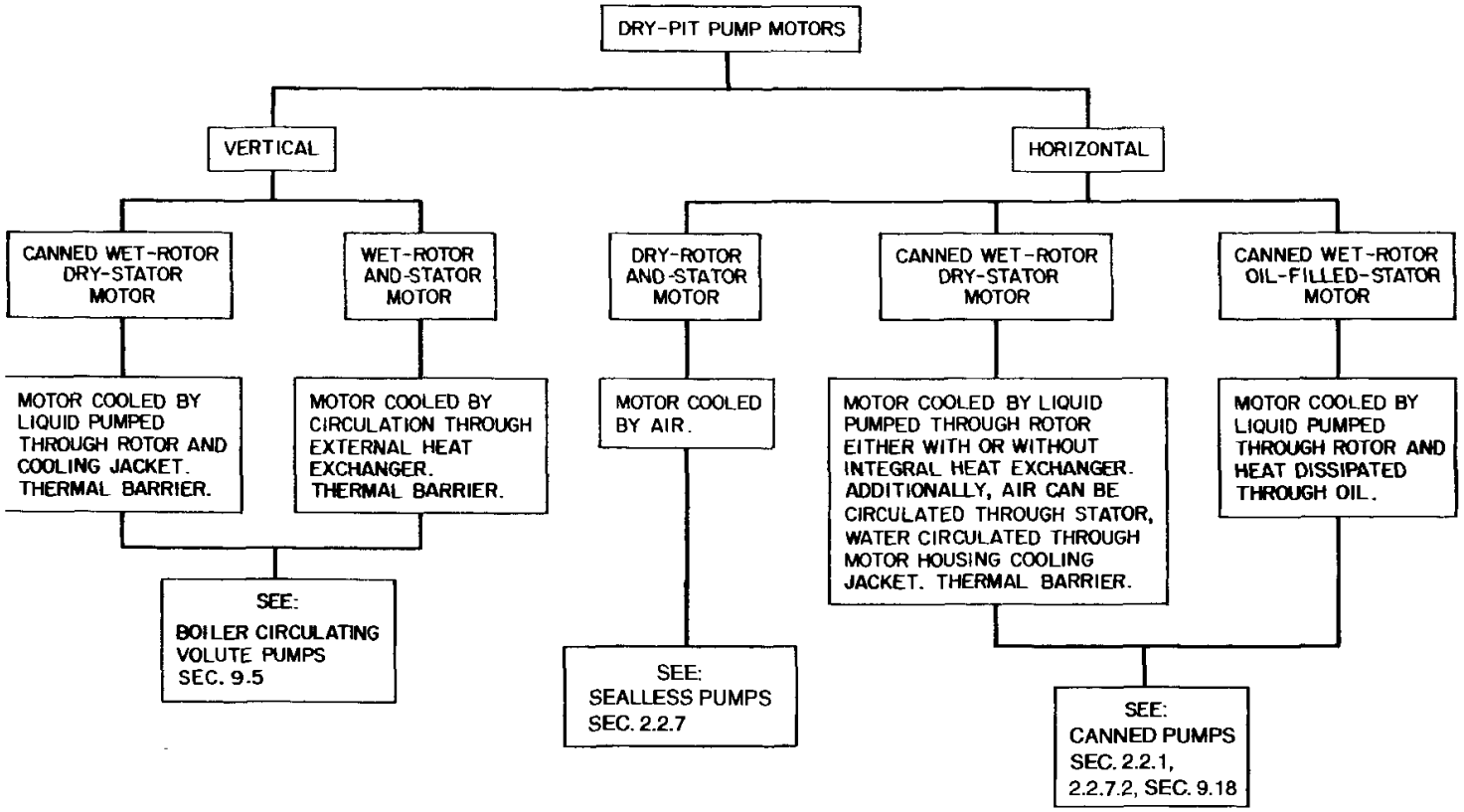


FIGURE 26B Pump motors for sealless centrifugal pumps: Dry pit pumps

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REFERENCES

1. Hendershot, J. R., Jr., *Design of Brushless Permanent-Magnet Motors*, Motorsoft, 1994.
2. Gieras, J. F., Wang, R.-J., and Kamper, M. J. *Axial Flux Permanent Magnet Brushless Machines*, Kluwer Academic Publishers, 2005.
3. Sloteman, D. P., and Piercy, M. "Developing Sealless Integral Motor Pumps Using Axial Field, Permanent Magnet Disk Motors." *Proceedings of the 17th International Pump Users Symposium.* Texas A&M University, College Station, TX, March 2000, pp. 53–67.
4. Hendershot, J. R., Jr., *Switched Reluctance Motors and their Control*, Motorsoft, 1993.
5. T.J.E. Miller, *Brushless Permanent-Magnet and Reluctance Motor Drives*, Clarendon Press–Oxford 1989.
6. S. A. Nasar, I. Boldea, L. E. Unnewehr, *Permanent Magnet, and Self Synchronous Motors*, CRC Press 1993.

9.1.2
STEAM TURBINES

WALLACE L. BERGERON

DEFINITIONS

"A steam turbine may be defined as a form of heat engine in which the energy of the steam is transformed into kinetic energy by means of expansion through nozzles, and the kinetic energy of the resulting jet is in turn converted into force doing work on rings of blading mounted on a rotating part."[1]

This definition may be restated:

"A steam turbine is a prime mover which converts the thermal energy of steam directly into mechanical energy of rotation."[2]

REASONS FOR USING STEAM TURBINES

Steam turbines are used to drive pumps for a variety of reasons:

1. The economical generation of steam often requires boiler steam pressures and temperatures that are considerably in excess of those at which the steam is utilized. Steam may also be used at two or more pressure levels in the same plant. Pressure reduction can be accomplished through valves, pressure-reducing stations, or use of a steam turbine.

   Pressure reduction by using a steam turbine and thereby developing power to drive a pump permits lower utility costs. The incremental increase in steam flow and consequently in fuel costs for the same lower pressure steam heating load is in most instances less than the cost of purchased power for a motor-driven pump.

2. A pump driven by a steam turbine may be operated over a wide speed range, utilizing the turbine governor system or a separately controlled valve in the turbine or in the steam line to the turbine. Operation at variable speeds is an inherent characteristic

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of steam turbines and does not require the use of special speed-changing devices, as is the case with other prime movers.

The overall efficiency of the turbine and pump unit can be optimized by operating at reduced speeds and at the resultant reduced power ratings. Pump performance can be controlled by reducing the speed of the pump rather than throttling it. Although the turbine efficiency normally declines when operating at a reduced speed, the steam flow will still be less than when the pump is throttled.

Operation at reduced power but at constant speed is also permitted by the speed governor, which throttles the steam to the nozzles as the power is reduced. Efficiency may be improved by equipping the turbine with auxiliary steam valves that are closed for reduced power operation. Closing these valves reduces the available nozzle area and reduces the pressure drop across the governor valve.

When the turbine is operated with the auxiliary steam valve closed, the steam flow will approximate that for the same turbine designed for the reduced rating.

3. The use of a steam turbine driver permits the driven pump to operate essentially independent of the electric power or distribution system. The steam turbine is not affected by electric power stoppages or interruptions and is therefore ideal for critical pumping operations.
4. A turbine may be used as a secondary driver for a pump; it may also drive an independent standby or emergency pump. The particular plant design may not afford sufficient steam for the pump to be normally driven by the steam turbine. However, in the event of an electric power failure or power system disturbance, a steam turbine may be employed as a dual drive or to drive a separate pump to assure continued operation of the plant until the electric power system is again operable.
5. The steam turbine controls—governor system and overspeed trip system—are inherently sparkproof. Consequently, steam turbines can be readily applied to drive centrifugal pumps in a wide variety of hazardous atmospheres without entailing additional cost for explosionproof or sparkproof construction.
6. Steam turbines can normally be readily altered to accommodate an increase in rating for increased pump output or for new pump applications. This inherent flexibility of a steam turbine also permits it to be readily altered to accommodate changes in the initial steam pressure and temperature and in the exhaust steam pressure at which the turbine operates.
7. Steam turbines have a starting, or breakaway, torque of approximately 150 to 180% of the rated torque. Additional starting torque can be readily furnished by designing the turbine for the additional required steam flow—and without reducing the efficiency at the normal operating rating by using an auxiliary steam valve. The additional starting torque can often be obtained without increasing the turbine frame size.
8. Steam turbines can be used to drive all types of pumps.
9. Steam turbines are inherently self-limiting with respect to the power developed. Special protective devices do not have to be furnished to prevent damage to the turbine because of overload conditions. The maximum power that can be developed by a turbine is a function of the flow areas provided in the design of the nozzle ring and governor valve. Application of a load greater than that which can be developed by the turbine causes the turbine to slow down to a speed at which the torque generated by the turbine matches that required by the pump.
10. When the pump application requires the driver to be designed with excess power or to permit operation of the pump "at the end of the curve," the steam turbine can be designed for the corresponding rating without reducing the turbine efficiency when operating at the normal rating. Closing an auxiliary steam valve furnished for operation at the normal rating preserves efficiency because the turbine governor valve is not throttling to obtain the power rating.
11. With respect to the operation of the various types of pump drivers and their supporting systems, steam turbines afford minimum maintenance, low vibration, and a quiet installation.

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TYPES OF STEAM TURBINES

Single-Stage Turbines A single-stage steam turbine is one in which the conversion of the kinetic energy to mechanical work occurs with a single expansion of the steam in the turbine—from inlet steam pressure to exhaust steam pressure.

A single-stage turbine may have one or more rows of rotating buckets that absorb the velocity energy of the steam resulting from the single expansion of the steam.

Single-stage turbines are available in wheel diameters of 9 to 28 in (22 to 71 cm). The overall efficiency of a turbine for a particular operating speed and steam conditions is normally dependent on the wheel diameter. The efficiency will generally increase with an increase in wheel size, and therefore the steam rate will be less (for the more usual speeds and steam conditions).

The larger-wheel-diameter steam turbines can be furnished with more nozzles to provide increased steam flow capacity and consequently greater power capabilities. The larger-wheel-diameter turbines are therefore furnished with larger steam connections, valves, shafts, bearings, and so on. Consequently the size of the turbine will generally increase with increases in power rating.

Multiple-Stage Turbines A multiple-stage turbine is one in which the conversion of the energy occurs with two or more expansions of the steam in the turbine. The number of stages (steam expansions) is a function of three basic parameters: thermodynamics, mechanical design, and cost. The thermodynamic considerations include the available energy and speed. The mechanical considerations include speed, steam pressure, steam temperature, and so on, most of which are material limits. Cost considerations include the number, type, and size of the stages; the number of governor-controlled valves; the cost of steam; and the number of years used as a basis for the cost evaluation.

The two factors generally used in selecting multistage turbines are initial cost and steam rate. Because these two factors are a function of the total number of stages, the application becomes a factor of stage selection. The initial cost increases with the number of stages, but the steam rate generally improves.

Multiple-stage turbines are normally used to drive pumps when the cost of steam or the available supply of steam requires turbine efficiencies greater than these available with a single-stage turbine, or when the steam flow required to develop the desired rating exceeds the capability of single-stage turbines.

Multiple-stage turbines can be furnished with a single or multiple governor valves. A single governor valve is often of the same design whether used in a single-stage or multiple-stage turbine and generally has the same maximum steam flow, pressure, and temperature parameters. Multiple valves are used when the parameters for a single valve are exceeded or to obtain improved efficiency, particularly at reduced power outputs.

Shaft Orientation Some steam turbines, particularly single-stage turbines, can be furnished with vertical downward shaft extensions. The application of such turbines can require considerable coordination between the pump and the turbine manufacturer to assure an adequate thrust bearing in the turbine, shaft length and details, mounting flange dimensions, and even shaft runout.

Vertical shaft pumps are frequently driven by horizontal turbines through a right-angle speed-reduction gear unit.

Direct-Connected and Geared Turbines Steam turbines can be directly connected to the pump shaft so the turbine operates at the pump speed or can drive the pump through a speed-reduction (and even speed-increase) gear unit, in order to permit the turbine to operate at a more efficient speed.

Turbine Stages The two types of turbine stages are impulse and reaction. The turbines discussed in this section employ impulse stages because steam turbines driving pumps normally have impulse-type stages.

In the ideal impulse stage, the steam expands only in the fixed nozzles and the kinetic energy is transferred to the rotating buckets as the steam impinges on the buckets while

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flowing through the passages between them. The steam pressure is constant, and the steam velocity relative to the bucket decreases in the bucket passages.

In a reaction stage, the steam expands in both the fixed nozzles and the rotating buckets. The kinetic energy is transferred to the rotating buckets by the expansion of the steam in the passages between the buckets. The steam pressure decreases as the steam velocity relative to the buckets increases in the bucket passages.

In an impulse stage, the steam can exert an axial force on the buckets as it flows through the blade passages. Although this force is usually referred to as a reaction, the use of the term does not imply a reaction-type stage.

The larger buckets used in the last stages of an impulse-type multistage turbine can be of a free-vortex design—twisted and tapered. Such a bucket is ideally subjected to a nearly pure impulse force at its root and a nearly pure reaction force at its tip, but, in reality, this bucket is a high reaction-design bucket compared with a normal impulse-stage bucket. A steam turbine stage with such a bucket design is still referred to as an impulse stage because the primary conversion of kinetic energy is by a reduction rather than an increase in relative steam velocity.

A reaction turbine has more stages than an impulse turbine for the same application because of the small amount of kinetic energy absorbed per stage, and requires a larger thrust bearing or a balancing piston because of the pressure drop across the moving blades. The small pressure drop per stage and the pressure drop across the moving blades require that the steam-leakage losses be minimized by elaborate sealing between the tips of the nozzle blades and the rotor, and the tips of the moving blades and the casing..

The small pressure differential across the rotating blades of an impulse stage results in smaller thrust bearings and no close blade-tip clearances. Consequently, impulse turbines can be started more quickly without thermal-expansion damage, and their stage efficiencies remain relatively constant over the life of the turbine.

CONSTRUCTION DETAILS

Component Parts The main components of a single-stage steam turbine are shown in Figure 1.

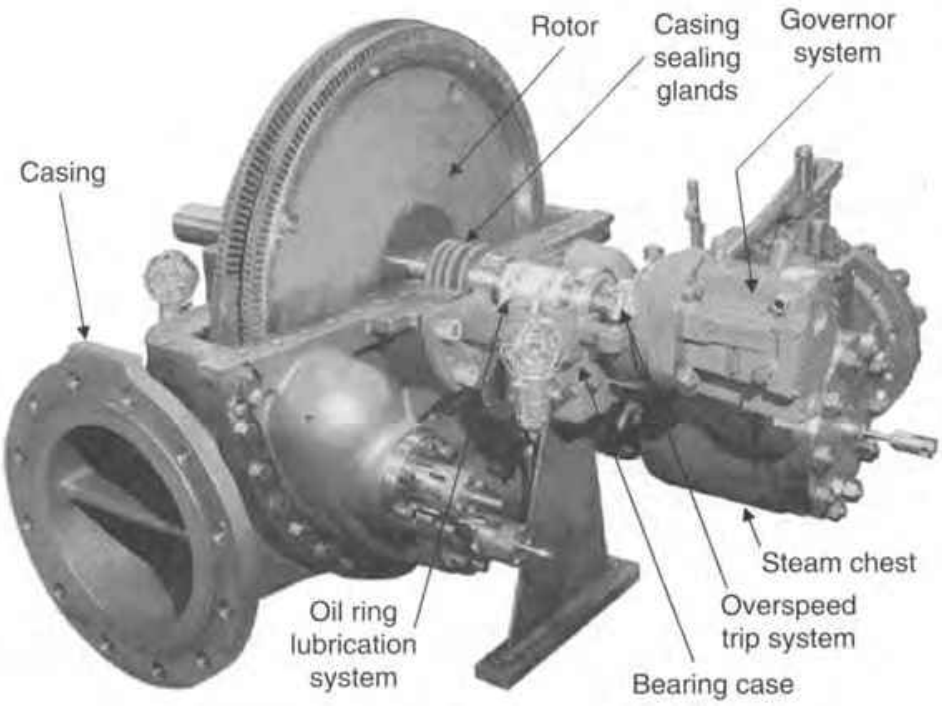


FIGURE 1 Component parts of steam turbines (Elliott)

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Function and Operation—Single-Stage Turbine The *steam* chest and the *casing* contain the steam furnished to the turbine, being connected to the higher-pressure steam supply line and the lower-pressure steam exhaust line, respectively. The steam chest, which is connected to the casing, houses the governor valve and the overspeed trip valve. The casing contains the rotor and the nozzles through which the steam is expanded and directed against the rotating buckets.

The *rotor* consists of the shaft and disk assemblies with buckets. The shaft extends beyond the casing and through the bearing cases. One end of the shaft is used for coupling to the driven pump. The other end serves the speed governor and the overspeed trip systems.

The *bearing cases* support the rotor and the assembled casing and steam chest. The bearing cases contain the journal bearings and the rotating oil seals, which prevent outward oil leakage and the entrance of water, dust, and steam. The steam end bearing case also contains the rotor positioning bearing and the rotating components of the overspeed trip system. An extension of the steam end bearing housing encloses the rotating components of the speed governor system.

The *casing sealing glands* seal the casing and the shaft with spring-backed segmented carbon rings (supplemented by a spring-backed labyrinth section for the higher exhaust steam pressures).

The *governor system* commonly consists of spring-opposed rotating weights, a steam valve, and an interconnecting linkage or servomotor system. Changes in the turbine inlet and exhaust steam conditions, and the power required by the pump will cause the turbine speed to change. The change in speed results in a repositioning of the rotating governor weights and subsequently of the governor valve.

The *overspeed trip system* usually consists of a spring-loaded pin or weight mounted in the turbine shaft or on a collar, a quick-closing valve that is separate from the governor valve, and interconnecting linkage. The centrifugal force created by rotation of the pin in the turbine shaft exceeds the spring loading at a preset speed. The resultant movement of the trip pin causes knife edges in the linkage to separate and permit the spring-loaded trip valve to close.

The trip valve may be closed by disengaging the knife edges manually, by an electric or pneumatic signal, by low oil pressure, or by high turbine exhaust steam pressure.

The two usual types of lubrication systems are oil-ring and pressure. The *oil-ring* lubrication system employs an oil ring(s) that rotates on the shaft with the lower portion submerged in the oil contained in the bearing case. The rotating ring(s) transfers oil from the oil reservoir to the turbine shaft journal bearing and rotor-locating bearing. The oil in the bearing case reservoirs is cooled by water flowing in cooling water chambers or tubular heat exchangers.

A *pressure* lubrication system consists of an oil pump driven from the turbine shaft, an oil reservoir, a tubular oil cooler, an oil filter, and interconnecting piping. Oil is supplied to the bearing cases under pressure. The oil rings may be retained in this system to provide oil to the bearings during startup and shutdown when the operating speed and bearing design permit.

Typical sectional drawings are shown in Figures 2, 3, and 4.

GOVERNORS AND CONTROLS

Governor systems are typically speed-sensitive control systems. The turbine speed is controlled by varying the steam flow through the turbine by positioning the governor valve. Variations in the power required by the pump and changes in steam inlet or exhaust conditions alter the speed of the turbine, causing the governor system to respond to correct the operating speed.

Control systems, unlike governor systems, are not directly speed-sensitive but respond to changes in pump or pump-system pressures and then reposition the turbine "governor" valve to maintain the preset pressure. Consequently, changes in turbine steam conditions or in the power required by the pump result in a repositioning of the turbine governor valve or of a separate steam valve only after the pressure being sensed by the controller has changed.

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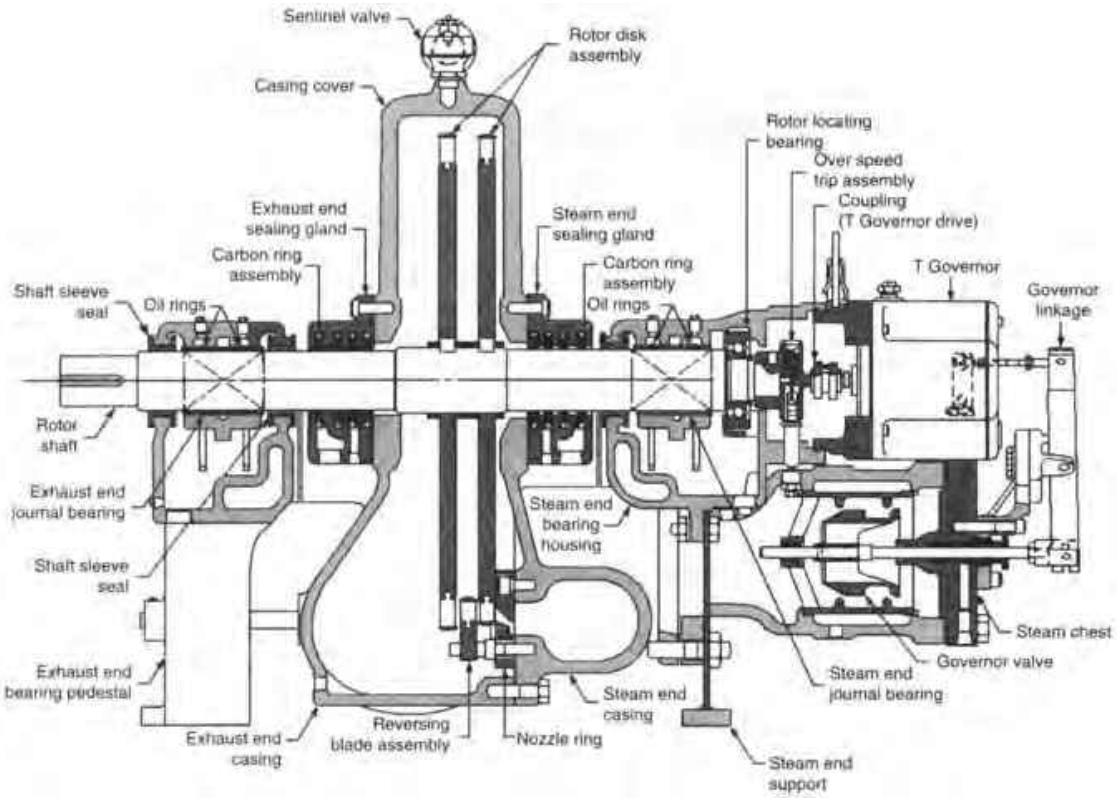


FIGURE 2 Section of single-stage turbine and governor system (Elliott)

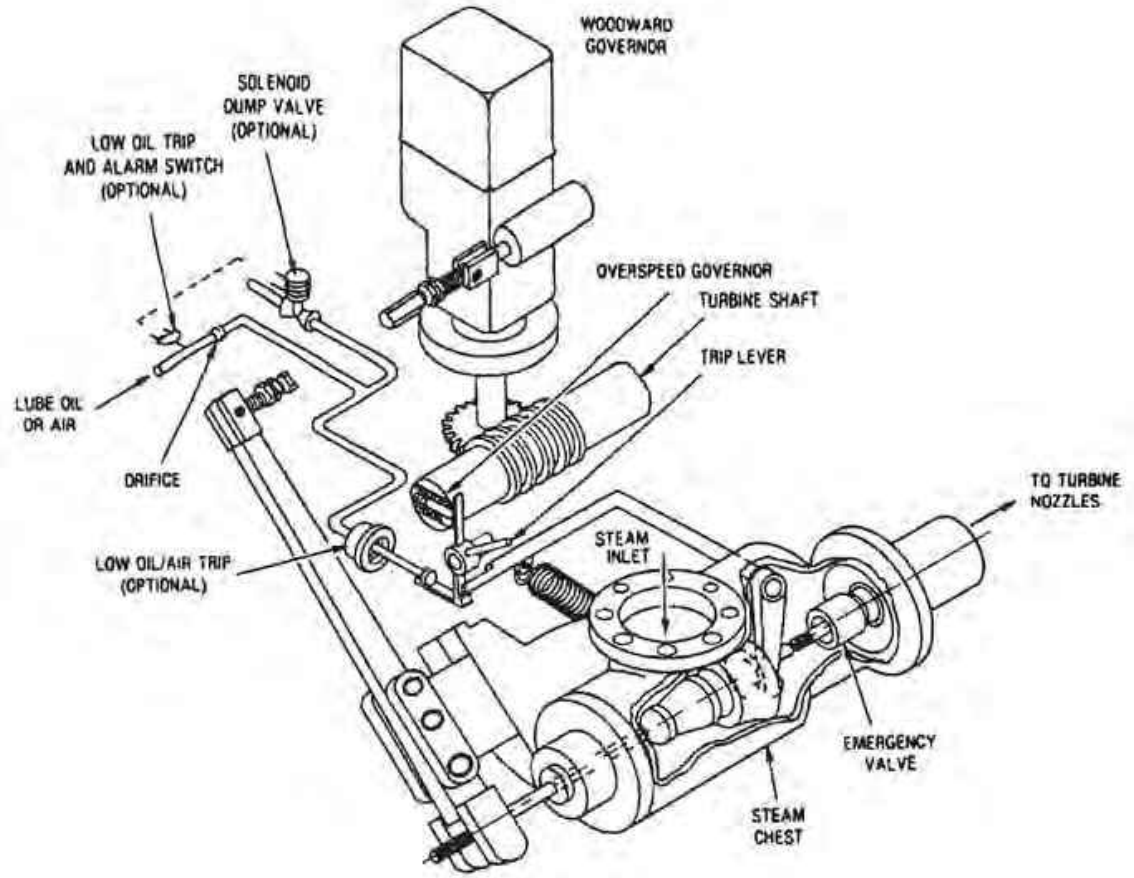


FIGURE 3 Typical turbine overspeed trip arrangement (Dresser-Rand)

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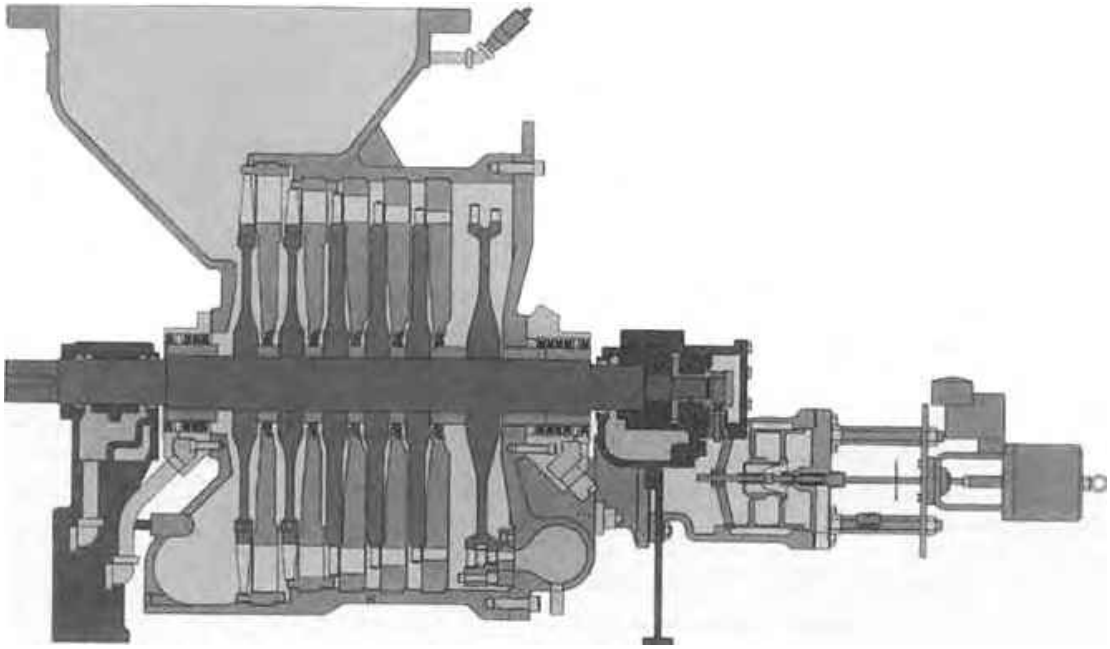


FIGURE 4 Section of multistage turbine (Elliot)

Even when a control system is furnished, a speed governor is also normally furnished. The speed governor is set for a speed slightly higher than the desired operating speed in order to function as a pre-emergency governor; that is, prevent the turbine from reaching the trip speed when the controller causes the turbine to operate at a speed above rated speed.

Governor systems are defined by their performance as follows[2]:

Class of governor system	Speed range, % (as specified)	Maximum speed regulation, %	Maximum speed variation, %, ±	Maximum speed rise, %
A	10–65	10	0.75	13
B	10–80	6	0.50	7
C	10–80	4	0.25	7
D	10–90	0.50	0.25	7

Speed range is the percentage below rated speed for which the governor speed setting may be adjusted. For example, a turbine with 4000 rpm rated speed and a governor system having a 30% range can be operated at a minimum speed of 2800 rpm:

$$4000 - \frac{30 \times 4000}{100} = 2800$$

If the speed range had been specified as plus 5% and minus 25%, for example, the maximum and minimum speeds would be 4200 and 3000 rpm.

Steady-state speed regulation is the change in speed required for the governor system to close the governor valve when the load is gradually reduced from rated load to no load with turbine steam conditions constant. Regulation is always expressed as a percentage of rated speed and calculated as follows:

$$\frac{\text{No-load speed} - \text{rated speed}}{\text{Rated speed}} \times 100 = \%\ \textit{regulation}$$

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Therefore

$$\text{No-load speed} = \text{rated speed}\left(1 + \frac{\%\ \text{regulation}}{100}\right)$$

A turbine with 4000 rpm rated speed and equipped with a NEMA A speed governor system would have 4400 rpm maximum no-load speed:

$$4000 \times \left(1 + \frac{10}{100}\right) = 4400 \text{ rpm}$$

Consequently, whenever the turbine is developing less than rated power, the operating speed will be greater than rated speed, as required for the governor system to reposition the governor valve.

Speed variation, expressed as a percentage, is the total magnitude of the fluctuations from the set speed permitted by the governor system when the turbine is normally operating at rated speed, power, and steam conditions. This is the "insensitivity" of the governor system. The speed variation equation is

$$\frac{\text{Maximum speed} - \text{minimum speed}}{\text{Rated speed} \times 2} \times 100 = \pm\%\ \textit{speed variation}$$

A turbine with 4000 rpm rated speed and NEMA A governor system would have ±30 rpm maximum speed variation:

$$4000 \times \frac{\pm 0.75}{100} = \pm 30 \text{ rpm}$$

Maximum speed rise represents the momentary increase in speed when the load is suddenly reduced from rated power to no-load power with the pump still coupled to the turbine shaft. Shortly after the sudden loss of load, the governor system will cause the turbine speed to be reduced to the no-load speed.

Maximum speed rise is also expressed as a percentage of rated speed and calculated as follows:

$$\frac{\text{Maximum speed} - \text{rated speed}}{\text{Rated speed}} \times 100 = \%\ \text{speed rise}$$

Therefore

$$\text{Maximum speed} = \text{rated speed} + \frac{\text{rated speed} \times \%\ \text{speed rise}}{100}$$

A turbine with 4000 rpm rated speed and equipped with a NEMA A governor will have a maximum speed rise to

$$4000 + \frac{4000 \times 13}{100} = 4520 \text{ rpm}$$

The setting of the *overspeed trip* is a function of the maximum speed rise of the governor system—it must be higher than the maximum speed rise. The recommended settings are

Class of governor system	Overspeed trip setting, (% of rated speed)
A	115
B	110
C	110
D	110

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The overspeed trip system for a turbine with 4000 rpm rated speed and equipped with a NEMA A governor is set for operation at

$$4000 \times \frac{115}{100} = 4600 \text{ rpm}$$

The speed-sensitive portion of the speed governor system is usually a set of spring-loaded rotating weights. Movement of the weights caused by a change in turbine speed positions the governor valve through a suitable linkage.

The speed-sensitive element can also be other devices that are speed-responsive, such as a positive displacement oil pump, electric generator, or a magnetic impulse signal generator.

The rotating-weight governor system is a direct-acting type and is classified as a NEMA A governor. *Direct-acting* designates a governor system in which the speed-sensitive element also provides the power for positioning the governor valve.

The NEMA B, C, and D governor systems have speed-sensitive elements that position the governor valve through a relay or servomotor system instead of actuating the valve directly. The speed-sensitive element can therefore be more precise and sensitive, as required for the improved governor system performance.

Electronic governors with pneumatic/hydraulic actuators are also available that offer more control, flexibility, and speed adjustment as well as other operating modes.

THEORY

A steam turbine develops mechanical work by converting to work the available heat energy in the steam expansion. Heat and mechanical work, being two forms of energy, can be converted from one to the other.

The heat energy is converted in two steps. The steam expands in nozzles and discharges at a high velocity, converting the available heat energy to velocity (kinetic) energy. The high-velocity steam strikes moving blades, converting the velocity energy to work. Because the total heat energy available in the steam is converted to velocity (kinetic) energy, the magnitude of the steam velocity is dependent upon the available energy.

The mechanical work that is developed in the turbine by the high-velocity steam striking the buckets is a function of the speed of the buckets. Maximum work occurs when the bucket velocity is approximately one-half the steam jet velocity for an impulse stage and one-fourth the steam jet velocity for a velocity-compounded impulse stage. Although the steam jet velocity is fixed by the available heat energy, the bucket velocity is fixed by the speed of the turbine and the diameter of the turbine wheel on which the buckets are mounted. The work developed, or the efficiency of the turbine, ignoring losses in the turbine, is therefore determined by the size of the turbine and the turbine (pump) speed for a fixed amount of available heat energy.

The most common single-stage turbine is the velocity-compounded (Curtis) type. The complete expansion from inlet to exhaust pressure occurs in one step. The Curtis stage, with two rows of rotating buckets, and two re-entry-type velocity-compounded stages are illustrated in Figure 5.

Single-stage turbines are available with a wide range of efficiencies because they are manufactured with a variety of wheel diameters: 9 to 28 in (22 to 71 cm).

Multistage turbines are manufactured with a more limited variety of wheel sizes. The efficiency of multistage turbines is varied primarily by varying the number of stages. When the total available energy of the steam results in a steam velocity greater than twice the bucket velocity (using convenient wheel sizes), a multistage turbine will be more efficient. In a multistage turbine, the total steam expansion is divided among the various impulse stages to produce the desired steam velocity for each row of buckets.

A steam turbine is normally evaluated using *steam rate*—the amount of steam required by the turbine to produce the specified power per hour at the specified speed—rather than *efficiency*. The steam rate is a direct function of the turbine efficiency.

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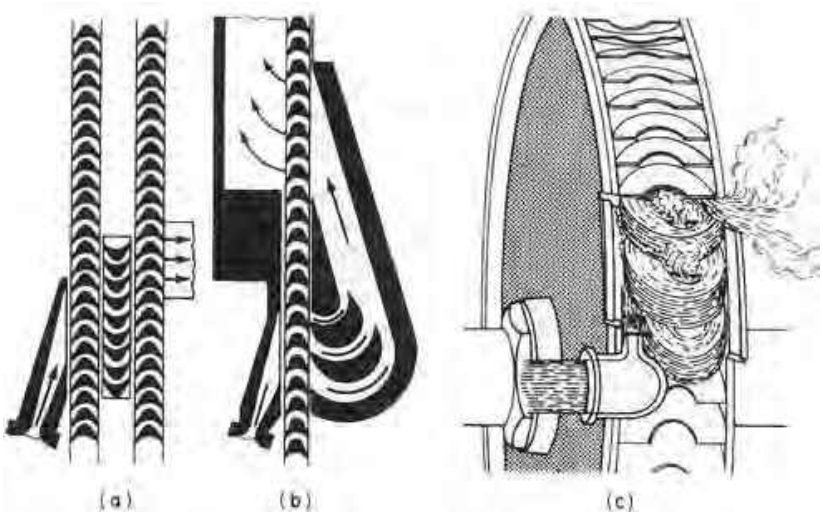


FIGURE 5 Velocity-compounded stages: (a) Curtis stage; steam flows once through moving buckets; (b) re-entry stage; steam flows twice through moving buckets; (c) re-entry stage; steam flows three times through moving buckets (reprinted with permission from *Power*, June 1962)

The steam consumption can be expressed either as steam rate, pounds of steam per horsepower-hour (kilograms per kilowatt-hour) or as steam flow, pounds of steam per hour (kilograms per hour). The higher the efficiency, the lower the steam rate or steam flow, and vice versa.

The total available energy of the steam is that available from an isentropic expansion. For given initial steam pressure and temperature and exhaust pressure, the available energy in British thermal units per pound (kilojoules per kilogram) of steam can be obtained from the tables or the Mollier chart in Reference 3.

The available energy can be converted to power units and expressed as the theoretical steam rate—pounds per horsepower-hour or pounds per kilowatt-hour (kilograms per kilowatt-hour). The theoretical steam rate is the steam rate for a 100% efficient turbine and therefore can be used more conveniently than energy in British thermal units per pound (kilojoules per kilogram) for the calculation of turbine steam rates. Theoretical steam rates can be obtained directly from theoretical steam rate tables (ASME) or from the polar Mollier chart (Elliott Company).

The actual steam rate for a turbine is greater than the theoretical steam rate because of the losses that occur in the turbine when the available energy is converted to mechanical work and because of the ratio of the steam velocity to bucket velocity. The energy remaining in the steam exhausting from the turbine is greater than that after an isentropic expansion, as illustrated in Mollier diagram shown in Figure 6, where

1 = energy in steam at initial steam pressure and temperature

2 = energy in steam at exhaust pressure for an isentropic expansion

3 = actual energy in steam at exhaust pressure

The efficiency of the turbine is

$$\frac{h_1 - h_3}{h_1 - h_2} \quad \text{or} \quad \frac{\text{theoretical steam rate}}{\text{actual steam rate}}$$

The governor and trip valve pressure drop losses are a function of the sizes of these two valves and the steam flow. The governor valve pressure drop will vary more than the trip valve pressure drop because of changes in valve position with power required by the driven pump, speed variations, and changes in inlet and exhaust steam conditions.

The nozzle loss is due to friction in the nozzles as the steam expands. The efficiency of the nozzles is a function of the ratio of the actual and ideal exit steam velocities squared. The efficiency is usually between 95 and 99%.

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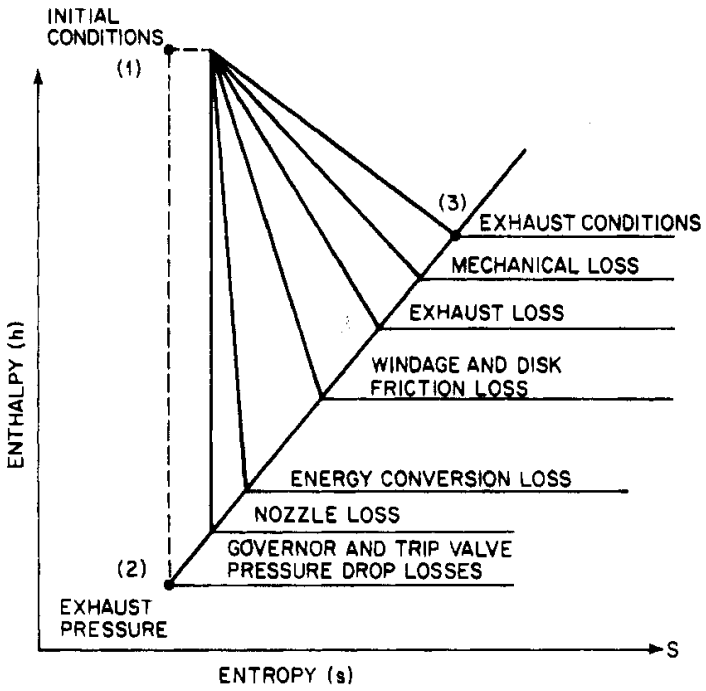


FIGURE 6 Mollier diagram with energy losses

The windage and disk friction losses are due to the friction between the stream and the disks and the blades fanning the steam. This loss varies inversely with the specific volume of the steam, increases with exhaust pressure, and increases with the diameter of the wheel and the length of the blades.

The use of a larger-diameter wheel may increase the efficiency, but the windage and disk friction losses will reduce the improvement and may even cause a net loss in overall efficiency.

The exhaust losses represent the kinetic energy remaining in the steam as a result of the velocity of the steam leaving the bucket and the pressure drop in the steam as it passes out the exhaust connection.

The energy conversion loss is due to the nonideal conversion of the steam velocity energy to mechanical work in the buckets as a function of the steam velocity and bucket velocity, plus nonideal nozzle and bucket angles, friction in the system, and so on.

The performance that can be expected from a single-stage Curtis-type turbine may be obtained from Figures 7, 8, and 9, and Table 1 after determining the theoretical steam rate.

$$\text{Steam rate} = \frac{\text{base steam rate}}{\text{superheat correction factor}} \times \frac{\text{power + power loss}}{\text{power}}$$

To obtain superheat, subtract temperature given in Table 1 from total initial temperature.

SAMPLE CALCULATION

STEAM CONDITIONS 250 lb/in^2 (1724 kPa) gage inlet, 575°F (302°C) inlet, and 50 lb/in^2 (345 kPa) gage exhaust.

DESIGN CONDITIONS turbine to develop 500 hp (373 kW) at 4000 rpm.

1. Theoretical steam rate = 26.07 lb/kWh (11.82 kg/kWh)
2. Base steam rate for 28-in (71.12-cm) wheel (Figure 7) = 36 lb/hp · h (22 kg/kWh)

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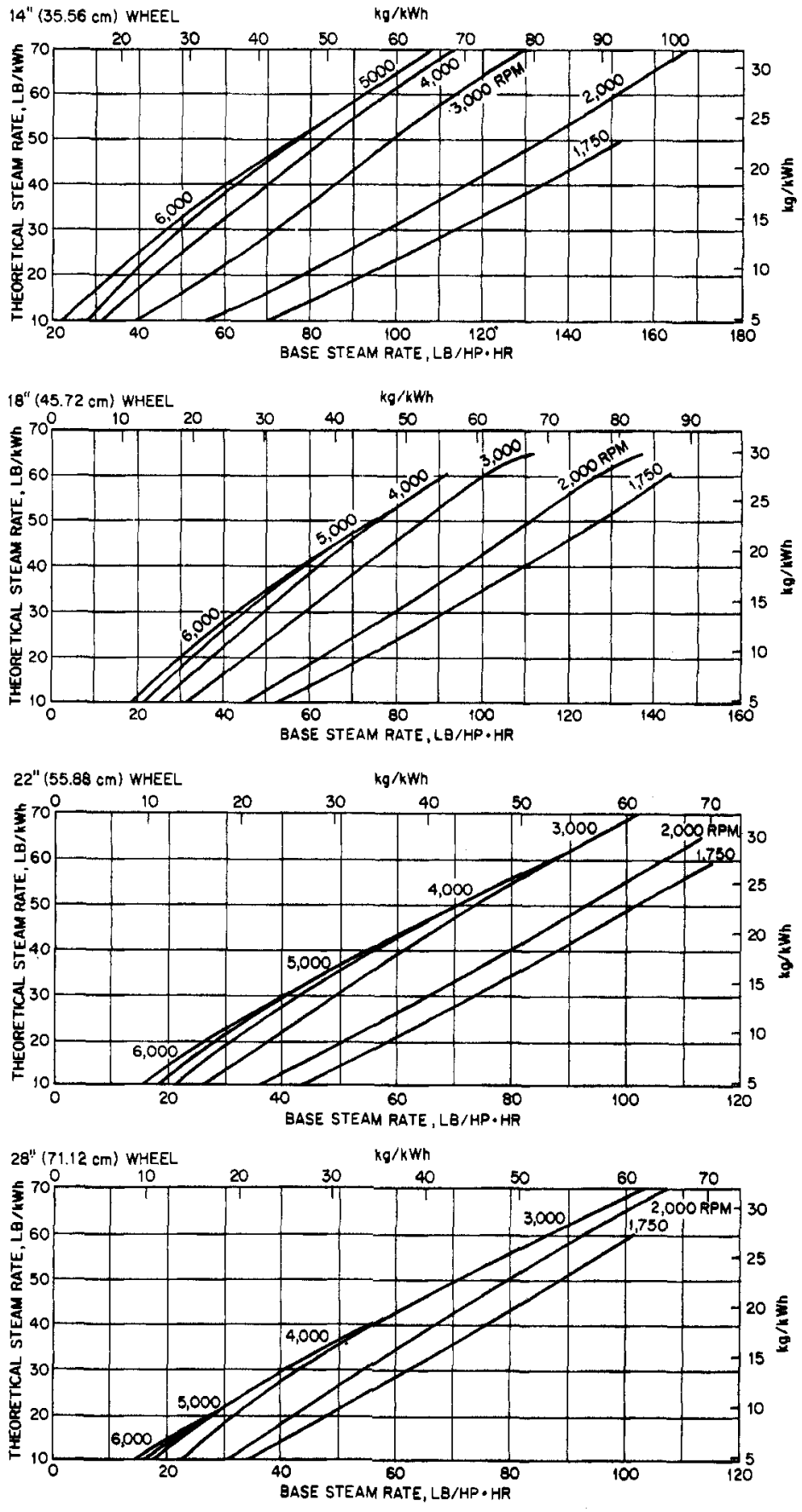


FIGURE 7 Base steam rates (Elliott)

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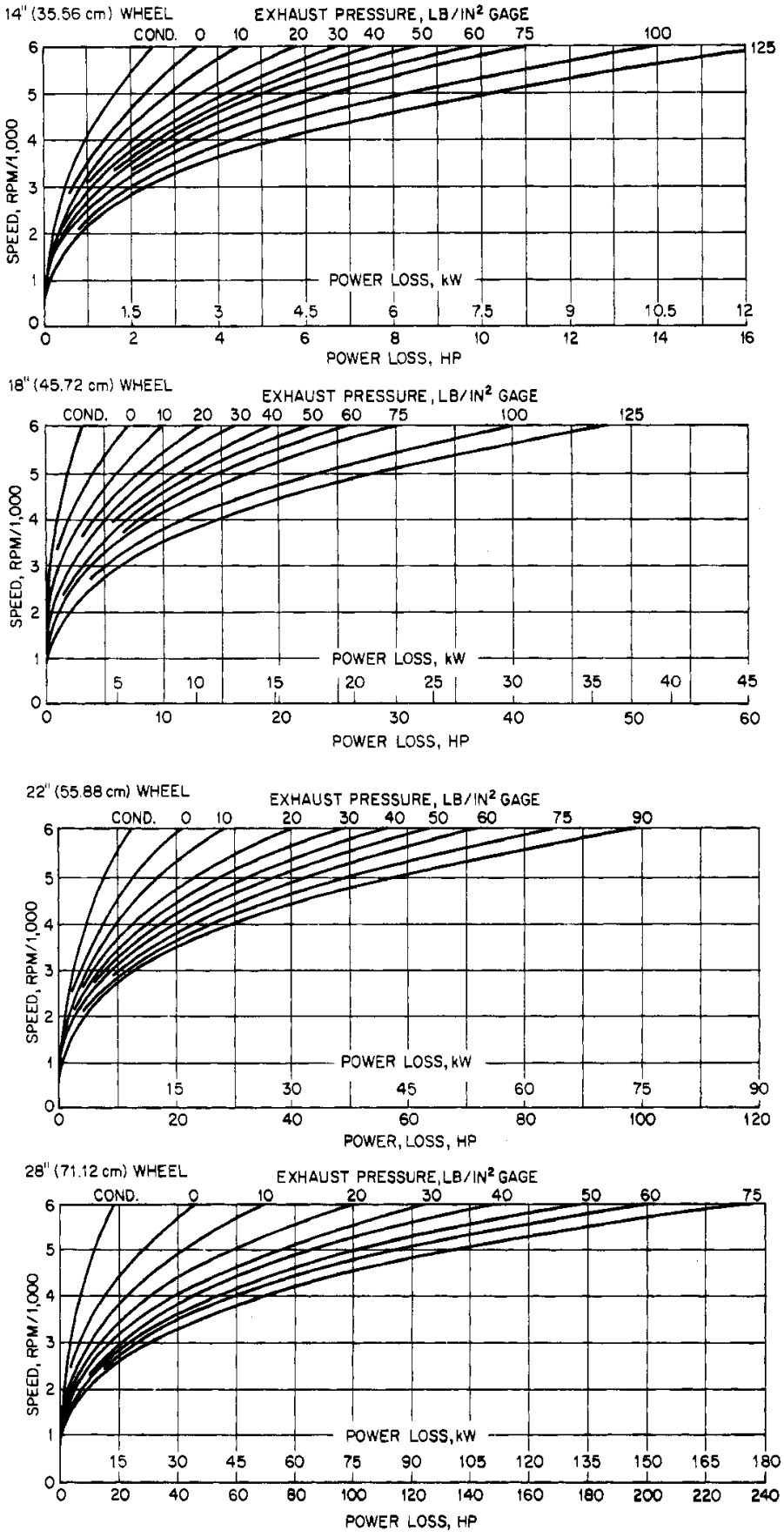


FIGURE 8 Power loss (lb/in² × 6.895 = kPa) (Elliott)

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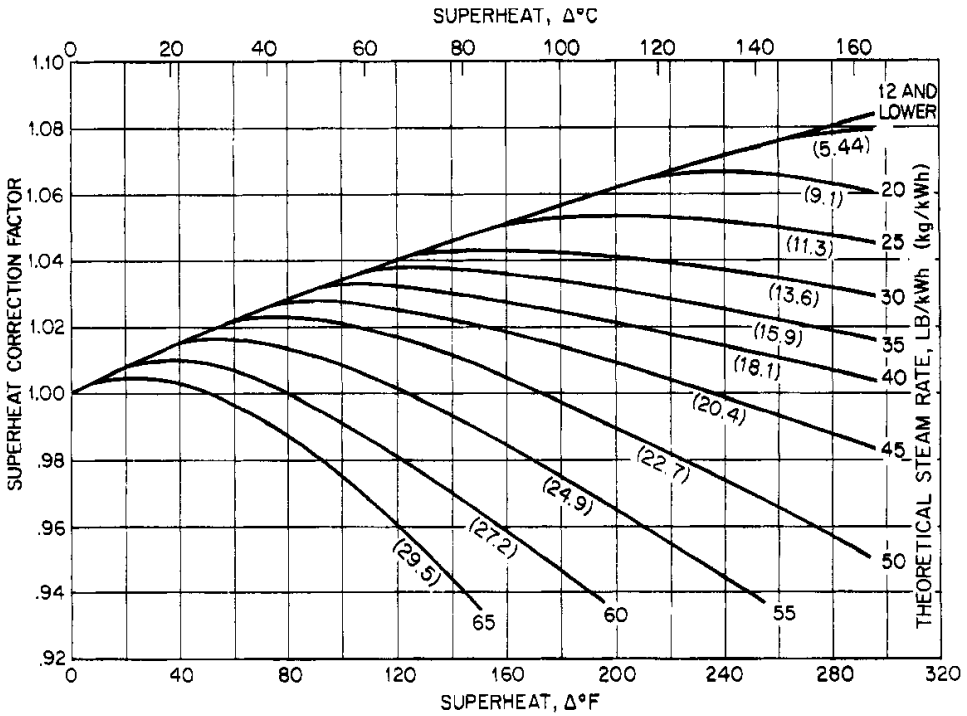


FIGURE 9 Superheat correction factor (Elliott)

3. Power loss for 28-in (71.12-cm) wheel (Figure 8) = 55 hp (41 kW)
4. Temperature of dry and saturated inlet steam (Table 1) = 406°F (208°C)
5. Superheat (Table 1) = 575 – 406 = 169Δ°F (302 – 208 = 94Δ°C)
6. Superheat corrections factor (Figure 9) = 1052
7. Steam rate =

$$\text{In USCS units} \qquad \frac{36}{1.052} \times \frac{500 + 55}{500} = 38.0 \text{ lb/hp·h}$$

$$\text{In SI units} \qquad \frac{21.89}{1.052} \times \frac{373 + 41}{373} = 23.1 \text{ kg/kWh}$$

The ability of a particular size turbine to develop the required power is determined primarily by

1. The flow capacity of the inlet and exhaust connection from Figures 10 and 11, where steam flow = power × steam rate
2. The flow capacity of the nozzles available in a particular turbine (the number and size of the nozzles vary considerably with each design of turbine manufactured, and thus a meaningful plot cannot be included here).

EVALUATION OF COSTS

The economical selection of a turbine considers the initial cost of the turbine and the operating costs. A lower-steam-rate (more efficient) turbine generally has a higher initial cost

TABLE 1 Temperature of dry and saturated steam

Lb/in² gage	Saturation temp., °F	Lb/in² gage	Saturation temp., °F	Lb/in² gage	Saturation temp., °F	Lb/in² gage	Saturation temp., °F
0	213	150	366	300	422	450	460
5	228	155	368	305	423	455	461
10	240	160	371	310	425	460	462
15	250	165	373	315	426	465	463
20	259	170	375	320	428	470	464
25	267	175	378	325	429	475	465
30	274	180	380	330	431	480	466
35	281	185	382	335	432	485	467
40	287	190	384	340	433	490	468
45	293	195	386	345	434	495	469
50	298	200	388	350	436	500	470
55	303	205	390	355	437	510	472
60	308	210	392	360	438	520	474
65	312	215	394	365	440	530	476
70	316	220	396	370	441	540	478
75	320	225	397	375	442	550	480
80	328	230	399	380	444	560	482
85	328	235	401	385	445	570	483
90	331	240	403	390	446	580	485
95	335	245	404	395	447	590	487
100	338	250	406	400	448	600	489
105	341	255	408	405	449	610	491
110	344	260	410	410	451	620	492
115	347	265	411	415	452	630	494
120	350	270	413	420	453	640	496
125	353	275	414	425	454	650	497
130	356	280	416	430	455	660	499
135	358	285	417	435	456	670	501
140	361	290	419	440	457	680	502
145	364	295	420	445	458	690	504

To obtain superheat, subtract temperature given above from total initial temperature. Pressure in kPa = 6.895 × lb/in². Temperature in °C = (°F − 32) ÷ 1.8.

than a turbine with a higher steam rate. The operating costs are the cost of the steam for the number of years upon which the evaluation is to be based:

$$\text{Total cost} = \text{initial cost} + \left(\text{power} \times \frac{\text{steam}}{\text{rate}} \times \frac{\text{steam}}{\text{cost}} \times \frac{\text{operating hours}}{\text{per year}} \times \frac{\text{number}}{\text{of years}}\right)$$

The cost of installation—foundation, steam piping, and cooling water service (also electric service if required)—is not normally considered unless there are significant differences between the turbines: single-stage versus multistage, direct-connected versus turbine and gear, vertical turbine versus horizontal turbine with a right-angle gear, and so on.

The economical selection of a steam turbine versus an electric motor, diesel engine, gas engine, gas turbine, and so on, must include an evaluation of all installation costs, costs of supporting equipment (starters, switch gear, fuel-supply system, and so on), operating costs, and the initial costs of the various drivers being considered.

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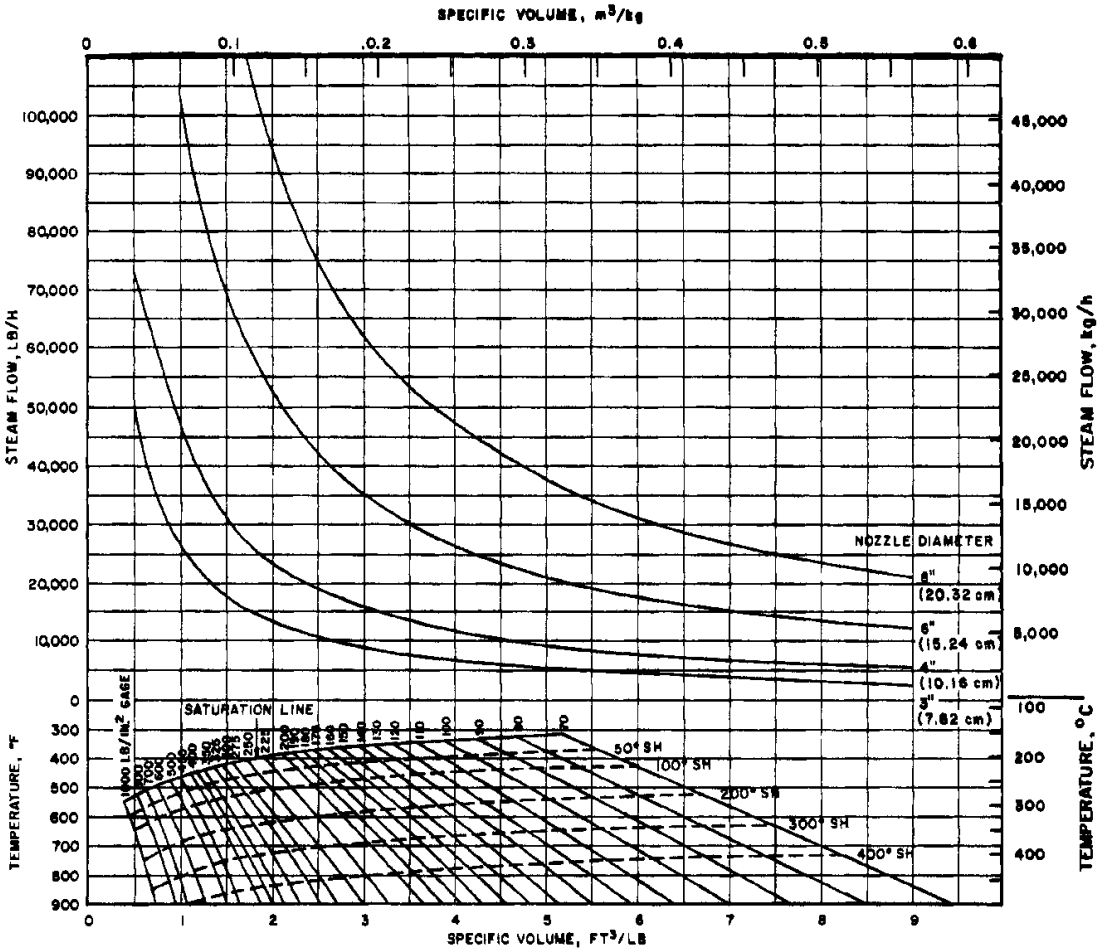


FIGURE 10 Nominal inlet flow capacity. Read inlet nozzle size required to pass maximum flow, based on 150-ft/s (45.7-m/s) steam velocity (lb/in² × 6.895 = kPa; °F_{SH} × 0.555 = °C_{SH})

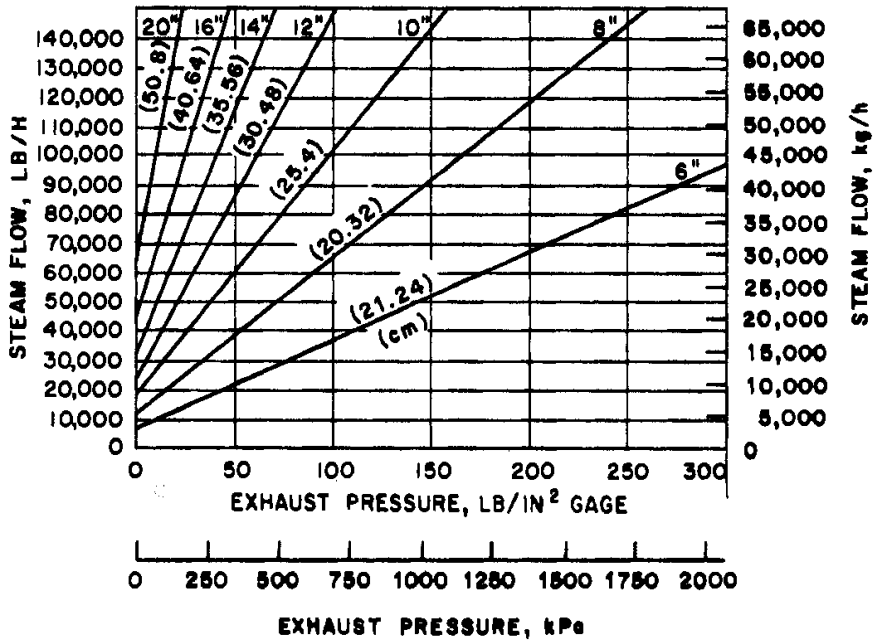


FIGURE 11 Nominal exhaust flow capacity. Noncondensing exhaust nozzles, based on 200-ft/s (61 m/s) steam velocity

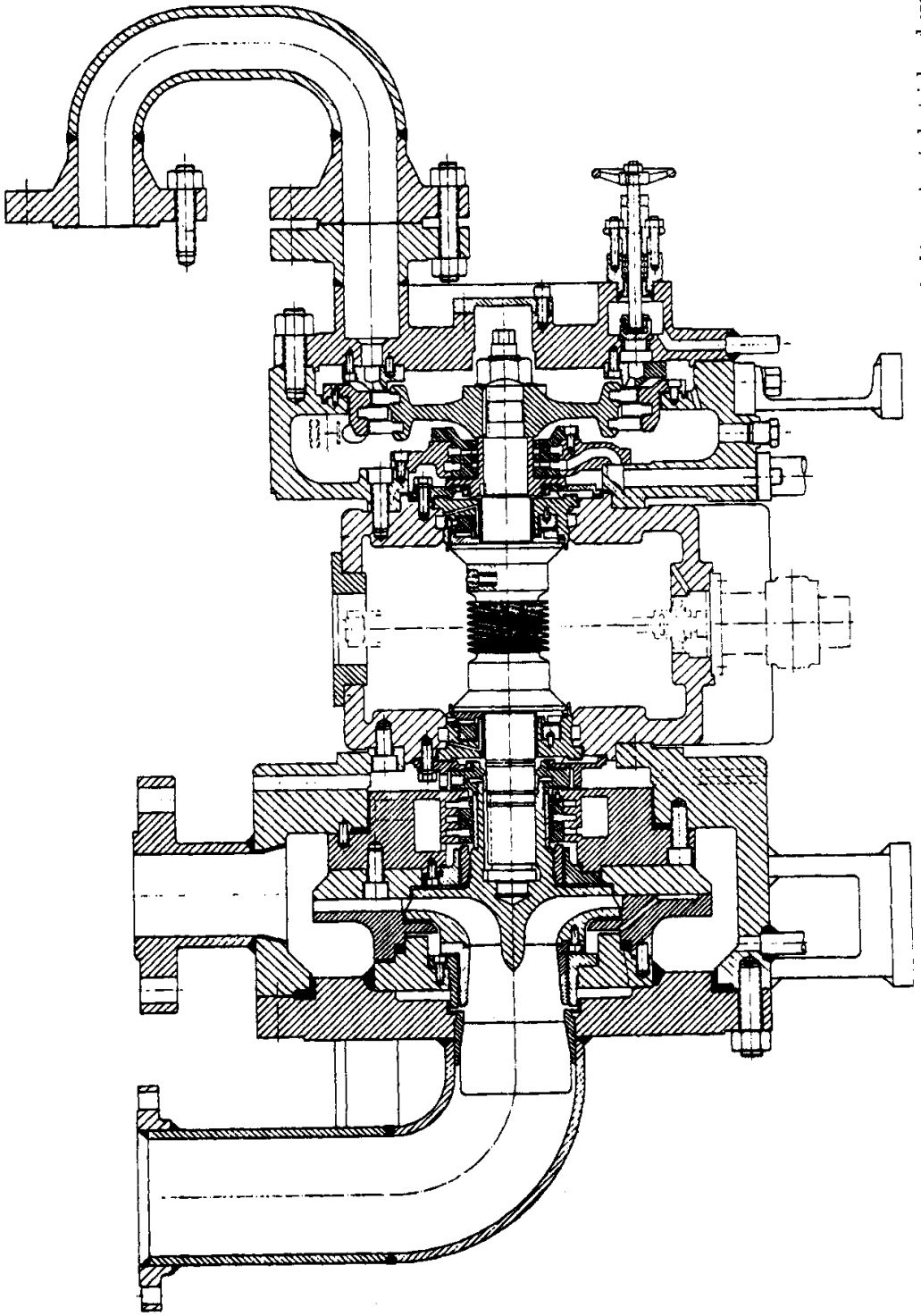


FIGURE 12 High-speed centrifugal pump and single-stage steam turbine mounted on a common shaft. Package is designed for boiler feed in marine, industrial, and process steam plants (Flowserve Corporation).

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REFERENCES

1. Church, E. F. *Steam Turbines.* McGraw-Hill, New York, 1950.
2. "Single-Stage Steam Turbines for Mechanical Drive Service." NEMA pub. SM22-1970, 1970.
3. Keenan, J. H., and Keyes, F. G. "Thermodynamic Properties of Steam." John Wiley & Sons, Inc., New York, 1936.

FURTHER READING

Bergeron, W. L. "Governors and Controls: Each Has Specific Operation." *Sugar J.,* March 1965.

"Multistage Steam Turbines for Mechanical Drive Service." NEMA pub. SM21-1970, 1970.

Skrotzki, B. G. A. "Steam Turbines." *Power* special report, 1962.

Steen-Johnsen, H. "Here's What You Should Know Before You Specify Control Systems for Mechanical Drive Steam Turbines." *Power Eng.,* February–March 1966.

Steen-Johnsen, H. "How to Estimate the Size and Cost of Mechanical Drive Steam Turbines." *Hydrocarbon Processing,* October 1967.

Steen-Johnsen, H. "Mechanical Drive Turbines." *Oil Gas Equip.,* March, April, May, 1967.

Steen-Johnsen, H. "Selecting the Right Steam Turbine for Industrial Use." *Power Eng.,* April 1965.

"Theoretical Steam Rate Tables." ASME, 1969.

9.1.3
ENGINES

F. J. GUNTHER

ENGINE SELECTION AND APPLICATION

The internal combustion engine is used extensively as a driver for centrifugal and displacement pumps. Depending on the application, the engine fuel may be gasoline, natural gas, liquid petroleum gas (LPG), sewage gas, or diesel fuel. It may be either liquid- or air-cooled.

Basic Design Variations The cylinder block construction is vertical or horizontal, in-line or V-type. The number of cylinders ranges from 1 to 20. The engine cycle is either four (one power stroke in two revolutions of the crankshaft) or two (one power stroke in one revolution of the crankshaft). The combustion chamber and cylinder head design are classed as L-head (valves in the cylinder block) or valve-in head. In the diesel engine, the combustion chamber may be of the precombustion chamber design, where an ante chamber is used to initiate combustion, or of the direct injection design, where the fuel is injected directly into the cylinder. The piston design action may be vertical, horizontal, at an angle as in the V-type engine, or opposed piston (two pistons operating in the same cylinder).

Power Ratings The power range of engines in current production, depending on displacement, number of cylinders, and speed, is as follows:

1. Air-cooled gasoline, natural gas, and diesel: 1.0 to 75 hp (0.75 to 56 kW)
2. Liquid-cooled gasoline: 10 to 300 hp (75 to 224 kW)
3. Liquid-cooled natural gas, LPG, and sewage gas: 10 to 15,000 hp (75 to 11,200 kW)
4. Liquid-cooled diesel: 10 to 50,000 hp (75 to 37,300 kW)
5. Dual fuel, natural gas, LPG, and diesel: 150 to 25,000 hp (112 to 18,700 kW)

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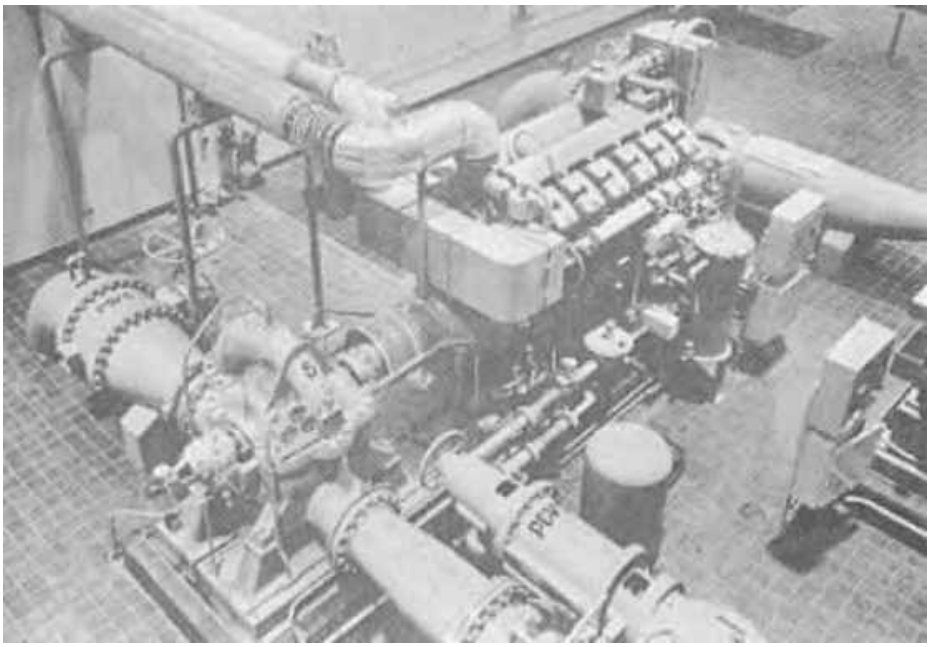


FIGURE 1 Natural gas engine driving a horizontal water pump for Winnipeg, Canada, water utility (Waukesha Motor)

Typical engine driver applications are represented in Figures 1 and 2.

The rating of the internal combustion engine is the most important consideration in making the proper selection. The general practice is to rate engines according to the severity of the duty to be performed. The most common rating classifications are maximum, standby or intermittent, and continuous.

The maximum output is based on dynamometer tests that are corrected to standard atmospheric conditions for temperature and barometric pressure. In applications, this power rating is reduced by accessories such as cooling fans, air cleaners, and starting systems.

Standby, or intermittent, and continuous ratings are arrived at by applying a percentage factor to the net maximum power rating. For example, 75 to 80% is used for continuous and 90% for intermittent.

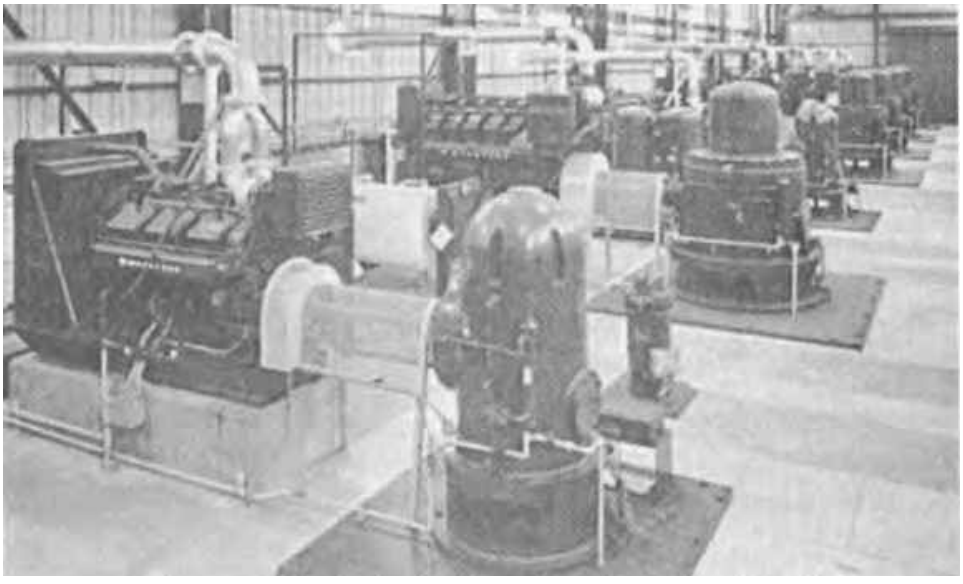


FIGURE 2 Diesel engine driving pumps for flood-control station in Seattle (Waukesha Motor)

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Duty cycle is a term used to describe the load pattern imposed on the engine. If the load factor (ratio of average load to maximum capabilities) is low, we call the duty cycle "light," but if it is high, we classify the cycle "heavy." Continuous, or heavy-duty, service is generally considered to be 24 h/day, with little variation in load or speed. Intermittent service is classified as duty where an engine is called upon to operate in emergencies or at reduced loads at frequent intervals.

In analyzing power problems when selecting a proper engine, certain terms are used in the industry:

DISPLACEMENT The displacement in cubic inches (cubic centimeters) of an engine is

In USCS units, D = bore (in^2) × 0.7854 × stroke (in) × no. of cylinders

In SI units D = bore (cm^2) × 0.7854 × stroke (cm) × no. of cylinders

TORQUE The twisting effort of the engine in pound-feet (Newton-meters) is

In USCS units $$T = 5252 \times \frac{\text{bhp}}{\text{rpm}}$$

In SI units $$T = 9545 \times \frac{\text{bkW}}{\text{rpm}}$$

ENGINE POWER Brake horsepower (bkw) is the useful power at the flywheel:

In USCS units $$\text{bhp} = T \times \frac{\text{rpm}}{5252}$$

In SI units $$\text{bkW} = T \times \frac{\text{rpm}}{9545}$$

SPECIFIC FUEL CONSUMPTION The mass flow rate of fuel consumed per unit time divided by the power output, usually at the engine rating, measures how efficiently an engine is using the fuel supplied to produce that power. It is measured in pound-mass per brake horsepower- hour (gram per kilowatt- hour).

In USCS units $$\text{bsfc} = \frac{\text{lb mass of fuel per hour}}{\text{bhp}}$$

In SI units $$\text{bsfc} = \frac{\text{grams of fuel per hour}}{\text{bkW}}$$

BRAKE MEAN EFFECTIVE PRESSURE The average cylinder pressure to give a resultant torque at the flywheel in pounds per square inch (kilopascals)

In USCS units $$\text{bmep} = \frac{792{,}000 \times \text{bhp}}{\text{rpm} \times D} \text{ (four-cycle)}$$

$$\text{bmep} = \frac{396{,}000 \times \text{bhp}}{\text{rpm} \times D} \text{ (two cycle)}$$

In SI units $$\text{bmep} = \frac{120 \times 10^6 \times \text{bkW}}{\text{rpm} \times D} \text{ (four-cycle)}$$

$$\text{bmep} = \frac{60 \times 10^6 \times \text{bkW}}{\text{rpm} \times D} \text{ (two cycle)}$$

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PISTON SPEED At a given speed, the average velocity of piston in feet per minute (meters per second) is

In USCS units $$\text{Piston speed} = \text{stroke (in)} \times 2 \times \frac{\text{rpm}}{12}$$

In SI units $$\text{Piston speed} = \text{stroke (cm)} \times 2 \times \frac{\text{rpm}}{6000}$$

In selecting an engine for a particular application, the following variables should be considered:

- Altitude
- Ambient air temperature
- Rotation and speed
- Bmep and piston speed
- Maintenance
- Type of fuel
- Operating atmosphere (dust and dirt)
- Vibrations and torsionals
- Engine pollutants

The observed power is that produced by an engine at the existing altitude and temperature. All engine manufacturers publish power ratings corrected to certain conditions; for conditions other than these, it is necessary to correct by applying a percentage factor for altitude and temperature. Generally this is $3\frac{1}{2}\%$ per thousand feet (305 m) above sea level and 1% for every 10°F (5.6°C) above 60°F (50.4°C). In a turbocharged engine, there is no established standard and the engine manufacturer should be consulted.

The basic rotation of engines in current production is counterclockwise when viewed from the flywheel end of the engine, although many of the larger engines are available in both counterclockwise and clockwise rotation. The speed of the engine is generally fixed by the equipment being driven. Through the use of speed-increasing or -reducing gear boxes, the proper engine for a given application may be selected. A gear box may also be used to correct a rotation problem.

Speed ranges for engines generally fall into three categories:

- High—above 1500 rpm
- Medium—700 to 1500 rpm
- Low—below 700 rpm

High-speed engines generally offer weight and size advantages as well as cost savings and thus are used for standby applications. On the other hand, medium- or low-speed engines, although heavier and larger, offer a gain in service life and lower maintenance costs.

The speed flexibility of an engine drive is important when the engine is to be used to drive a pump that must move variable quantities of liquid. The engine speed may be changed very simply either manually or through the use of liquid or pressure controls.

Bmep is generally a measure of load, and piston speed a measure of potential wear and maintenance. Although the introduction of the turbocharged and intercooled engine has somewhat changed the consideration given these factors, it is still important to consider them in selecting engines where long life is a factor.

Maintenance of engines has been considered by some as objectionable and more costly than electric power. A recent innovation of engine manufacturers, in the form of a service contract for installations where trained personnel are not available or desirable for economic reasons, can eliminate these objections and costs. The complete maintenance of the engine is done on a fixed-fee basis for a designated period of time.

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The exhaust gases of spark ignition and compression ignition (diesel) engines contain pollutants that for many engine applications are increasingly the subject of legislation restricting the quantity of pollutants the engine can emit. Examples of pollutants are carbon monoxide (CO), oxides of nitrogen (NOx), unburned hydrocarbons (HC), and, for diesels, particulates in the form of carbon soot. Pollutants can be measured on a specific basis, such as g/bhp-hr and ppm (parts per millions by volume), or on a site basis, such as lb/hr or tons/year. Engine manufacturers are designing "clean" engines that incorporate features to minimize the formation of pollutants during combustion and may include catalytic converters and particulate fibers in the exhaust system to further reduce the pollutants emitted in the exhaust gases.

The remaining conditions listed previously will be discussed in detail later.

FUEL SYSTEMS

Gasoline Gasoline is used primarily with standby pumping units. Inasmuch as the spark ignition system first introduced the internal combustion engine to power applications, gasoline was used as the primary fuel. Commercial gasoline has an average heating value of 19,000 Btu/lb (44.2 MJ/kg). It is easy to transport and handle and, unlike gaseous fuels, does not require pressure storage and regulating equipment. The starting capabilities of a gasoline engine are satisfactory, provided the engine is in good operating condition. With the high-power engine of today, refinery control can produce a fuel matched to the octane requirement of the engine.

Gasoline does have some disadvantages that are reducing its use as an engine fuel. In small-volume usage, it is safe and easily handled. In larger volumes, it becomes expensive and hazardous. Gasoline will form gums and resins when exposed to air over a period of time. There is also the possibility of condensation of water in the fuel, which is detrimental to good operation. The danger of fire is always present because of leaks in the system. Finally, the increased production and distribution of gasoline have made it a target of increasing taxation, making it economically prohibitive in many installations.

Gas A gaseous fuel system using natural gas, LPG, or sewage gas may be a simple manually controlled system, such as a gasoline engine, or a carefully engineered automatic system. The basic gas carburetion system consists of a carburetor and pressure regulator mounted on the engine. A gas distribution system, like a water supply system, must be at some designated pressure and flow, and so a field pressure regulator is required. The characteristics of this regulator will depend upon the gas analysis, the displacement of the engine, the speed range, and local regulations. A typical schematic of a gas fuel system is shown in Figure 3. The location of the regulator is generally under the jurisdiction of the gas utility that supplies it. In most cases, a single field regulator is all that is required, but at times this can cause problems. For example, subsequent installation of gas-burning equipment used intermittently may cause gas pressure regulation not compatible with the

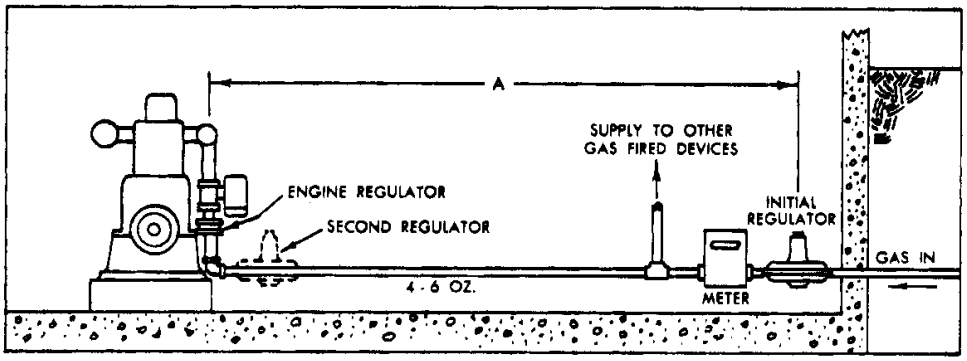


FIGURE 3 Typical natural gas fuel system (Waukesha Motor)

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small amount required for pilot lighting. A single regulator installed some distance from the engine could result in hard starting because the engine vacuum is not sufficient for a full gas flow. To eliminate this problem, the initial regulator is set at a higher pressure in order to give a readily available supply of fuel for all devices.

Natural gas has an average heating value of 800 to 1000 Btu/ft^3 (29.8 to 37.3 MJ/m^3). Commercial butane has a value of 2950 Btu/ft^3 (110 MJ/m^3), and propane a value of 3370 Btu/ft^3 (126 MJ/sn^3). Commercial LPG fuel, which is a mixture of butane and propane, varies in both amount and corresponding heating value.

LPG fuel is produced by mechanical and compression processes, and the methods of distribution and handling must meet regulations. Natural gas is usually supplied under moderate pressures, seldom exceeding 50 lb/in (345 kpa), whereas LPG fuel is supplied as a liquid under pressures as high as 200 lb/in^2 (1380 kPa) in warm weather. As a result, natural gas will readily mix with air and burn, whereas LPG fuel must be changed from a liquid to a vapor with the addition of heat, as shown in Figure 4.

As a natural process in the modern waste-treatment plant, sewage gas may be produced in the sewage digester. This gas has the same basic qualities as natural gas and is composed of about 65 to 70% methane. It has a heating value of 550 to 700 Btu/ft^3 (20.5 to 26.1 MJ/m^3). The same basic carburetion system used for natural gas is used. Sewage gas contains inert substances, particularly hydrogen sulfide or free sulfur, which in the presence of free moisture or moisture resulting from combustion will form sulfurous acid, which is corrosive and thus damaging to the valves, pistons, and cylinder walls of an engine. An engine can tolerate from 10 to 30 g of sulfur per 100 ft^3 (350 to 1100 g per 100 m^3). Beyond this, a filtering system to remove the sulfur and moisture is advisable.

Diesel The diesel engine over the years has been used for larger power systems. The initial cost of the system is justified to an extent by lower cost of the fuel. The better fuel economy of the diesel engine and its torque characteristics also are important factors in the selection of this fuel for many applications. Diesel fuel has one distinct advantage: it does not form the dangerous fuel vapors that other fuels do. It does require, however, a good fuel-filtering system because of contaminants in the fuel that can create problems for the precision design of the fuel-injection system.

Commercial diesel fuels are the residue that remains after the more volatile fractions of crude oil have been removed. The heating value is generally about 19,000 Btu/lb (44 MJ/kg). In the diesel engine, the fuel is injected into the cylinder at the end of the compression stroke in an atomized form. The compression stroke results in a temperature sufficient to ignite the fuel without the use of any ignition device. Although fuel systems from different engine manufacturers vary, the basic components are the same.

The larger diesel engines may be designed to operate on a dual fuel system. The engine operates on gaseous fuels with a pilot injection of diesel fuel for ignition. In case of a loss of the gaseous fuel supply, the engine will convert to 100% diesel fuel.

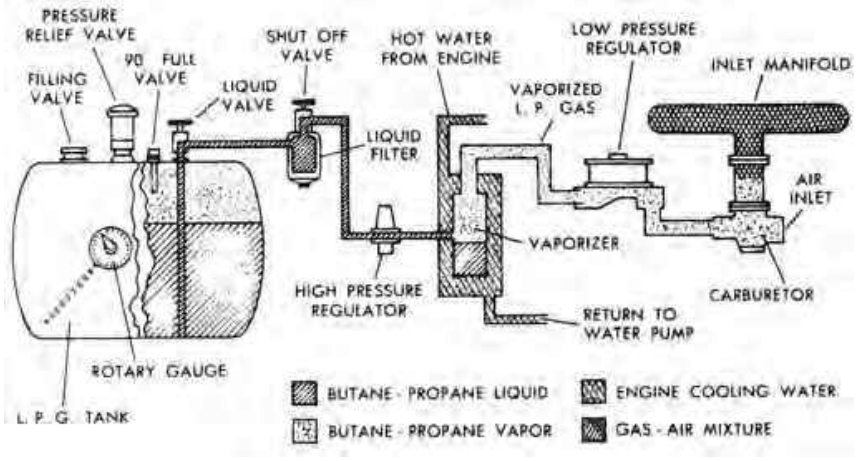


FIGURE 4 Typical LPG fuel system (Waukesha Motor)

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Relative Performance Curves Typical performance curves comparing power, torque, and part-load fuel economy are shown in Figure 5.

COOLING SYSTEMS

Cooling is essential in all internal combustion engines because only a small portion of the total heat energy of any fuel is converted to useful energy. The remainder is dissipated into the coolant, exhaust, and lubricating oil and by radiation. A hypothetical heat balance is shown in Figure 6.

Specific data and recommendations on cooling requirements vary from one manufacturer to another. In general, the heat rejection to an engine cooling system will range between 30 and 60 Btu/hp · min (25 and 51 MJ/kWh) for diesel engines and up to 70 Btu/hp· min (59 MJ/kWh) for natural gas and gasoline engines. This heat must be transferred to some form of heat-exchange medium.

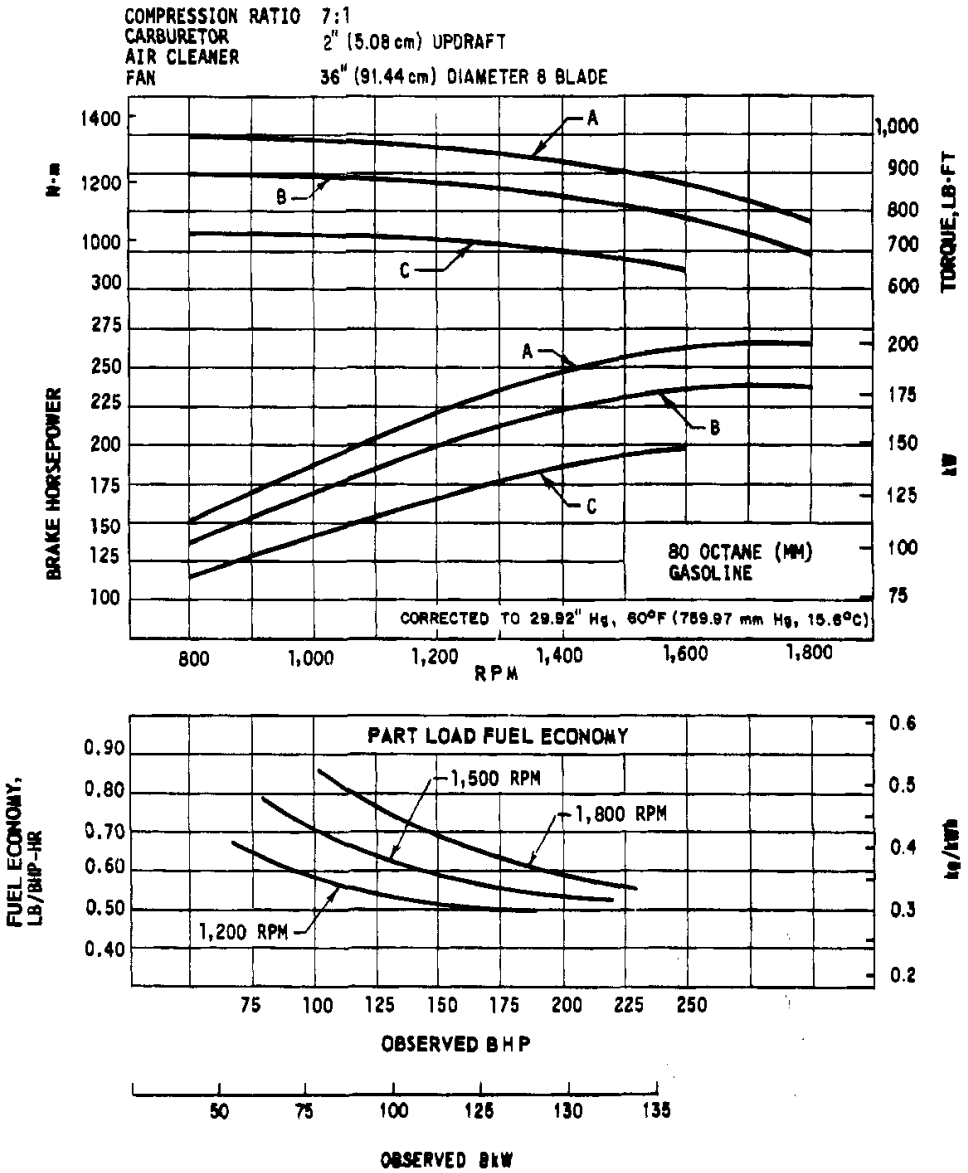


FIGURE 5A Gasoline engine performance curves: (A) maximum, (B) intermittent, (C) continuous ratings of engines with accessories (Waukesha Motor)

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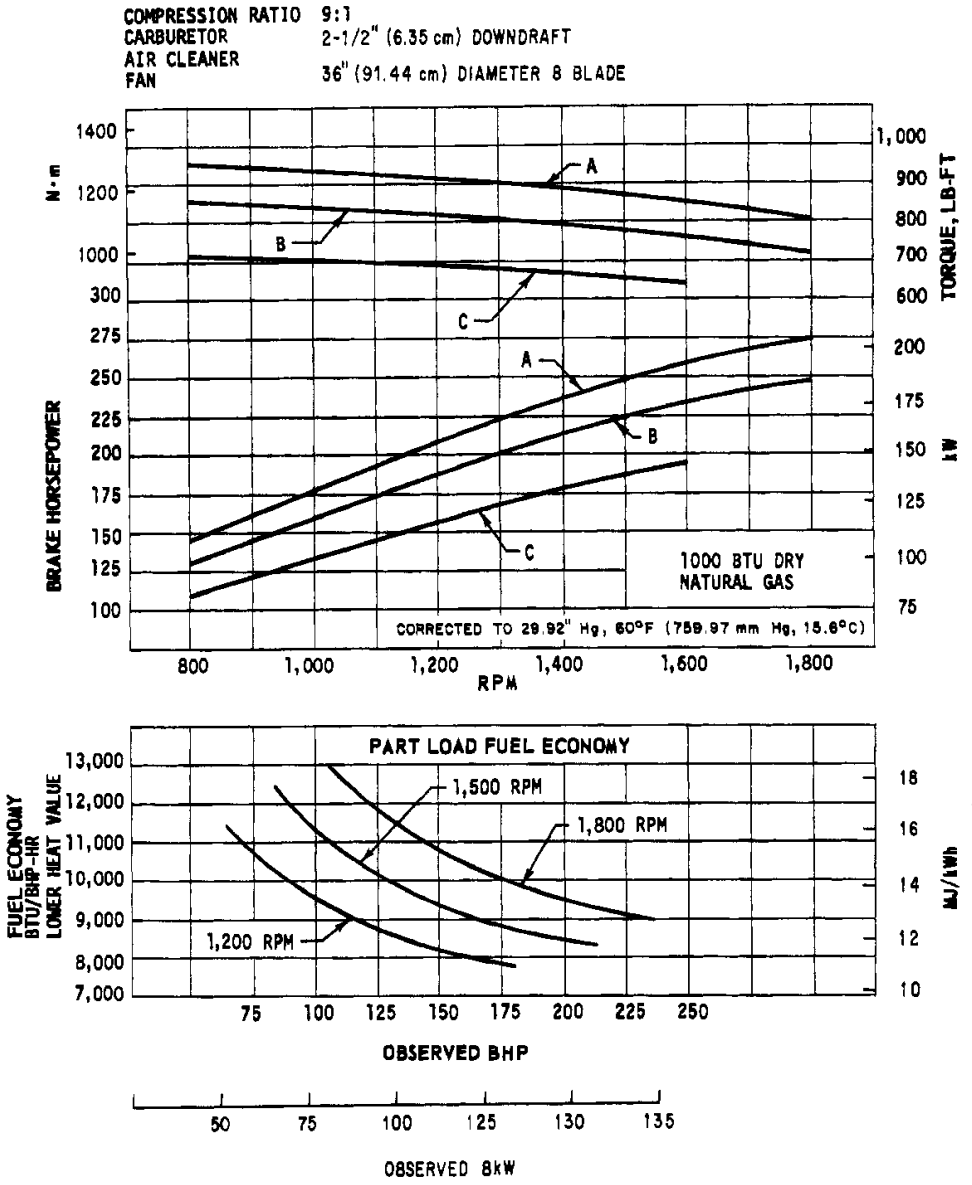


FIGURE 5B Gas engine performance curves: (A) maximum, (B) intermittent, (C) continuous ratings of engines with accessories (Waukesha Motor)

In designing any cooling system, certain factors must be considered:

- Additional heat from driven equipment, such as the cooling of speed-reducing or -increasing gears where the engine coolant is the medium
- Water-cooled exhaust manifolds on the engines or water-cooled exhaust turbochargers or after-coolers
- High ambient temperatures or heat from nearby equipment
- Variation in the heat-exchanger coolant temperature
- Entrapment of substantial quantities of air in the coolant water
- Inability to maintain a clean cooling system

With any cooling system, one of the most important factors of design is the temperature drop across the engine. Most engine manufacturers desire a temperature differential of no more than 10 to 12°F (5.6 to 6.7°C), and closer values are desirable. A jacket-water

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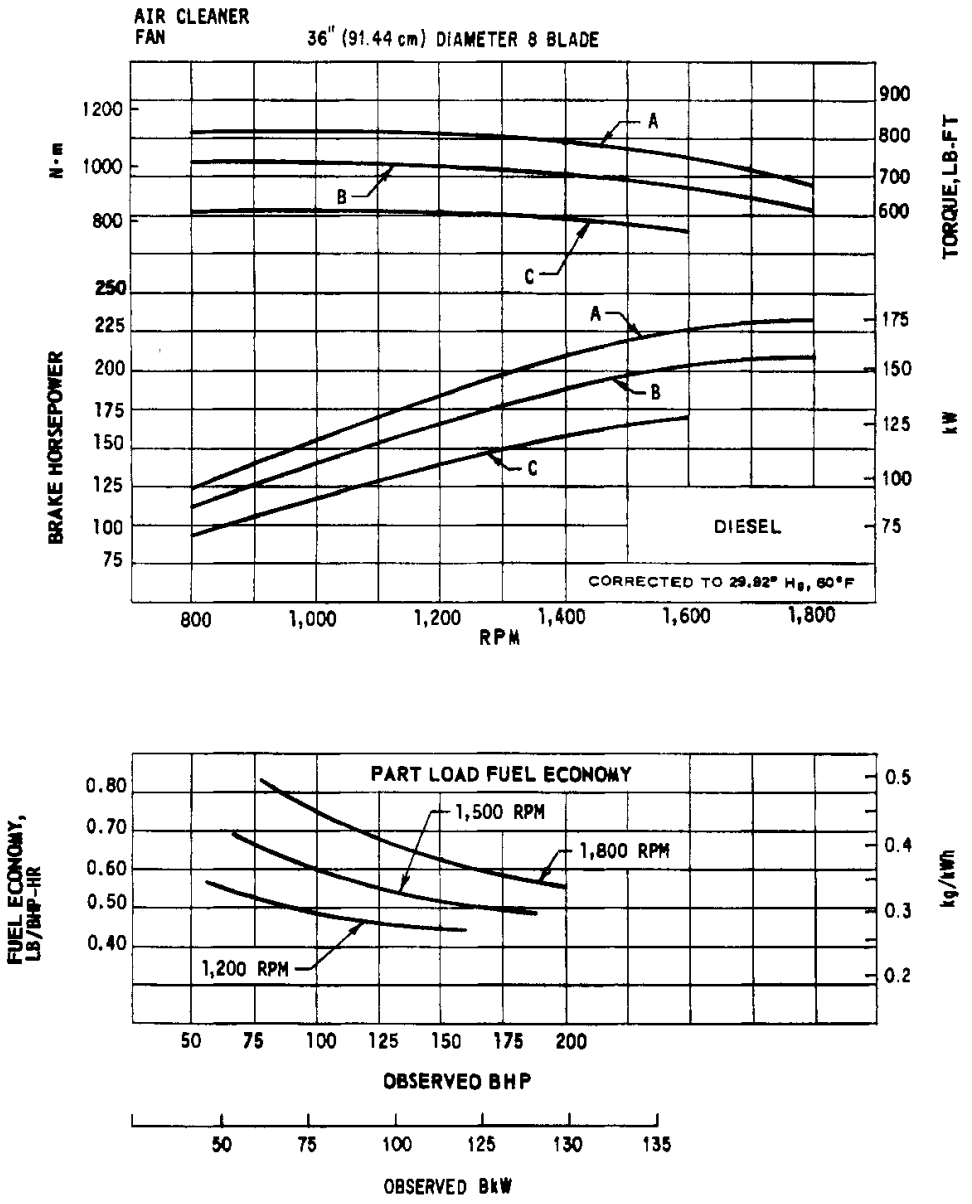


FIGURE 5C Diesel engine performance curves: (A) maximum, (B) intermittent, (C) continuous ratings of engines with accessories (Waukesha Motor)

temperature across the engine of 170°F (77°C) is preferred, and in high-temperature or waste-heat-recovery systems, temperatures of 200°F (93°C) or more are common and not harmful.

Radiator The radiator cooling system is perhaps the most common and best understood method of cooling. It is based on a closed system of tubes through which the jacket water passes. The heat is dissipated by a fan, creating a stream of moving air passing through the tubes. The fan is driven either by the engine or by an auxiliary source of power (Figure 7).

An engine in a fixed outside installation can be cooled without much difficulty. Certain factors must be considered, such as ambient temperature, direction of the prevailing wind, and presence of foreign airborne materials. In high temperatures (usually above 110°F [48°C]), a larger radiator is required. If the prevailing winds are extremely high, the unit can be located to offset normal fan flow. Screening can be used to prevent the clogging of the air passes in the radiator where the atmosphere tends to contain foreign airborne material, such as dust.

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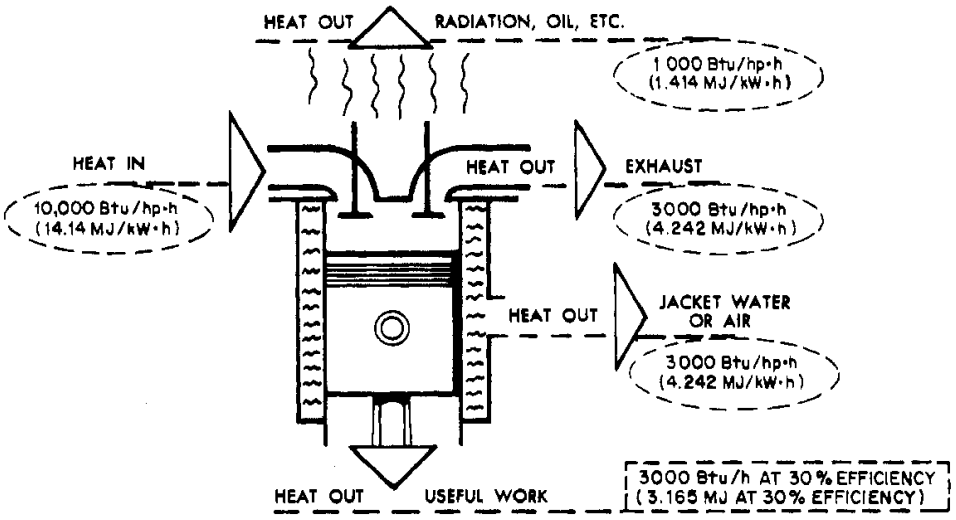


FIGURE 6 Hypothetical heat balance (Waukesha Motor)

Radiator cooling may be used in an inside installation, but there are certain problems which, unless properly anticipated, limit this system. The recirculation of cooling air and the radiation of exhaust heat from the engine create a problem. As was previously pointed out, every 10°F (5.6°C) rise above 60°F (15.6°C) results in a 1% loss in power. When 5 to 10% of the total heat put into an engine is radiated, some means of power ventilation must be provided to remove this heat through ducts, louvers, or forced ventilation with a separate fan. An installation of this type is illustrated in Figure 8.

Heat Exchanger The heat exchanger cooling system (Figure 9) is the best system for a stationary engine installation. Using a tube bundle in a closed shell, the cooling exchange medium is water (often called raw water). This water may be plant or process water; it may be recirculated or, in standby installations, allowed to pass to waste. The system has the advantage of the radiator cooling system in that it is self-contained: the quantity and quality of the water in the engine can be controlled. It has the further advantage of not being affected by the flow of heat to air movement if the heat of radiation is taken into account in the design of the system. On the other hand, the cooling medium, unless used in a plant system, is a disadvantage because it is costly. A separate pump is required to provide the necessary water for cooling unless city water or process water is under sufficient pressure.

City Water and Standpipe City water cooling is designed to take water directly from the city main or from the pump the engine is driving. It is used on some emergency or

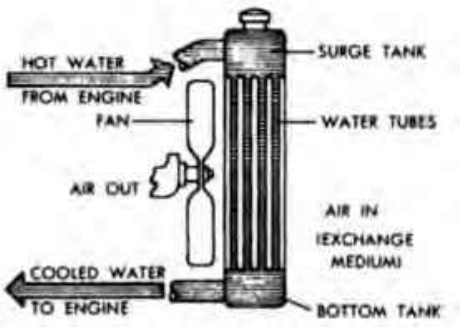


FIGURE 7 Radiator cooling system (Waukesha Motor)

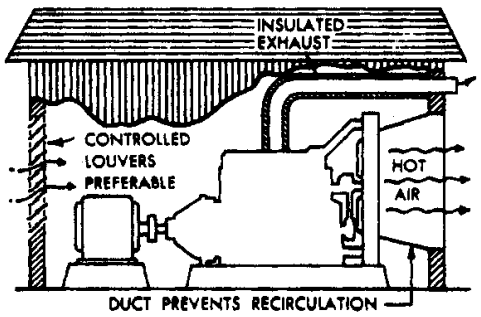


FIGURE 8 Radiator cooling system for an inside installation (Waukesha Motor)

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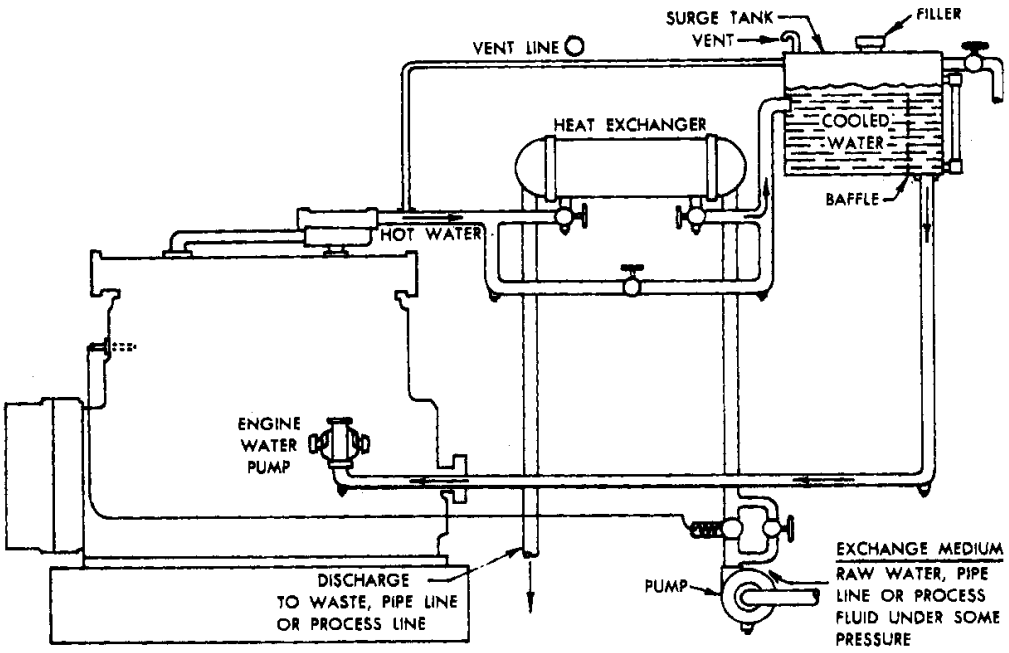


FIGURE 9 Heat exchanger cooling system (Waukesha Motor)

standby installation. It is simple and inexpensive, gives unlimited cooling for moderate-size engines, is easily understood, and will operate instantly in an emergency. On the negative side, the cooling water is wasted, corrosive elements may be introduced into the engine jacket water system, and it may create excessive temperature changes across the engine jacket.

Standpipe cooling (Figure 10) is basically the same as city water cooling except that a thermostatic valve is employed to admit makeup water as required. The vertical pipe is a

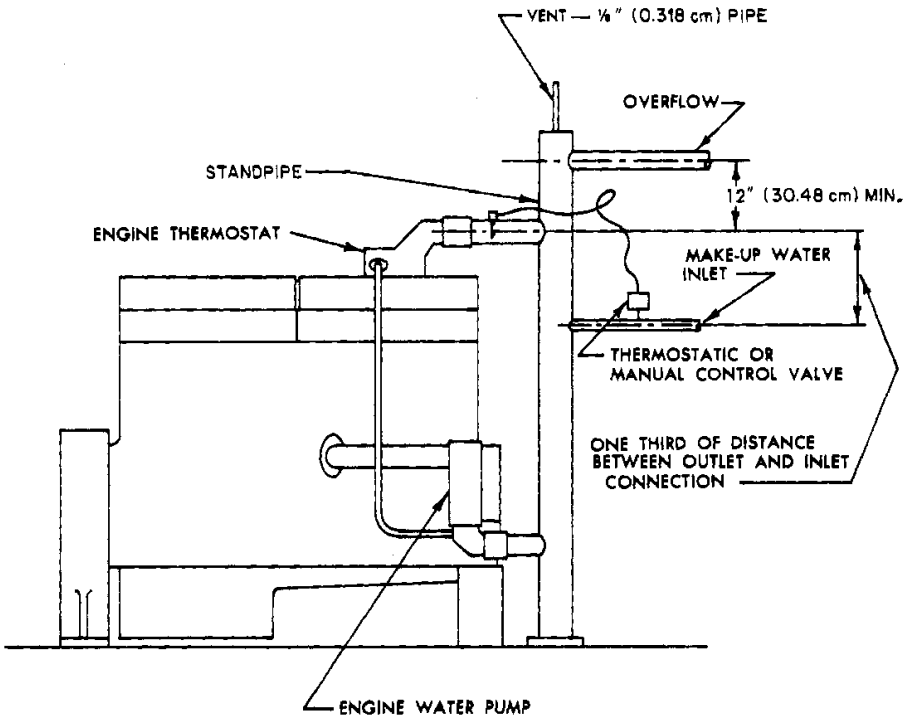


FIGURE 10 Standpipe cooling system (Waukesha Motor)

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blending tank into which city water is introduced only in the amount necessary for makeup. The standpipe system is inexpensive and simple to operate.

Ebullition In installations where heat is required for process equipment, a method of high-temperature, or ebullition, cooling is being used as a very economical method, particularly with larger installations. This system has been termed *steam cooling, high-temperature,* or *Vapor-phase* (a registered trademark). In this system, the coolant leaves the engine at a temperature equal to or above its atmospheric boiling point and under sufficient pressure to remain liquid until discharged from the engine into a flash chamber, where a drop in pressure causes the formation of steam, which is condensed and returned to the engine at very near the discharge temperature. A schematic of this system is shown in Figure 11. This system has the advantages of a very small temperature differential across the engine, which minimizes distortion of all working parts, and a constant working temperature regardless of load. Because of the higher working temperatures, the combustion area and crankcase of the engine have fewer liquid by-products of combustion and corrosive materials. Of prime importance is the waste heat that can be recovered for plant process with a very small amount of makeup water for cooling.

Cooling Tower Cooling towers are used in some large or multiple-engine installations. Through the use of a current of air, produced either by a natural draft or by mechanical means, a tower causes a sensible heat flow from the cooling water to the air. Atmospheric, or natural, draft towers depend upon natural wind velocities and thus can vary widely. Mechanical draft towers, where the air supply can be controlled, can be put in any area, but the limit to their cooling capacity is the power required to operate them. As the water volume increases, the volume of air required and the pressure the fan has to operate against increase. A point is reached where the cost of installation and operation becomes prohibitive. A diagram of a system that combines components of the cooling systems mentioned plus the waste-heat recovery system silencer, to be mentioned later, is shown in Figure 12.

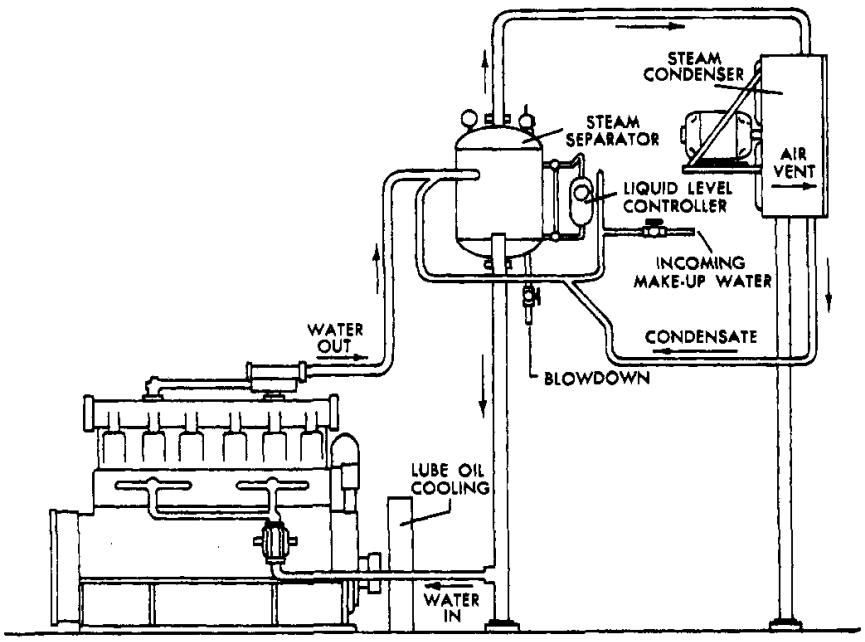


FIGURE 11 High-temperature cooling system (Waukesha Motor)

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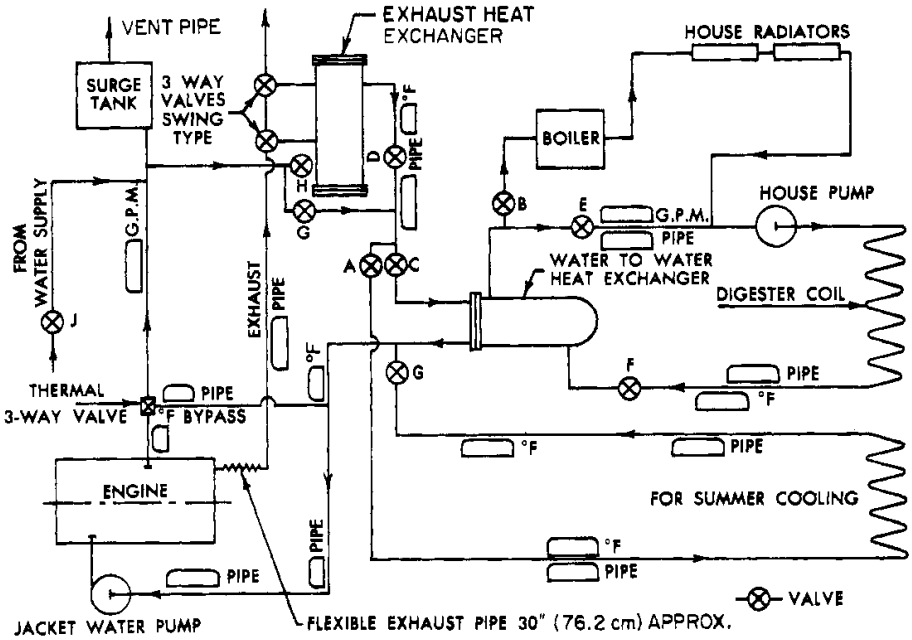


FIGURE 12 Complete cooling system, including waste-heat recovery (Waukesha Motor)

AIR-INTAKE SYSTEMS

A most important consideration in the application of an engine to any pump drive is the engine's ability to "breathe." As air is required for combustion, the design engineers of any project must take into account the necessary provisions for this air. The environment, the service, the speed range of the engine, the duty cycle, and the location from which the combustion air is to be taken are of vital importance.

The first step in the selection of any intake system is to determine the volume of air, in cubic feet per minute (cubic meters per second) required for combustion, which may be calculated by the formula:

$$V_{\text{ideal}} = \frac{B^2 \times S \times \text{rpm} \times N}{C \times K}$$

where B = cylinder bore, in (cm)
S = piston stroke, in (cm)
rpm = engine speed, revs/minute
N = number of cylinders
C = 2,200 for V_{ideal} in CFM or 76.39×10^6 for V_{ideal} in m^3/s
K = 1 for two-stroke cycle and 2 for four-stroke cycle

Volumetric efficiency, defined as the ratio of actual air flow to ideal air flow, will vary with engine design, but an average of 80% may be used to determine the air cleaner size. Two-stroke cycle engines will require approximately 140% air flow calculated by the above equation. For supercharged and turbocharged engines, the air flow requirements should be obtained from the manufacturer.

The two basic cleaners available are wet and dry. The wet, or oil-bath, cleaner consists of either an oil wire mesh or an oil bath through which the air must pass. A dry cleaner uses a paper or cloth filter that traps dust, lint, and so on but allows the air to pass through.

The installation of a suitable air cleaner is important. In cases where adequate air can be supplied through proper ventilation of the area surrounding the engine, it is best to

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mount the cleaner on the engine. If it is necessary to bring air to the engine from outside the area of the building, certain design factors must be considered. The pipe connections from an outside cleaner should be tight and mechanically strong, and fabric hose should not be used unless the length is relatively short. The air to the engine should not be heated by close proximity to the engine or any other heating device because, as previously mentioned, power loss occurs with air temperatures above 60°F (15.6°C). To avoid restrictions in the system, there should be no sharp bends in the piping. Finally, any outside air cleaner must be designed so moisture such as rain or snow cannot enter the system.

The air cleaner system on a turbocharged engine must be able to remove any impurities in the air that would be detrimental to the efficiency of the turbocharger. Because of the increased air requirements, a larger air-cleaner system must be used on such units.

EXHAUST SYSTEMS

An engine consumes a large volume of air for combustion, and that volume must be removed after combustion. It is necessary that the exhaust back pressure be kept at a minimum while this volume is being removed. The exhaust piping should be properly sized, and long sweep elbows should be employed if necessary. Unless the back pressure is kept low, the following conditions can result:

- Loss of power
- Poor fuel economy
- High combustion temperatures with increased maintenance
- High jacket water temperatures
- Crankcase sludging with resulting corrosion and bearing wear

All internal combustion engines create noise. Depending upon the location of the engine, this noise can be a problem. Normally, when measured from a distance of about 10 ft (3 m), an unmuffled engine creates a decibel noise level ranging from 100 for the small- and medium-size engine to 125 for the larger engine. Thus most engine installations incorporate some form of exhaust silencer, or muffler. Depending upon the degree of silencing, a muffler will reduce the unmuffled decibel reading by 30 to 35 dB. Various types are manufactured to meet the required conditions and are classified as follows:

- Standard or industrial
- Semiresidential (high-degree)
- Residential or hospital (supercritical)

The basic exhaust silencer is designed as either a dry type or wet type. The latter is used in installations such as sewage or water treatment plants, where the recovery of heat for plant processes is important. Basically this type may be classified as a low-pressure boiler. Water is admitted to the silencer through tubes or coils to pick up the heat from the exhaust. As shown in Figure 6, approximately 30% of the total heat input into the engine is exhausted. The wet-type silencer is designed to regain about 60 to 70% of this beat. This silencer has an advantage in its ability to operate either wet or dry. Thus when heat is not required, it may be operated dry and vice versa.

STARTING SYSTEMS

Starting methods for engines fall into two broad categories: direct and auxiliary.

Direct The direct system, used primarily with large engines, employs some means of exerting a rotating force on the crankshaft, such as the introduction of high-pressure air

directly into the cylinders of the engine. A direct system on small air- or water-cooled engines using either a rope or a hand crank has to a large extent been discarded in favor of an auxiliary method.

Auxiliary The auxiliary system employs a small gear that meshes with a larger gear (ring gear) on the engine flywheel. The ratio of the number of gear teeth on the large gear to those on the small gear is called the *cranking ratio.* Generally, the larger the ratio, the better the cranking performance. The auxiliary system uses several means to drive the small gear:

- Electric motor
- Air motor
- Hydraulic motor
- Auxiliary engine

ELECTRIC MOTOR The electric motor may be either dc or ac. The dc motor most commonly used is available in 6, 12, 24, or 32 V. The voltage size will depend upon the size of the engine, the ambient temperature, and the desired cranking speed of the engine. The dc system requires a source of outside power, usually in the form of a storage battery. To assure prompt starting, a charging system for the battery is required in the form of either a charging generator driven by the engine or an ac-powered battery trickle charger. The latter is recommended for standby installations where an engine-driven charging generator functions only when the engine is operating. During idle periods, the battery will lose its charge unless maintained by a trickle charger. In recent years, there has been a trend toward the use of an ac generator, or alternator, which has the advantages of small size, higher voltage and amperage, competitive price, and good charging ability under idle speed conditions.

Another type of electric motor is a line voltage starter available in 110, 220, or 440V ac. It has the advantages of faster and more powerful cranking, the elimination of the battery and charging system, less maintenance, and sustained cranking through unlimited available electric power. Its disadvantages are a higher initial cost, the requirement of high line voltage at the site, the danger to personnel due to the high voltage, and the requirement to conform to existing wiring and installation codes.

AIR MOTOR The air motor, which is usually of the rotary-vane type, uses high-pressure air in the range of 50 to 150 lb/in^2 (340 to 1030 kPa) to turn it in starting the engine. It is mounted on the engine flywheel housing to mesh with the gear on the flywheel in the same manner as the electric motor. An outside source of air from an air compressor, usually with a 250-lb/in^2 (1720-kPa) capacity, is required. A pressure-reducing valve is installed in the line to the engine. The high-pressure air stored in an adequate receiver is sufficient for several starting cycles. This starting system has the advantages of faster cranking, sustained cranking as long as the air supply lasts, suitability in hazardous locations where an electric system might be dangerous, and ability to operate on either compressed air or high-pressure natural gas. Its disadvantages include a higher initial cost, the requirement of an air-compressor system, and, finally, a shutdown condition if the air supply is depleted before the engine starts.

HYDRAULIC MOTOR The hydraulic motor system consists of the motor, an oil reservoir, an accumulator, and some means of charging the accumulator. The accumulator, which is a simple cylinder with a piston, is charged on one side with nitrogen gas. As the hydraulic fluid, usually oil, is pumped into the other side, the gas is compressed to a very high pressure. When released, the fluid turns the motor, which in turn rotates the engine. The system can be charged by hand, with an engine-driven pump, or with an electric motor-driven pump. Generally, an engine-driven or electric-motor-driven pump is used in conjunction with the hand pump in case of an engine or electrical failure. This system has the same basic advantages of the air motor except that there is no prolonged starting. If the engine is in good operating condition, the cranking is fast and a start is instantaneous, but, if not, it is necessary to recharge the system before another start can be made.

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AUXILIARY ENGINE A small auxiliary air-cooled or water-cooled engine is sometimes employed for starting. It may be mounted on the engine in the same manner as the other systems, or a belt drive may be employed. Some form of speed reduction is required to reduce the higher speed of the auxiliary engine to that required for proper cranking. The principal advantage of such a system is a complete independence from outside sources of power, such as batteries, air, or pumps, but this is offset by a higher initial cost and the required regular maintenance of the engine.

IGNITION SYSTEMS

The internal combustion engine requires some means of igniting the combustible charge in the cylinder at the proper time. Today's high-compression gasoline and gas engines demand a system that will produce a high-tension spark across a short gap in the combustion chamber for the ignition of the charge. It is obvious that the design of the combustion chamber must be such that this combustible mixture of fuel and air is present between the discharge gap when the spark occurs; otherwise, ignition will not take place.

Ignition systems for gasoline or gas engines are considered high or low tension. The energy system for the high-tension system is either an electric generator and battery or a magneto. The generator and battery produce a direct current at 6 to 12 V potential, and the magneto produces an alternating current with higher peak voltages. With either energy source, this system has a primary circuit for the low-voltage current and a secondary circuit for the high-voltage current.

The primary circuit consists of the battery, an ammeter, an ignition switch, a primary coil, and breaker points and a condenser in the distributor. When the ignition switch and the breaker points close, a current flows through the circuit and builds up a magnetic field in the primary coil. Opening the breaker points breaks this circuit, causing the magnetic field to start collapsing. As the field collapses, it produces a current that flows in the same direction in the primary circuit and charges the condenser plates. The condenser builds up a potential opposing flow, which discharges back through the current. This results in a sudden collapse of the remaining magnetic field and the induction of a high voltage into the secondary winding of the coil. The breaker points are opened and closed by a cam which is engine-driven, usually at half engine speed.

The secondary circuit consists of the secondary coil winding, the lead to the distributor rotor, the distributor, the spark plug leads or wires, and the spark plug. A magneto eliminates the battery, but includes in its construction the balance of both primary and secondary circuits. It may have either a rotating coil and stationary permanent magnets or a stationary coil and rotating magnets. The relative movement of the primary coil winding and the magnets induces an alternating current in the primary circuit, the breaking of which induces a high-voltage current in the secondary circuit.

The development of the modern engine has required many refinements in spark plug design, but basically a spark plug consists of two electrodes, one grounded through the shell of the plug and the other insulated with porcelain or mica. The insulated electrode is exposed to the combustion. The heat flow occurs from this electrode to the spark plug shell through the grounded electrode.

Recent developments in ignition systems have produced the low-tension and breakerless systems. The breakerless system has eliminated most of the moving parts in the distributor system. The breaker points in the distributor system are actually a switch that opens and closes the primary circuit of an ignition coil. In the breakerless system, the use of solid-state devices provides a switch with no moving parts to wear or require adjustment.

The low-tension magneto system has been developed to reduce electrical stresses in the ignition circuit. The secondary coil has been removed from the magneto proper and relocated near each spark plug. The low voltage generated by the magneto is transmitted through the wiring harness to secondary coils, which then step up to the voltage to be transmitted through short leads to the spark plug. These leads may be insulated to withstand the stresses imposed upon them. This results in a minimum of electrical stresses with a resulting longer life of all components of the system.

Previous Page

As was mentioned previously under fuel systems, the diesel engine used the heat of compression for ignition; thus no auxiliary systems, such as the systems mentioned above, are required. The fuel is injected into the combustion chamber under relatively high pressure through the use of a fuel pump and injection nozzle. This system may be either an individual pump and nozzle for each cylinder, commonly called a *unit injection,* or a multicylinder pump that maintains a high pressure in a common fuel line connected to each injection nozzle. The latter is normally called the *common rail system.*

ENGINE INSTALLATIONS

Foundation The correct foundation, mounting, vibration isolation, and alignment are most important to the success of any engine installation. All stationary engines require a foundation or mounting base. There are many variations, but all basically serve to isolate the engine from the surrounding structures and absorb or inhibit vibrations. Such a base also provides a permanent and accurate surface upon which the engine and usually the pump may be mounted.

To meet these requirements, the foundation must be suitable in size and mass, rest on an adequate bearing surface, provide an accurately finished mounting surface, and be equipped with the necessary anchor bolts.

The size and mass of the foundation will depend upon the dimensions and weight of the engine and the pump (if a common base is considered). The following minimum standards should be followed:

1. Width should exceed the equipment width and length by a minimum of 1 ft (0.3 m).
2. The depth should be sufficient to provide a weight equal to 1.3 to 1.5 times the weight of the equipment. This depth may be determined by the following formula:

In USCS units
$$H = \frac{(1.3 - 1.5)W}{L \times B \times 135}$$

In SI units
$$H = \frac{(1.3 - 1.5)W}{L \times B \times 2162}$$

where H = depth of foundation, ft (m)
W = weight (mass) of equipment, lb (kg)
L = length of foundation, ft (m)
B = width of foundation, ft (m)
135 = density of concrete, lb/ft^3 (2162 kg/m^3)

The soil-bearing load in pounds per square foot (kilograms per square meter) should not exceed the building standard codes. It may be calculated by the formula

$$\text{Bearing load} = \frac{(2.3 - 2.5)W}{B \times L}$$

Foundation or anchor bolts used to hold the equipment in place should be of SAE grade No. 5 bolt material or equivalent. The diameter, of course, is determined by the mounting holes of the equipment. The length should be equivalent to a minimum embedded length of 30 times the diameter plus the necessary length for either a J or an L hook. An additional 5 to 6 in (13 to 15 cm) should be provided above the top surface of the foundation for grout, sole plate, chocks, shims, equipment base washers, and nuts, plus small variations in the surface level. Around the bolts, it is a good practice to place a sleeve of iron pipe or plastic tubing to allow some bending of the bolts to conform with the mounting hole locations. This sleeve should be about two-thirds the length of the

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bolt, with its top slightly above the top surface of the foundation to prevent concrete from spilling into the sleeve.

Sole plates running the length of the equipment are recommended for mounting directly to the foundation. Made of at least $\frac{3}{4}$-in (19-mm) hot- or cold-rolled steel and a width equivalent to the base-foot mounting of the equipment, they will provide a level means of mounting and will avoid variations in the level of the concrete. These plates should be drilled for the mounting holes and drilled and tapped for leveling screws, which will permit the plates to be leveled and held during the pouring of grout.

Alignment Although the alignment will vary with the type of engine and the pumping equipment, the basic objective remains the same. The driven shaft should be concentric with the driver shaft, and the centerlines of the two shafts should be parallel to each other. Rough alignment should be made through the use of chocks and shims. A dial indicator should be used to check deflection by loosening or tightening the anchor bolt nuts until there is less than a 0.005-in (0.13 mm) reading at each bolt. Shims should be added or removed to arrive at this point. A final check should be made with all the conditions "hot," as the engine and its driven equipment expand at the rate of 0.000006 in/°F (0.27 μm/°C) above ambient hot to cold. Although the coupling, or driving member between the engine and the pump is not discussed in this section, it must be considered in the final alignment.

Vibration Isolation It is desirable to isolate the engine, and at times the pumping equipment, from the building structure because of vibrations. Cork (Figure 13) is used in the larger and heavier installations. A combination of cork and rubber pads may be used at each mounting hole on small- or medium-size installations, and spring isolators may be used on a complete installation if flexible hoses are used for fuel, water, and air connections where required. The manufacturer of the engine and the pumping equipment should be consulted in the use of any isolating material or device.

Vibrations are closely associated with the driving and driven equipment, couplings, and other connections. These linear vibrations may be caused by improper supports of the unbalanced parts, which produce a *torsional* vibration. An understanding of this vibration is important because its elimination is the responsibility of the engine and the driven-equipment manufacturer. It is complex and cannot be detected without the use of calculations and special instruments.

The basic concept involves an elastic element, such as an engine crankshaft, which tends to twist when any firing impulses are applied. When these forces are removed, the

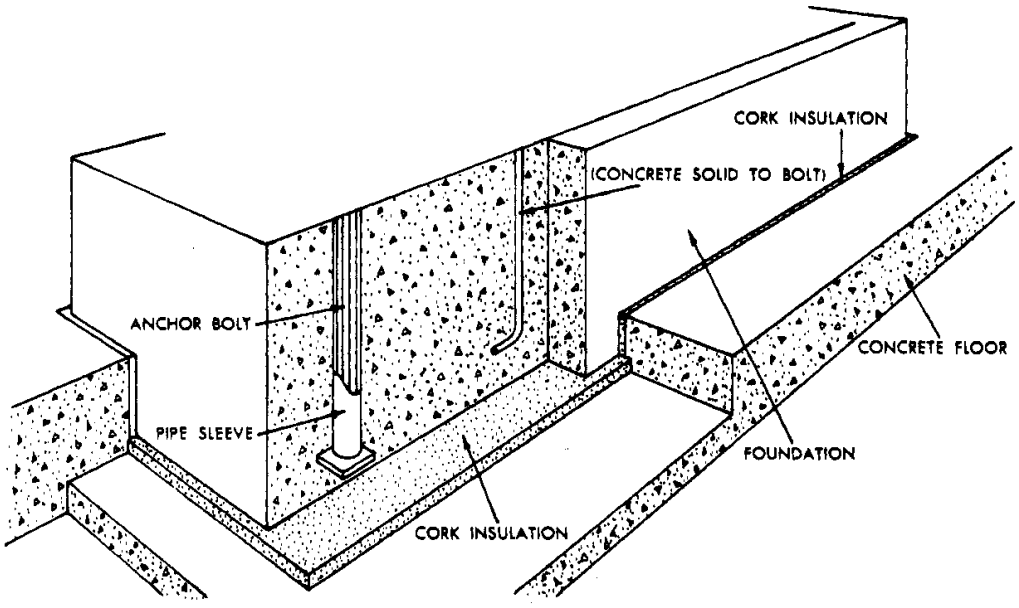


FIGURE 13 Engine foundation installation (Waukesha Motor)

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elastic body will try to return to its original position. The driven mass and the connecting elements tend to resist these external impulses. The natural elasticity of the crankshaft and its connecting system allows a small amount of torsional deflection and tends to reduce the deflection as the external impulses reduce in force. Other reciprocating forces in the engine and external forces in the driven equipment may excite vibrations in the entire system. When all these forces come into resonance with the natural frequency of the entire system, torsional vibration will occur. This vibration may or may not be serious but, being complex, cannot be solved hastily.

The engine and pump manufacturer designs and constructs the product so critical harmonic vibrations will not be present under normal speeds and loads. However, there is no way to control the combination. An analysis of the complete system should be made. This requires a study of the mass elastic system of the combination, involving the mass weight and radius of gyration of all rotating parts. This study should be made either by the engine or pump manufacturer or by a torsional-analysis specialist.

FURTHER READING

Gunther, F. J. "Gas Engine Power for Water and Wastewater Facilities." *Water & Sewerage Works,* **112** and **113** (November 1965 to July 1966).

Heywood, J. B. "Internal Combustion Engine Fundamentals." McGraw-Hill Book Company New York, 1988.

9.1.4
HYDRAULIC TURBINES

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PUMPING APPLICATIONS

In many pumping applications, the liquid remains at a high pressure after it has completed its cycle. It is often economically desirable to recover some of the energy, which will otherwise have to be dissipated when the liquid is brought back to a lower pressure. Instead of running the liquid through a pressure-reducing valve to destroy the energy, a hydraulic turbine may be installed. This turbine can therefore assist in driving the process pumps. There are many such applications in use. Among these are the use of turbine-driven pumps in gas-cleaning operations. Here the solutions from scrubber towers are passed through turbines that in turn drive pumps. Some of the solutions involved in such a process are water saturated with carbon dioxide at a temperature of 40 to 70°F (4 to 21°C) and potassium carbonate containing dissolved carbon dioxide at a specific gravity of 1.31. A liquid at the much lower specific gravity of 0.84 has been used in power-recovery turbines in a glycol-ethylene hydration process.

One of the oldest applications of turbine-driven pumps was for fire-fighting equipment in mills having a natural head of water. The large volume of low head water was routed through a hydraulic turbine that drove the pump to pressurize a sprinkler system or provide high-pressure water to the fire-hose connections. Today, hydraulic turbines are used to drive pumps that generate fire-fighting foam.

At thermal power plants, cooling water returning from the cooling towers has been used to drive a hydraulic turbine directly connected to a pump that provides part of the cooling water. Naturally, in such an application, the power from the turbine alone is not enough to maintain the pumping system. Therefore an auxiliary power source is also required.

Turbines using oil pressure have been employed at thermal power plants. Large steam turbines obtain bearing oil and control oil pressure from a main feed pump that is directly connected to the steam turbine shaft. A small amount of the oil at this high pressure is used for the relay control, with the majority of the oil going to the bearings at a lower pressure. To reduce the oil pressure from that required by the relays to that needed by the

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bearings, the oil is passed through a turbine driving a booster pump that pressurizes the oil at the intake of the main pump. At locations with high flood levels, it is preferable to locate pumps and electrical equipment above flood level. Some recent steam power plant installations have used the return from the condenser to the river (normally an appreciable drop in head) to drive a hydraulic turbine, which in turn may assist in driving the condensate pumps. Also, effluent from sewage treatment plants may have a significant discharge head.

It is often economical to use a small volume of water at a high head to move a large volume of water at a low head. Of course, the reverse application can also be accomplished. The former procedure is employed at hydroelectric projects where it is necessary to operate fishways to enable migrating fish to continue traveling upstream over the dam. The large volume of water is used both to attract the fish to the fish flume and to transport the fish to the fish ladders. An example of such an installation can be seen in Figure 1.

Turbines have also been used to start large pumping units when the hydraulic conditions are suitable. The impulse turbine, which develops maximum torque at zero speed, is especially useful for this application. The power required on such a starting turbine would be less than the power required on a starting motor to do the same job. Many electric controls can be eliminated when starting with a turbine. An illustration of this application is shown in Figure 2.

Desalinization plants have been built wherein salt is removed from seawater by pumping it through a membrane at high pressure. Only pure water goes through the membrane, with most of the sea water being used to carry away the salt. This excess high-pressure sea water is then put through a power-recovery turbine that helps to drive the high-pressure pumps.

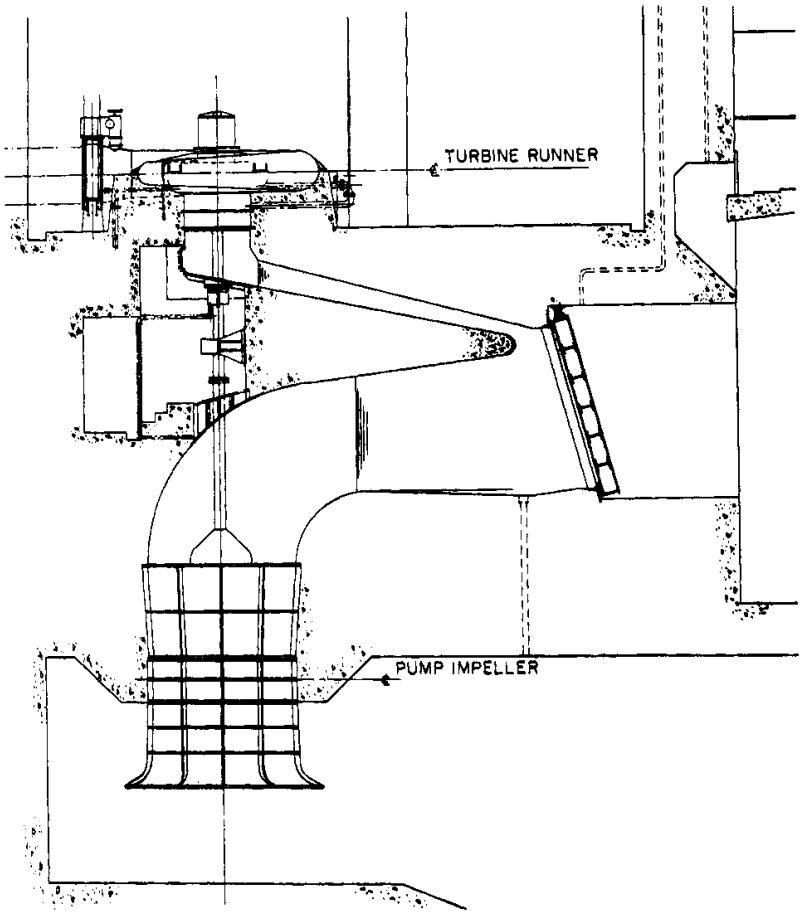


FIGURE 1 Turbine-driven fishway pumps. High head, low volume Francis-type turbine drives a low head, high-volume propeller pump at Rocky Reach, Washington.

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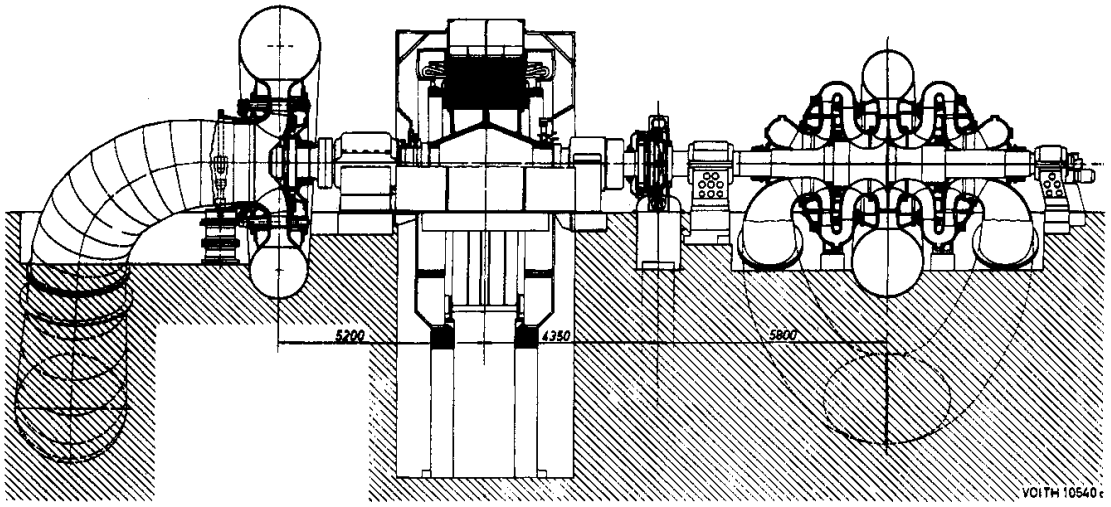


FIGURE 2 Impulse runner used to start a large pump. Left to right: Francis turbine, generator motor, starting impulse turbine, and two-stage double-suction storage pump. (Adapted from Voith Publication 1429, J. M. Voith GmbH, Heidenheim, Germany).

It is often desirable to have the turbine and pump running at different speeds, and this has been accomplished by means of a variable-speed coupling between turbine and pump. Reliable low-speed gear increasers or reducers of up to 35,000 hp (26,000 kW) are being built by many manufacturers. In applications such as starting a pump by means of a turbine, a mechanism may be needed to disengage the turbine after the pump has been started, in order to minimize windage and friction. Disengaging couplings may be hydraulic or mechanical and may be actuated when the unit is stationary or rotating, depending on the application. However, it is possible to allow the turbine to spin in air with the pump after the pump has been started. Figure 3 is an illustration of a turbine-drive pump arrangement.

TYPES OF TURBINES

Three types of turbines will be discussed: propeller (fixed and adjustable blades), Francis, and impulse. Figures 4 to 7 are illustrations of these turbine runners. Figure 8 shows a complete Francis turbine unit. The fixed- and adjustable-blade propellers and runners are essentially the same, with the exception that the adjustable is suited to a much wider range of loading conditions. This versatility is reflected in a higher manufacturing cost for the adjustable blade. The Francis runner is much like a centrifugal pump impeller running backward. The impulse runner (or Pelton wheel) is for high-head applications. The water is first channeled through a nozzle that directs a jet of water into the bowl-shaped runner buckets. This jet then discharges into the atmosphere. When a high back pressure is present in the housing, the impulse wheel cannot discharge properly. The performance drops off rapidly as the back pressure is increased. However, it is sometimes possible to admit low air pressure into the impulse runner housing to lower the level of the liquid surface to below the level of the runner. When the high turbine back pressure cannot be eliminated, a Francis turbine will perform much better than an impulse turbine. Figure 9 illustrates the relative efficiencies of the different types of runners. The fixed-blade propeller and Francis runners operate most efficiently in a range near full load, and operating time should be limited at low loads. The impulse and adjustable-blade runners, however, are designed for high efficiencies over a large load range. The adjustable-blade propeller accomplishes this by changing the angle of its blades by means of linkage in the hub of the runner. With the impulse runner, the size of the jet is controlled by the nozzle needle, which enables the runner to maintain high efficiencies at low loads.

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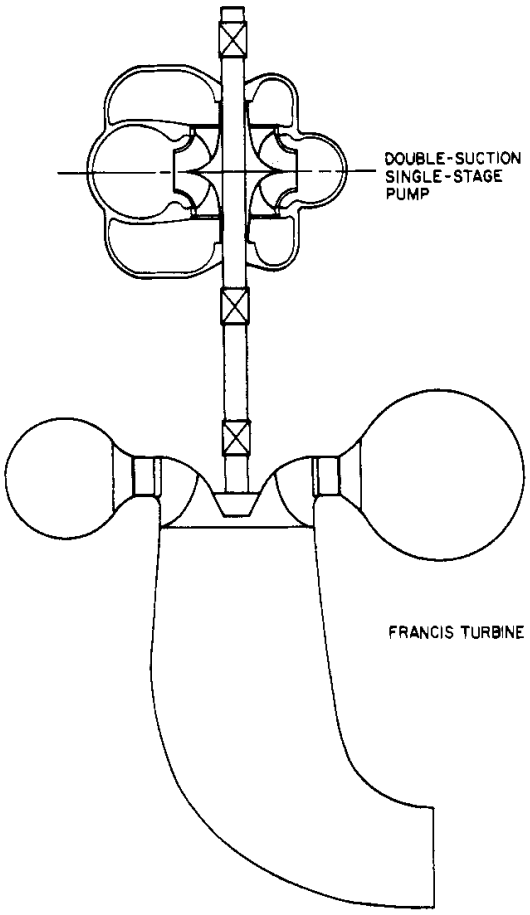


FIGURE 3 Double-suction single-stage pump driven by a Francis turbine (Voith Siemens Hydro)



FIGURE 4 Mixed flow propeller runner (Voith Siemens Hydro)

TURBINE ARRANGEMENTS

As is typical with pumps, hydraulic turbines are usually arranged with their power shafts either vertical or horizontal. In recent years, several low head installations have had inclined shafts to reduce excavation costs. The large (above 10,000 hp or 7,000 kW) vertical propeller turbines with concrete water passages have not been used with pumps. This is also true with very large, high head Francis turbines with steel-lined spiral shaped water passages. Mid-size and small hydraulic turbines of all three types may have vertical or horizontal shafts and usually steel water passages when used with pumps. Horizontal shaft arrangements have the advantage of better access to each piece of equipment. They usually require a greater floor area but lower power/pump house. For low heads (under 30' or 10 meters), propeller turbine installations have used inclined shafts with Bulb and Pit generator housings. TUBE turbines with a nearly straight draft tube have also been arranged with inclined shafts. For very low heads (under 15' or 5 meters), water wheels extending the width of a spillway are being considered where conventional hydraulic turbines are too costly.

Each project or site is likely to have special conditions, which will influence selection of the optimum turbine and pump arrangement.

Flow Units The size of hydraulic turbines has become so large that the earlier conventional flow quantity of cubic feet/minute (cfm) is today cubic feet/second (cfs) or cubic meters per second (cms). The following conversions are therefore useful:

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FIGURE 5 Adjustable-blade propeller runner. "Kaplan" turbines have adjustable blades coordinated with adjustable wicket gates. (Voith Siemens Hydro)

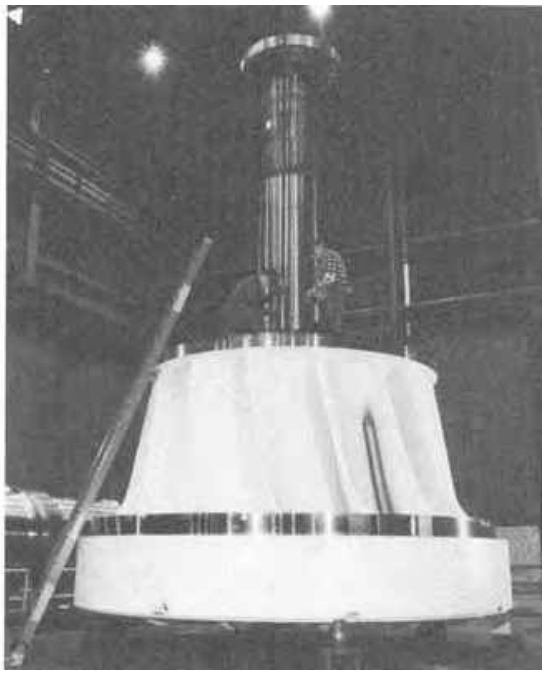


FIGURE 6 Francis runner has shorter buckets than a centrifugal pump impeller (Voith Siemens Hydro)

1 cfs = 449 gpm or 1.077 mgd
1 cubic foot = 7.48 US gallons
British imperial gallon = 1.2009 US gallon
1 cms = 35.3 cfs

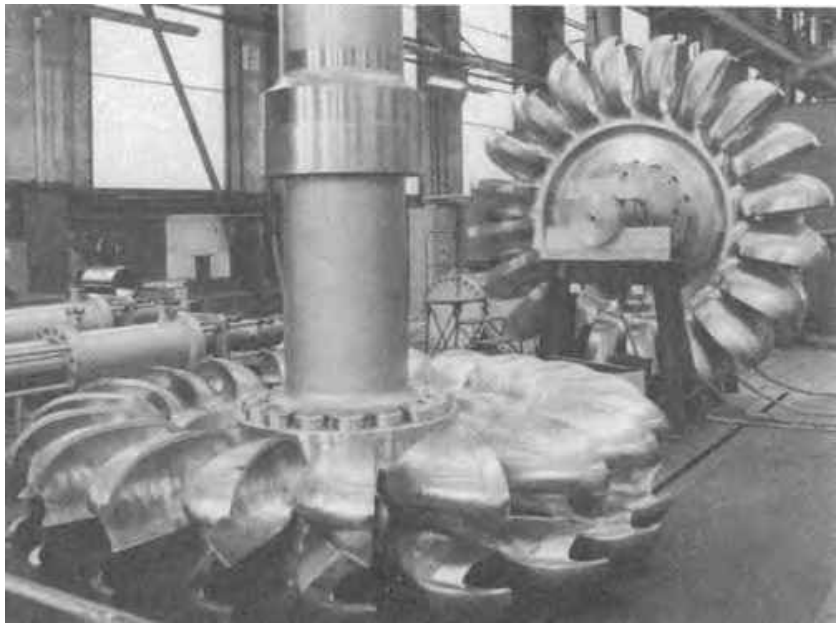


FIGURE 7 Impulse (or Pelton) runners (Voith Siemens Hydro)

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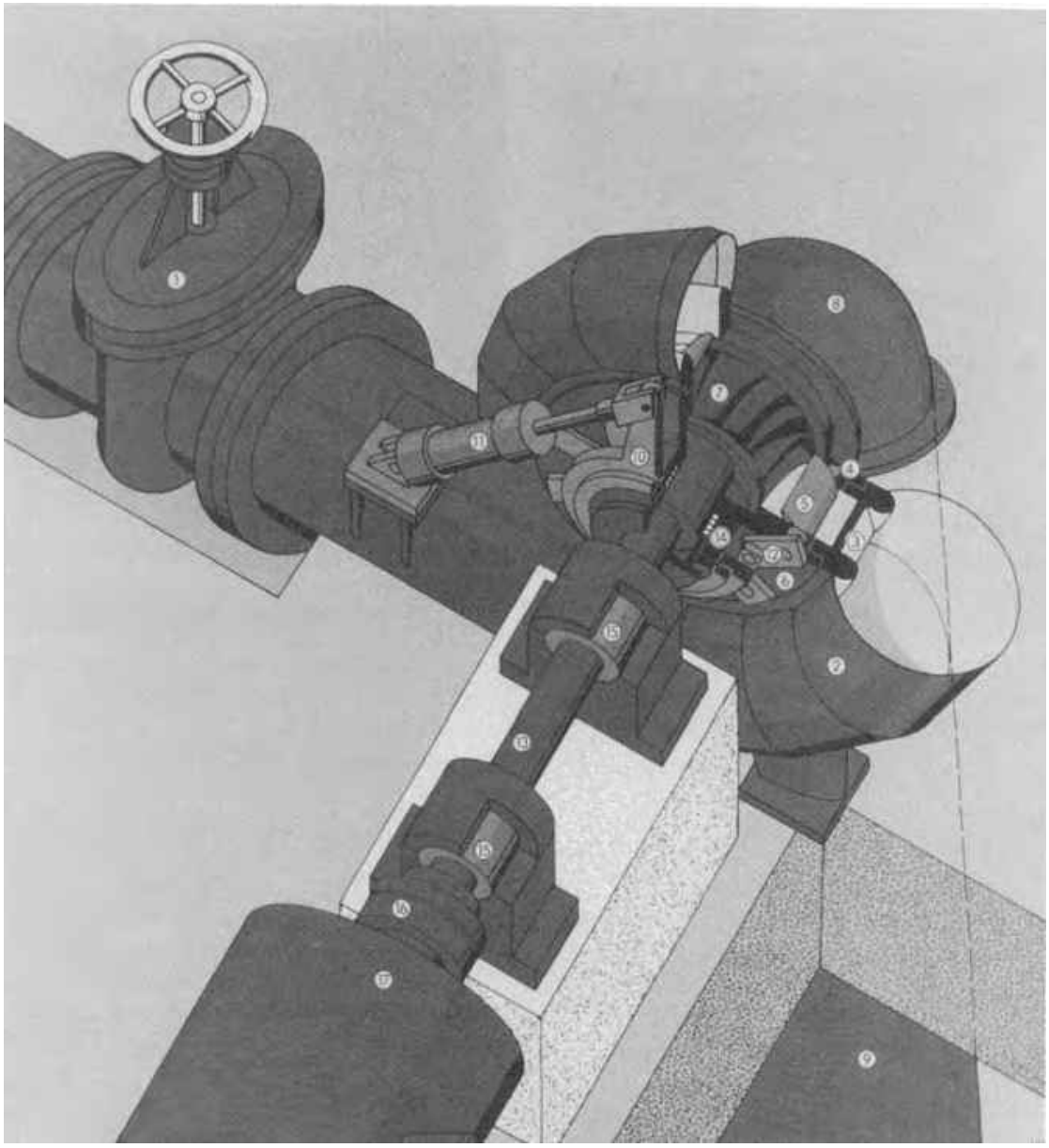


FIGURE 8 Complete standard Francis turbine unit with spiral case and elbow draft tube: 1) Shut-off valve, 2) Spiral case, 3) Stay ring, 4) Curb ring, 5) Guide vane, 6) Turbine cover, 7) Runner, 8) Draft tube elbow, 9) Draft tube, 10) Operating ring, 11) Servomotor, 12) Lever, 13) Turbine shaft, 14) Shaft seal, 15) Bearing, 16) Flexible coupling, 17) Generator (Voith Siemens Hydro)

Specific speed This is the speed at which a runner will rotate if the runner diameter is such that, under 1 ft (1 m) net head, it will develop 1 hp (1 kW):

In USCS units
$$N_s = \frac{\text{rpm} \times \text{hp}^{1/2}}{\text{ft}^{5/4}}$$

In SI units
$$N_s = \frac{\text{rpm} \times \text{kW}^{1/2}}{\text{m}^{5/4}}$$

The specific speed is an important factor governing the selection of the type of runner best suited for a given operating range. The impulse wheels have very low specific speeds

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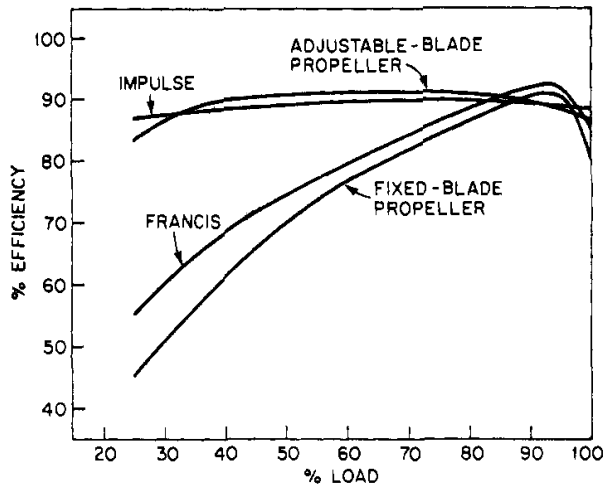


FIGURE 9 Efficiency versus load for different runner types

relative to propellers, and the specific speed of a Francis turbine lies between the impulse and propeller. Table 1 shows some statistics for existing units.*

SIZES AND RATINGS

Although power-recovery turbines are relatively small, there are much larger turbines in existence that could be used to drive pumps. Turbines of the following sizes do exist: propeller, 405.5-in (10.3-dm) diameter; Francis, 288-in (7.3-m) diameter; and impulse, 176-in (4.5-m) diameter. The extreme range of power, head, and speed is shown in Table 2.

GENERAL CHARACTERISTICS

Sigma As in pumps, the problem of cavitation also exists in turbines. Sigma is defined as follows:

$$\sigma = \frac{H_a - H_s}{H}$$

where H_a = feet (meters) of atmospheric pressure minus vapor pressure

H_s = distance in feet (meters) the centerline of the blades is above tailwater for vertical units or distance in feet (meters) the highest point of the blade is above tailwater for horizontal units

H = feet (meters) of elevation between inlet head water surface and tailwater surface elevation

Critical sigma is defined as that point at which cavitation begins to affect the performance of the turbine. Figure 10 shows the relationship between critical sigma and specific speed and also shows which type of runner is best for a given specific speed. The impulse

*Universal specific speed Ω_s (defined in Chap. 1 and Section 2.1.1) is found from these turbine N_s-values as follows:

$$\Omega_s = \frac{N_s(USCS)}{4.344\sqrt{\eta_t \times \text{sp. gr.}}} = \frac{N_s(SI)}{16.564\sqrt{\eta_t \times \text{sp. gr.}}}$$

where η_t is the turbine efficiency in percent.

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TABLE 1 Turbine statistics of existing units

Speed, rpm	Power, hp[a]	Head, ft[b]	N_s, USCS[c]	Type[d]	Use[e]	Supplier[f]
450	2,000	925	3.94	I	SUP	BLH
450	450	485	4.19	I	SUP	BLH
750	170	575	3.47	I	SUP	BLH
340	570	485	3.57	I	SUP	BLH
1,775	264	1,085	4.63	I	SUP	BLH
437.5	425	405	4.97	I	SUP	BLH
1,770	640	960	8.38	I	SUP	BLH
1,775	230	1,085	4.32	I	SUP	BLH
720	885	460	10.05	I	SUP	BLH
900	640	866	4.85	I	SUP	BLH
3,550	85	1,920	2.58	I	SUP	BLH
690	535	86.5	60.50	F	FW	BLH
126	670	80	13.63	F	FW	BLH
450	2,600	118	58.90	F	SUP	LEF
525	1,050	83.5	67.39	F	SUP	LEF
900	750	123	60.18	F	SUP	LEF
1,000	375	96	64.43	F	SUP	LEF
700	53	150	9.71	I	SUP	LEF
750	160	30	135.13	P	SUP	LEF
882	128	70	49.26	F	SUP	LEF
1,750	145	135	45.78	F	SUP	LEF
1,750	220	135	56.39	F	SUP	LEF
280	200	14	146.20	P	SUP	A-C
550	400	35	129.21	P	IR	A-C
3,450	7.5	196	12.89	I	PP	A-C
1,775	353	1,085	5.36	I	ST	A-C
1,185	280	231	22.01	I	ST	A-C
1,775	300	1,510	3.27	I	CH	A-C
1,300	1.6	150	3.13	I	PP	A-C
700	1,760	798	6.92	I	ST	A-C
600	392	460	5.58	I	SUP	A-C
122	973	65	20.61	F	FW	A-C
108	1,200	75	16.95	F	FW	A-C

[a]kW = 0.746 hp

[b]m = 0.3048 ft

[c]N_s (SI) = 3.814 N_s (USCS)

[d]I = impulse, P = propeller, F = Francis

[e]SUP = supplement electric motor to drive pump, FW = drives fishway pumps, ST = scrubber-tower application, PP = turbine drives petroleum pumps, CH = power recovery in chemical plant, IR = irrigation project.

[f]BLH = Baldwin-Lima-Hamilton Corp., LEF = The James Leffel Co., A-C = Allis-Chalmers Corporation, since 1986 Voith Hydro, Inc., also American Hydro Corp.

wheel is not affected by sigma because it is a free jet action and therefore not subject to low-pressure areas.

Affinity Laws The relationships between head, discharge, speed, power, and diameter can be seen in the following equations, where Q = rate of discharge, H = head, N = speed, P = power, D = diameter, and subscripts denote two geometrically similar units with the same specific speed:

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TABLE 2 Range of power, head, discharge, and speed of existing units of one manufacturer

	Propeller		Francis		Impulse	
	Low	High	Low	High	Low	High
Power hp[a]	82.5	268,000	1.2	820,000	1.6	330,000
Head ft[b]	6.0	180	4.0	2,204	75.0	5,790
Speed, rpm	50	750	56.4	3,500	180	3,600

[a]kW = 0.746 hp
[b]m = 0.3048 ft

$$\frac{Q_1}{N_1 D_1^3} = \frac{Q_2}{N_2 D_2^3}$$

$$\frac{Q_1^2}{H_1 D_1^4} = \frac{Q_2^2}{H_2 D_2^4}$$

$$\frac{N_1^2 D_1^2}{H_1} = \frac{N_2^2 D_2^2}{H_2}$$

$$\frac{P_1}{N_1^3 D_1^5} = \frac{P_2}{N_2^3 D_2^5}$$

Most designs used are tested as exact homologous models, and performance is stepped up from the model by the normal affinity laws given above. Because of difficulties in measuring large flows at the field installation, only approximate or relative flow metering is normally done.

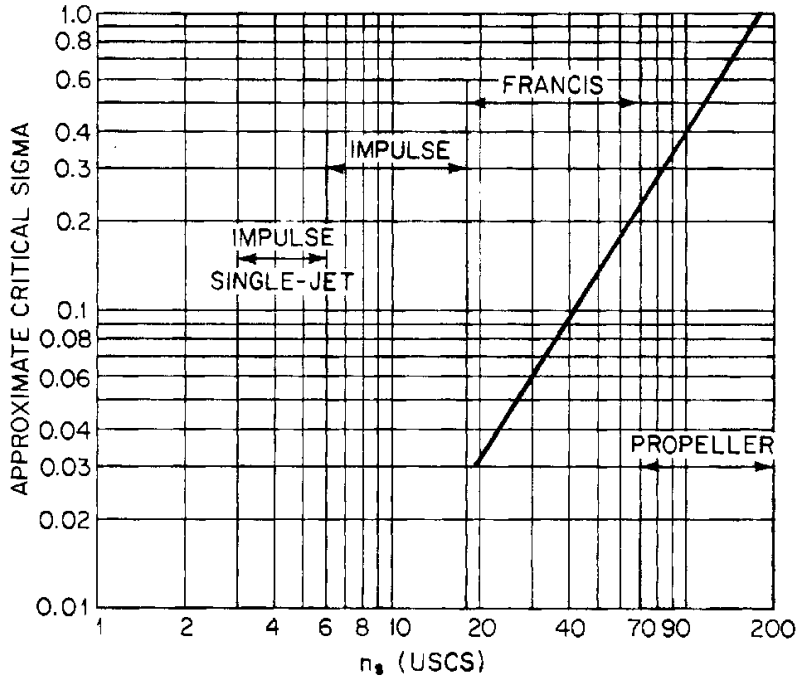


FIGURE 10 Critical sigma versus N_s. Specific speed N_s (SI) = 3.814 N_s (USCS)

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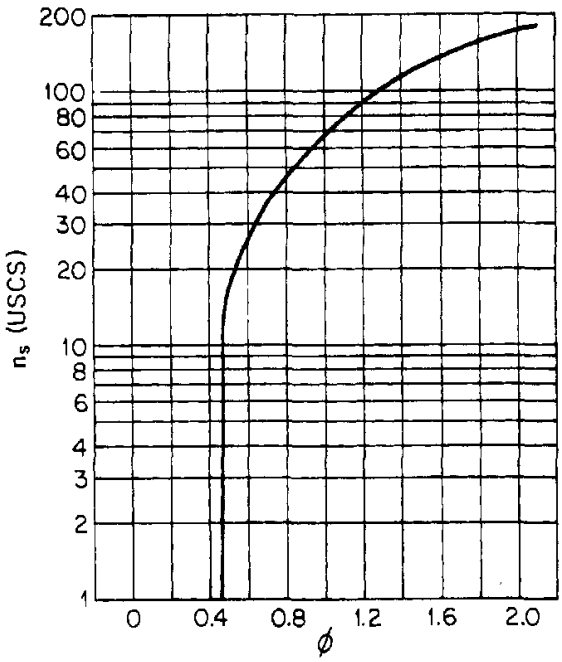


FIGURE 11 Specific speed versus ϕ. Specific speed N_s (SI) = $3.814N_s$ (USCS)

Speed Characteristic This is defined as the peripheral speed of the runner divided by the spouting or free discharge velocity of the water:

$$\phi = \frac{u}{(2gH)^{1/2}}$$

where ϕ = speed characteristic
u = peripheral speed of runner, ft/s (m/s)
H = head, ft (m)
g = 32.2 ft/s^2 (9.807 m/s^2)

A plot of ϕ versus speed can be seen in Figure 11, which shows that ϕ becomes constant at after the specific speed has dropped into the impulse-runner region. Theoretically, for maximum energy conversion, ϕ should equal 0.5 for impulse runners. However, because of small losses in the runner, this value is set at approximately 0.46. Control of the hydraulic turbine can be accomplished by means of a governor or a flow or load control system.

Torque All hydraulic turbines have maximum torque at zero speed; therefore, they have ideal starting characteristics. The pump can be accelerated to design speed by gradually opening the turbine gates or inlet valve while keeping within the limitations of hydraulic transients.

DATA REQUIRED FOR TURBINE SELECTION

The turbine supplier should have the following information in order to select the best combination of size and type runner:

1. Head available, head range, and head duration.
2. Power and speed required to drive pump.

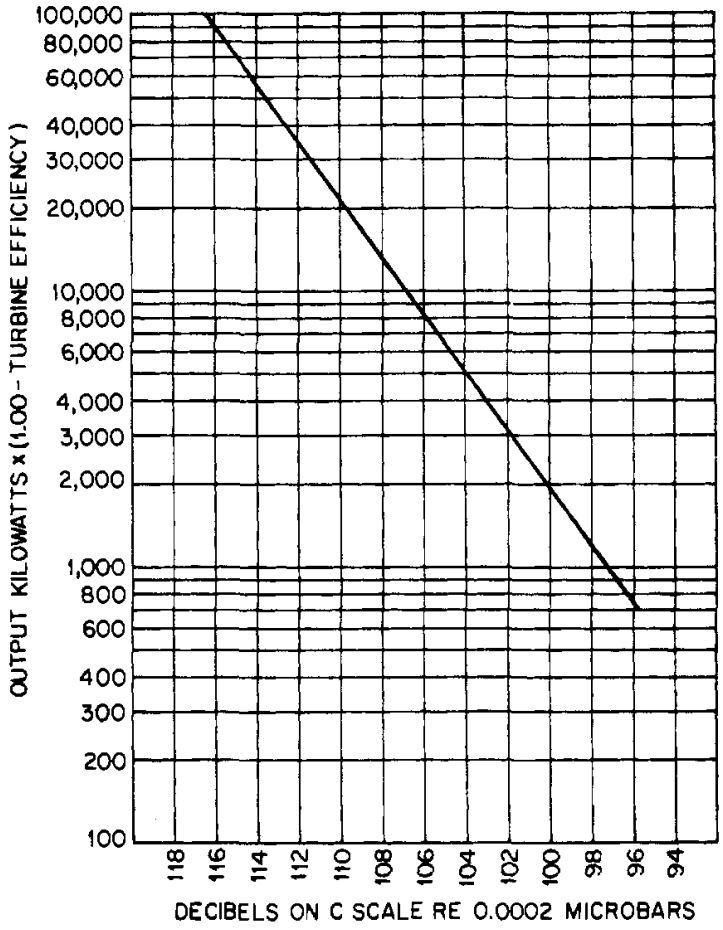


FIGURE 12 Sound level readings taken 3 ft (0.9 m) from main shaft. Hydraulic losses are approximated output in kilowatts × (1.00 − turbine efficiency). (From L. F. Henry: "Selection of Reversible Pump/Turbine Specific Speeds." Paper presented at meeting on pumped-storage development and its environmental effects, University of Wisconsin, September 1971.)

3. Description of fluid to be handled, including chemical composition and specific gravity.
4. Possibility of adding air to system. Turbine will operate satisfactorily without air, but air may be added to system to reduce pressure fluctuations (normally one-third of rpm in frequency) at part load and possibly to smooth unit operation at full load.
5. If corrosive fluid is to be handled, description of the materials required.
6. Controls from pumping process that will affect turbine operation. If speed is a controlling factor, the possibility of using a speed-varying device should be considered.
7. Back pressure on the turbine.
8. Noise-level trends for the turbine. Figure 12 illustrates noise levels recorded on some rather large turbine units. Ear protection is required for noise levels above 85 db.

FURTHER READING

ASME Hydro Power Technical Committee. "The Guide to Hydropower Mechanical Design." HCI Publications, Inc., 1996.

Daugherty, R. U, and Ingersoll, A. C. *Fluid Mechanics.* McGraw-Hill, New York, 1954.

Henry, L. F. "Selection of Reversible Pump/Turbine Specific Speeds." Paper presented at meeting on pumped-storage development and its environmental effects, University of Wisconsin, September 1971.

Roth, H. H., and Armbruster, T. F. "Starting Large Pumping Units." *Allis-Chalmers Eng. Rev.* 31 (3)(1966).

Schlichting, H. *Boundary Layer Theory.* McGraw-Hill, New York, 1960.

Stepanoff, A. J. *Centrifugal and Axial Flow Pumps, 2nd edition.* Krieger Publishing, Malabar, Florida, 1957.

Warnick, C. C. "Hydropower Engineering." Prentice Hall, New Jersey, 1984.

Wilson, P. N. "Turbines for Unusual Duties." *Water Power.* June 1971.

9.1.5
GAS TURBINES

RICHARD G. OLSON

THERMODYNAMIC PRINCIPLE AND CLASSIFICATIONS

The gas turbine is an internal combustion engine involving steady flow processes carried out in separate components. The basic processes involve compression of air in a compressor, introduction of the compressed air and fuel into the combustion chamber(s), and finally expansion of the gaseous combustion products in a power turbine. Figure 1 illustrates a simple gas turbine.

The Brayton, or Joule, Cycle Figure 2 shows an ideal Brayton, or Joule, cycle illustrated in *PV* and *TS* diagrams. This cycle is commonly used in the analysis of gas turbines.

The inlet air is compressed isentropically from point 1 to point 2, heat is added at an assumed constant pressure from point 2 to point 3, the air is then expanded isentropically in the power turbine from point 3 to point 4, and finally heat is rejected at an assumed constant pressure from point 4 to point 1. From Figure 2, the compressor work is $h_2 - h_1$, the turbine work is $h_3 - h_4$, and the difference is the net work output. The heat input is $h_3 - h_2$. One can then derive an expression for thermal efficiency as follows:

$$\eta = \frac{(h_3 - h_4) - (h_2 - h_1)}{h_3 - h_2}$$

In reality, the cycle is irreversible and the efficiency of the compression, combustion, and expansion must be taken into account. However, an examination of the thermal efficiency equation shows the need of high compressor and turbine efficiencies in order to produce an acceptable amount of work output. The importance of h_3 is also readily apparent. In noting that h_3 is directly proportional to temperature, one can appreciate why continuous

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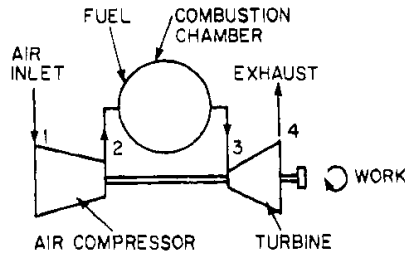


FIGURE 1 Components of simple gas turbine

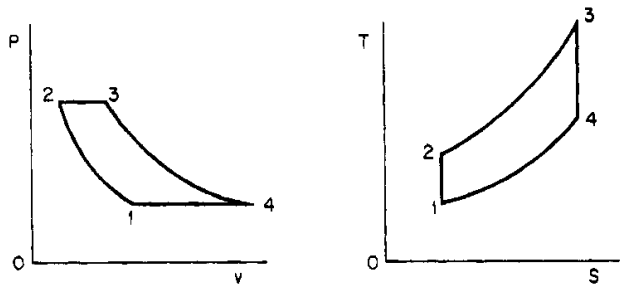


FIGURE 2 PV and TS diagrams for an ideal Brayton, or Joule, cycle

emphasis is being placed on the development of materials and techniques that permit higher combustion temperatures and correspondingly higher turbine inlet temperatures.

Classifications In an *open-cycle* gas turbine, the inlet air mixes directly with the combustion products and is exhausted to the atmosphere after passing through the power turbine. The *closed-cycle* gas turbine uses a heat exchanger to transfer heat to the working fluid, which is continuously recirculated in a closed loop. A *combined cycle* uses the principles of both the open and closed cycles. In current practice, the combined cycle uses an open cycle to provide shaft work while the heat from the exhaust is partially recovered in a waste-heat boiler. The heat recovered then proceeds through a standard steam power cycle until heat is rejected to the most readily available low-temperature reservoir.

Of the above configurations, the open-cycle gas turbine is most extensively used today for driving centrifugal pumps. This is probably due to the important consideration of minimum capital investment for each power output.

Two distinct types of open-cycle gas turbines have evolved: the *single-shaft* and *split-shaft* versions. The single-shaft gas turbine was developed primarily for the electric power industry and uses a compressor and a power turbine integrated on a common shaft. As the unit is used continuously at a single rotational speed, the compressor and power turbine efficiency can be optimized.

The split-shaft gas turbine was developed primarily for mechanical drive applications where output power and speed might be expected to vary. Figure 3 illustrates such a turbine. A typical curve of power output versus shaft speed is illustrated in Figure 4.

Split-shaft gas turbines are available in *conventional* and *aircraft-derivative* versions. The conventional gas turbine evolved from steam turbine technology and is illustrated in Figure 5. Figure 6 shows a modified jet engine used as a source of hot gas to a power turbine.

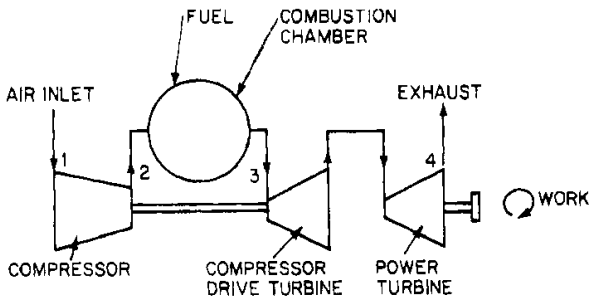


FIGURE 3 Components of a split-shaft gas turbine

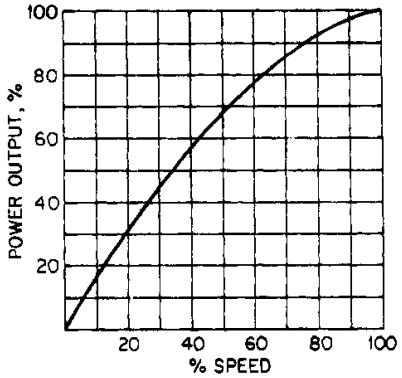


FIGURE 4 Typical curve of power output versus shaft speed for a split-shaft gas turbine

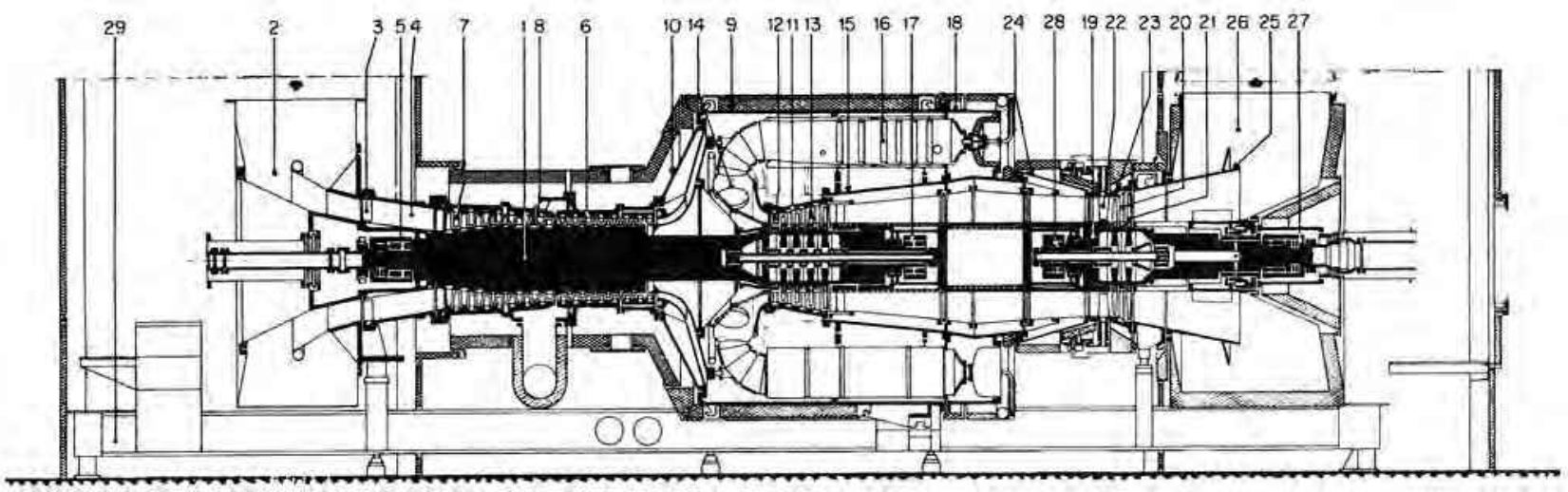


FIGURE 5 Gas turbine cross section: (1) gas generator turbine rotor, (2) inlet air casing, (3) compressor inlet guide vanes, (4) conical inlet casing with bearing support, (5) compressor radial-axial bearing, (6) compressor stator, (7) compressor stator blades, (8) compressor rotor blades, (9) combustion chamber main casing, (10) intermediate conical casing with diffuser, (11) gas generator turbine stator blade carrier, (12) gas generator turbine stator blades, (13) gas generator turbine rotor blades, (14) hot gas casing, (15) intermediate casing, (16) nine incorporated combustion chambers, (17) compressor radial bearing, (18) central casing, (19) power turbine rotor, (20) power turbine stator blades, (21) power turbine rotor blades, (22) power turbine adjustable rotor blades, (23) power turbine stator blade carrier, (24) conical intermediate piece, (25) outlet diffuser, (26) outlet casing, (27) power turbine radial-axial bearing, (28) power turbine radial bearing, (29) metallic foundation (Turbodyne)

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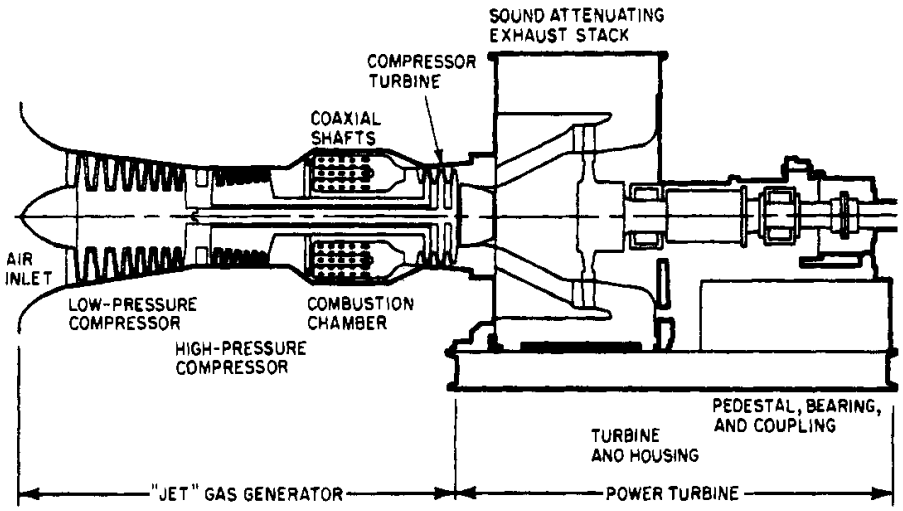


FIGURE 6 Cross section of aircraft-derivative gas turbine driver (Turbodyne)

Note that the jet engine combines the compression, combustion, and power turbine necessary to drive the compressor.

RATINGS

In the evolution of the gas turbine as a prime mover, various organizations have put forth standard conditions of inlet temperature and elevation to allow direct comparison of various gas turbines.

Four common standards exist:

ISO (International Standards Organization): sea level and 59°F (15°C)

NEMA (National Electrical Manufacturers Association): 1000 ft (304.8 m) above sea level and 80°F (27°C)

CIMAC (Congres International des Machines a Combustion): sea level and 59°F (15°C)

Site: actual elevation and design temperature at installation site

With the evolution of higher combustion temperatures and with the greater need for power over relatively short daily periods, new ratings have developed: emergency (maximum intermittent), peaking (intermittent), and base load. These classifications are based on the number of hours per unit of time that a gas turbine is operated and are related to the material used in the power turbine blading. A common standard is to use materials suitable for 100,000 h of continuous operation. Higher temperatures are permitted, but at the sacrifice of the life of the material and an increase in maintenance costs.

In a pump-driving application, the cycle of operations should be considered in specifying a gas turbine driver. A typical curve of gas turbine output as a function of inlet temperature (Figure 7) clearly indicates the necessity of specifying an accurate design temperature. Figure 8 is a typical correction curve for altitude.

FUELS

A wide range of fuels, from natural gas to the bunker oils, may be burned in simple-cycle gas turbines. In most cases, units can operate on gas or liquid fuels, and some turbines have automatic switchover capability while under load.

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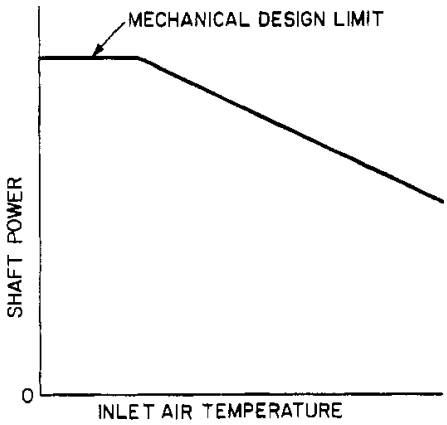


FIGURE 7 Typical curve of gas turbine output versus inlet temperature

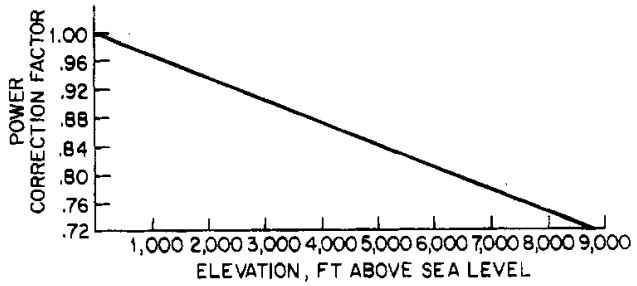


FIGURE 8 Typical curve of gas turbine power correction versus altitude (ft × 0.3048 = m)

Although it would appear advantageous to choose the cheapest fuel available, there may be disadvantages in such a choice: impurities such as vanadium, sulfur, and sodium definitely result in high-temperature corrosion; solid impurities and a high ash content will lead to erosion problems. One is therefore often faced with a trade-off between the cost of treating a fuel and the higher original cost.

With ever-increasing firing temperatures, research is continuing in the field of high-temperature corrosion with or without accompanying erosion problems. Present approaches to the problem include the use of inhibitors in the fuel as well as coatings for power turbine blades. Another solution that has been used on some crude oil pipelines is the topping unit. This is a small still that removes a specific fraction of the crude oil for burning as gas turbine fuel. It is good practice to include a detailed fuel analysis as part of any request for bids.

ENVIRONMENTAL CONSIDERATIONS

Legislation at both the federal and state levels has been concerned with the protection of our environment. An engineer must now consider the effects of the project on the environment.

Noise Any piece of dynamic mechanical equipment will emit airborne sounds that, depending on frequency and level, may be classified as noise.

The primary sources of noise in the gas turbine are the turbine inlet and exhaust and the accessories and support systems, such as fin fan lubrication oil coolers, auxiliary air blowers and fans, starting devices, and auxiliary lubrication oil pumps. It is common practice for manufacturers to provide inlet and exhaust silencers as well as some form of acoustic treatment for auxiliaries.

In considering the degree of acoustic treatment necessary for each installation, the practicing engineer must consider the following parameters:

1. Federal law (Walsh-Healy Act) limiting the time a worker may spend in a noisy environment
2. Local codes and their interpretation
3. Plant location and existing noise level at site
4. Site topography, including any noise-reflective surfaces
5. Applicable ASME standards and other relevant standards

Emissions The federal air quality acts require each state to develop a plan for achieving satisfactory air quality. Specifically, goals have been put forth to limit suspended particulate temperature matter and oxides of sulfur and nitrogen. Therefore,

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the engineer must investigate existing regulations during the planning stage of a pump installation.

In general, gas turbines have low particulate emissions. The amount of sulfur oxides exhausted to the atmosphere is in direct proportion to the content of sulfur in the fuel, and current practice calls for elimination at the source. The formation of oxides of nitrogen is a direct result of combustion temperatures. Dry low-NOx emission combustion exist based on lean combustion in pre-mixed burners.

GAS TURBINE SUPPORT SYSTEMS

Starting Systems A form of mechanical cranking is necessary to bring a gas turbine up to its self-sustaining speed. The amount of energy necessary will depend on each manufacturer's design. Available systems include electric motors, diesel engines, and gas-expander turbines.

Lubrication The gas turbine manufacturer normally provides a combined pump-turbine lubrication oil system. A main lubrication oil pump of sufficient capacity for the combined system is necessary, in addition to a standby pump in the event of failure of the main pump. A reservoir sized for a retention time of at least 4 minutes is usually specified. Filtration to 10-μm particle size should be adequate for most gas turbines and pumps.

Lubrication oil can be cooled by various means, the selection of which depends upon local conditions. The simplest and cheapest method uses a shell-and-tube heat exchanger with water as the cooling medium. In arid regions, a fin fan cooler with direct air-to-water cooling is commonly used.

Inlet Air Filtration The degree of inlet air filtration needed is primarily a function of the size and number of particles in the atmosphere surrounding the installation. In most cases, a simple tortuous-path precipitator will suffice. For dirtier atmospheres or in arid regions where sandstorms occur, inertial separators followed by a rolling-media filter should be considered.

Any filtration will result in a loss in performance because of the pressure drop across the inlet filter. Conservative design practice calls for a face velocity of 500 ft/min (150 m/min) for a rolling-media filter and 1700 ft/min (520 m/min) for a tortuous-path precipitator.

Control Most manufacturers offer control packages which provide proper sequencing for automatic start-up, operation, and shutdown. During automatic start-up, the sequencer receives signals from various transmitters to ensure that auxiliaries are functioning properly and, with the aid of timers, brings the unit on line through a planned sequence of events.

Key operating parameters—for example, output speed, gas turbine compressor speed, turbine inlet temperature, lubrication oil temperature and pressure—are continually monitored during normal operation. Signals from a pump discharge-pressure transmitter can be fed into a speed and fuel controller to cause the unit to respond to changes in speed and output.

As with start-up, normal and emergency shutdowns are accomplished through the sequencer. Most standard control packages are easily adaptable to remote control by means of cable or microwave.

APPLICATION TO PUMPS

Gas turbines are available to drive centrifugal pumps in a wide range of speeds and sizes, from 40 bp (30 kW) to over 20,000 hp (15,000 kW). It is not practical to list here all the

available units because they are too numerous and are continually being upgraded and added to. A listing can be found in *Sawyer's Gas Turbine Catalog,* which is published annually.

Pipeline Service Crude oil pipeline service has seen a tremendous growth in the use of gas turbine drivers since the mid-1950s, and this trend will continue as crude oil production becomes more and more remote from its markets. The advantages of the gas turbine for this application are as follows:

1. Installed cost is usually lower than that of a corresponding reciprocating engine.
2. Variable-speed operation allows maintenance of a specific discharge pressure under a wide range of operating conditions, thus achieving maximum flexibility.
3. Normal gas turbine control system is easily adapted to unattended operation and remote control.
4. Operating experience has proved the gas turbine to have a high degree of reliability.
5. Gas turbines can be packaged into modules for ease of transportation and erection.

Water Flooding An important aspect of oil production is the use of secondary recovery methods to increase the output of crude oil reservoirs whose pressure does not allow the crude to flow freely to the surface. Flooding the reservoir with high-pressure water has been a primary technique for years. The development of high-pressure centrifugal pumps has allowed increased water flow into oil fields.

As most oil production is located in remote areas, the packaged gas turbine driver has become increasingly popular. Figure 9 shows a typical 3300-bhp (2460-brake kW) split-shaft gas-turbine-driven centrifugal water flood pump package. A cutaway view of a typical 1100-hhp (820-brake kW) gas turbine is shown in Figure 10; note the reduction gearing at the exhaust end for direct driving of a pump. Supporting systems, such as starter, lubrication oil pumps, governor, and fuel oil pumps, are driven off the accessory pods located at the air inlet end. Offshore platform installations require drivers with a minimum vibration as well as small, unbalanced inertial forces. The gas turbine fits both of these descriptions very well.

Cargo Loading Another interesting application of this prime mover is in the field of cargo loading, where units are currently in operation charging tankers with crude oil. Selection of a gas turbine pumping unit with critical speeds above the normal operating range allows great flexibility of operation, particularly during final topping operations.

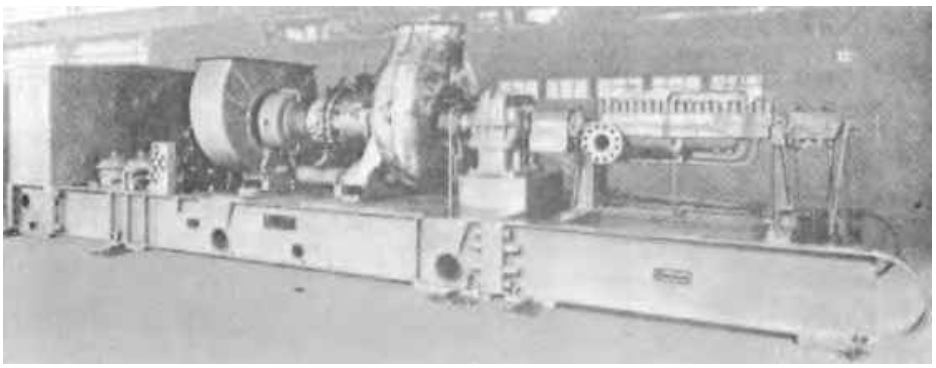


FIGURE 9 A 3300-bhp (2460-bkW) gas turbine-driven waterflood package (Solar Division of International Harvester)

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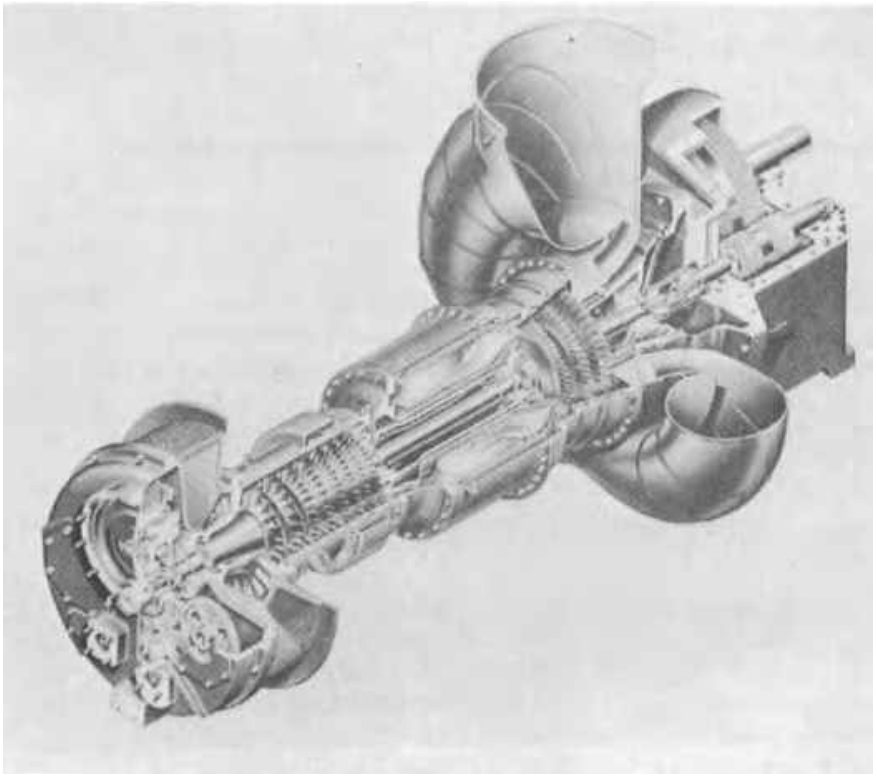


FIGURE 10 Cutaway view of 1100-bbp (820-bkW) split-shaft gas turbine driver (Solar Division of International Harvester)

Application Considerations The application of gas turbine drivers can vary from a simple driver for one pump operating at constant flow and discharge pressure to a multiplicity of units operating at variable speed on a pipeline. For the purposes of this discussion, it is assumed that the pumping system has been analyzed, a pump selection has been made, and all possible operating conditions have been analyzed so brake horsepower (brake kilowatt) and speed requirements are known.

The brake horsepower (brake kilowatt) output of the gas turbine must equal or exceed that required by the pump. This output can be determined by the use of specific performance curves similar to Figure 7 as corrected for elevation (Figure 8). Gear losses as necessary are added to the brake horsepower (brake kilowatt) required by the pump. Intermediate brake horsepower (brake kilowatt) and speed requirements should then be checked against a gas turbine output versus speed curve (Figure 4).

A torsional analysis of the combined unit is made (usually by the gas turbine manufacturer) to ensure the absence of any critical speeds in the operating range. Table 1 is a typical data sheet recommended for use when purchasing a gas turbine to drive a centrifugal pump.

TABLE 1 Typical data sheet for gas turbine driven centrifugal pump

Information from purchaser

Pumping requirements

Service_____
Liquid _____
Pumping temp. _____
Capacity (total) normal/max_____/____
No. pumps operating _____
Specific gravity at pump temp. _____
Viscosity at pumping temp. _____
Total head_____
NPSH available_____

Pump type, materials, and accessories

Site conditions

Elevation, ft (m) _____
Range of site ambient temperature

	Dry bulb	Wet bulb
Design, °F (°C)	_____	_____
Maximum, °F (°C)	_____	_____
Minimum, °F (°C)	_____	_____

Atmospheric air

Dust below 10 μm, ppm _____
10 μm and above, ppm_____
Corrosive constituents:
Sulfur, ammonia, ammonium salts, salt or seacoast, other _____

Noise specifications

City, state, federal, other_____

Emission specifications

City, state, federal, other _____

Utilities available at site

Steam: Pressure, lb/in^2 (kPa) gage _____
Temp., °F (°C)_____
Quantity, lb/h (kg/h) _____
Electricity

	V	Phase	Cycles
ac	_____	_____	_____
dc	_____	_____	_____

Cooling water
Source_____quality_____
Supply temp._____min_____max
Supply press, lb/in^2 (kPa) gage
Max return, °F (°C) _____

Information from manufacturer

Pumping requirements

Service_____
Liquid _____
Pumping temp. _____
Capacity (total) normal/max_____/____
No. pumps operating _____
Specific gravity at pump temp. _____
Viscosity at pumping temp. _____
Total head_____
NPSH available_____
Bhp (brake W)

required	_____normal	_____max
Speed, rpm	_____normal	_____max
Efficiency	_____normal	_____max

Pump type, materials, and accessories

Gas turbine excluding gear

	Design	Max	Min
Dry bulb temp., °F (°C)	_____	_____	_____
Output, hp (kW)	_____	_____	_____
Heat rate, Btu/hp · h (J/kWh) (LHV)	_____	_____	_____
Output shaft speed, rpm	_____	_____	_____
Turbine inlet temp., °F (°C)	_____	_____	_____
Gear loss	_____	_____	_____
Net hp (kW)	_____	_____	_____

Total utility consumption

Cooling water, gpm (l/s) _____
Electric power, kW ac/dc _____
Steam, lb/h (kg/h) _____
Compressed air, standard ft^3/min (m^3/min)_____

Shipping data

	Turbine	Aux. items
Shipping wt, tons (kg)	_____	_____
Max erection wt, tons (kg)	_____	_____
Max maint. wt, tons (kg)	_____	_____
Length, ft-in (m)	_____	_____
Width, ft-in (m)	_____	_____
Height, ft-in (m)	_____	_____

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TABLE 1 Typical data sheet for gas turbine driven centrifugal pump (*Continued*)

Information from purchaser	Information from manufacturer		
Fuel Gas _____liquid_____ Analysis attached	Accessories included (as required by purchaser)		
	Inlet air filter	___yes	___no
	Inlet air silencer	___	___
	Exhaust silencer	___	___
	Exhaust duct	___	___
	Starting equipment	___	___
	Load gear (if required)	___	___
	Driven pump	___	___
	Coupling	___	___
	Fire protection system	___	___
	Equipment enclosure	___	___
	Baseplate or soleplates	___	___
	Combined turbine-pump	___	___
	Lub. oil system	___	___
	Main lube pump	___	___
	Auxiliary lub. pump	___	___
	Lub. reservoir	___	___
	Lub. filter	___	___
	Lub. oil cooler	___	___
	Unit control panel	___	___
	Auxiliary motor	___	___
Accessory items required (see list in right-hand column)	Control center	___	___

Source. Adapted from API Standard 616.

FURTHER READING

Bathie, W. W. "Fundamentals of Gas Turbines." 2nd ed. John Wiley & Sons, Inc. New York, 1996.

Baumeister, T., and Marks, L. *Standard Handbook for Mechanical Engineers.* 8th ed. McGraw-Hill, New York, 1978.

Combustion Gas Turbines for General Refinery Services, API Standard 616, 1982.

Dacy, I. R. "Gas Turbines Used on Arctic Pipelines." Oil Gas J. 200:26, 1973.

Proposed Gas Turbine Procurement Standard, ASME.

Sawyer, J. W., ed. *Sawyer's Gas Turbine Engineering Handbook.* 3rd ed. Gas Turbine Publications, Stamford, CT, 1985.

Schiefer, R. B. "The Combustion of Heavy Distillate Fuels in Heavy Duty Gas Turbines." *ASME* 71-GT-56, 1971.

Shepherd, D. G. *Introduction to the Gas Turbine.* 2nd ed. Van Nostrand, Princeton, NJ, 1960.

SECTION 9.2
SPEED-VARYING DEVICES

9.2.1
PERMANENT MAGNET ADJUSTABLE SPEED DRIVES

G. S. HIGHFILL
L. A. HALVERSON

OVERVIEW

The permanent magnet adjustable speed drive is a coupling used where there is a need to connect a constant-speed driver to a load and obtain variable-speed operation. Pump drivers are usually electric motors and they inherently operate at a fixed speed. The inclusion of a permanent magnet adjustable speed drive into the drive train will transmit rotary motion and torque, and allow variable-speed adjustment for the pump/load. Variable speed technology has resulted in significant energy savings, improved pump performance, and reduced maintenance costs, compared with constant speed alternatives.

Two types of magnetic slip couplings are available in industry and they both function similarly. The heritage (earlier technology) product known as the eddy-current coupling is an electromechanical torque-transmitting device. The latest technology product known as the permanent magnet adjustable speed drive is a mechanical torque-transmitting device.

Both work on the principle of magnetic induction and torque is generated as a function of slip between the input and output members. The primary difference between these two types of devices is how the magnetic field is generated—one uses high strength rare-earth permanent magnets and the other uses a field winding that is excited by direct current, usually from a static power supply. The electrical power source is eliminated when using permanent magnets. Another difference is how the device is mounted in the drive train. Many permanent magnet adjustable speed drives can be shaft mounted, similar to a conventional spacer type coupling. This eliminates the base-mounted housing required for eddy-current couplings.

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PERMANENT MAGNET ADJUSTABLE SPEED DRIVES

The permanent magnet adjustable speed drive (ASD) is a mechanical torque-transmitting device. The magnetic technology utilizes powerful rare-earth permanent magnets to transmit torque across an air gap; therefore, the input and output members are mechanically independent. A permanent magnet ASD is nonenergized. The drive unit consists of two primary component assemblies—a magnet rotor that revolves freely within a conductor rotor. When the driver rotates, the relative motion and interaction between the magnet rotor and conductor rotor creates eddy currents that develop torque across the air gap. Varying the width of the air gap provides adjustable torque and variable output speed to the load. A typical shaft-mount air-cooled permanent magnet ASD is shown in Figure 1.

Rare-earth permanent magnets are made of a combination of neodymium, iron, and boron (NdFeB). The magnets are typically plated for corrosion protection. NdFeB magnets have a high energy product, permitting small size and high torque transmission capability, and they have a half life of over 2000 years. NdFeB magnets will retain their performance up to 250 to 350°F, but they can demagnetize at prolonged exposure to higher temperatures.

The magnet rotor assembly contains two aluminum magnet rotors populated with several permanent magnets and is connected to the load (output) shaft. An even number of magnets installed with opposing polarity (north and south in turn) is normal. The conductor rotor assembly is comprised of copper disks mounted to steel plates and is connected to the driver (input) shaft and rotates at constant speed. When the conductor

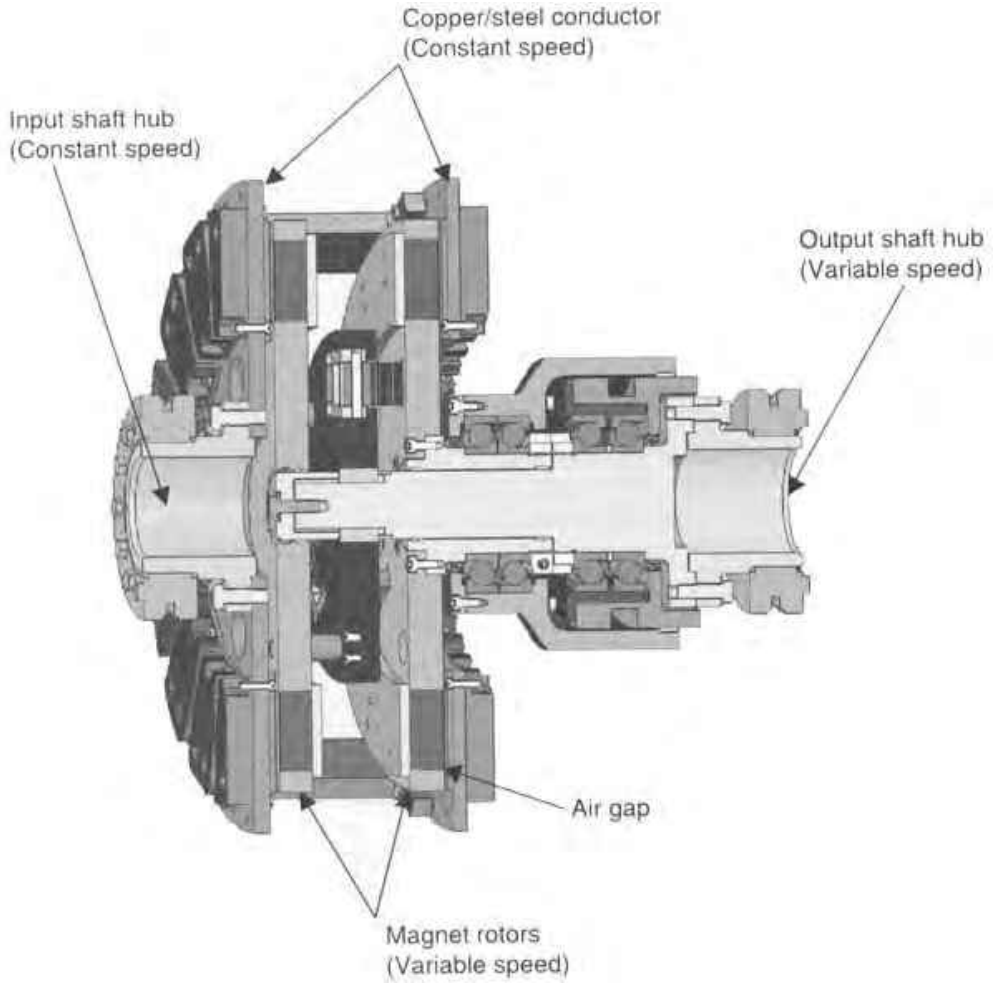


FIGURE 1 Cutaway of a shaft-mount air-cooled permanent magnet ASD (MagnaDrive Corp.)

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rotor rotates relative to the magnet rotor, the magnetic flux from the magnet poles permeates the air gap and creates eddy currents in the copper conductor disks. The eddy currents create poles that interact with the magnet poles, whereby the magnetic flux develops a tangential force tending to turn the magnet rotor in the same direction as the rotating conductor. The net result is a torque available at the output shaft for driving a load. The magnet properties, rotor geometry, and materials of construction determine torque capability.

The magnet rotors move axially about a center pivot to increase or decrease the width of the air gap between the conductor rotors, thereby varying the amount of torque transferred and providing speed control. The center pivot is locked onto a center shaft where a cam driver and follower allow positioning of the magnet rotors through uniform motion. The drive mechanism is actuated through an external control arm that is supported about the center shaft by a pair of antifriction bearings. The center shaft is fastened to the output member and properly aligned by a machined register fit to the shaft coupling hub. Relative position between input/output rotors is maintained by the driver and load shafts, or when installed inside an enclosure, by a pair of rolling element bearings. Air gap, output torque, and load speed are usually not proportional, therefore, load torque characteristics and speed range must be known for proper permanent magnet adjustable speed drive size selection.

EDDY-CURRENT COUPLINGS

The eddy-current coupling is an electromechanical torque-transmitting device installed between a constant-speed prime mover and a load to obtain adjustable-speed operation. In most cases, the eddy-current coupling has an appearance very similar to that of its driving motor except that the coupling has two shaft extensions. One shaft, which operates at constant speed, is connected to the motor; the other shaft, providing adjustable speed output is connected to the load. A typical self-contained air-cooled eddy-current slip coupling is shown in Figure 2.

The input and output members are mechanically independent, with the output magnet member revolving freely within the input ring or drum member. An air gap separates the members and a pair of rolling element bearings maintains their proper relative position.

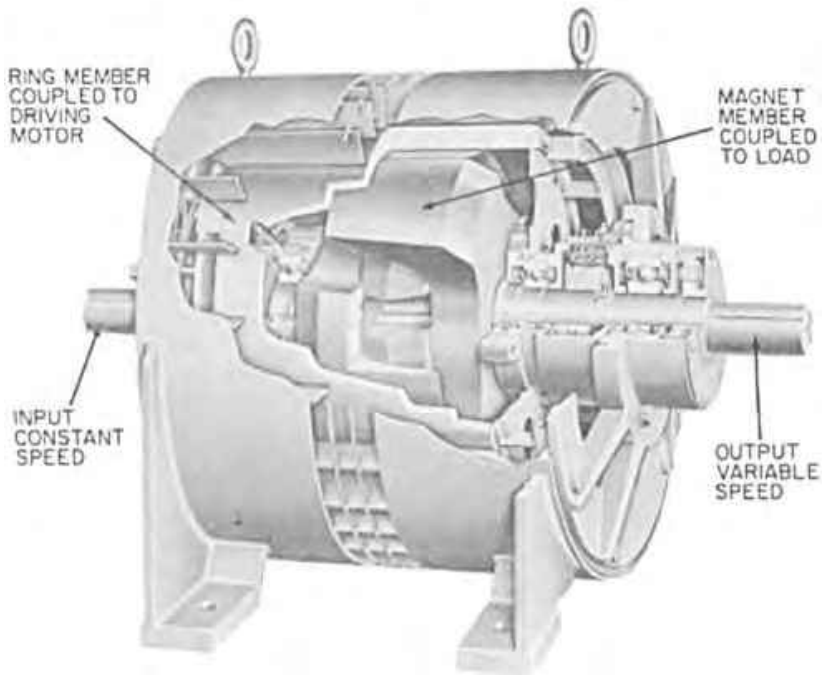


FIGURE 2 Cutaway of an ampli-speed eddy-current coupling (Electric Machinery Mfg.)

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The magnet member has a field winding that is excited by direct current, usually from a static power supply. Application of this field current to the magnet induces eddy currents in the ring. The interaction between these currents and magnetic flux develops a tangential force tending to turn the magnet in the same direction as the rotating ring. The net result is a torque available at the output shaft for driving a load. An increase or decrease in field current will change the value of torque developed, thereby allowing adjustment of the load speed. Field current, output torque, and load speed are usually not proportional, therefore, load torque characteristics and speed range must be known for proper eddy-current coupling size selection.

Most eddy-current couplings have load speed control. An integral part of this system is a magnetic pickup that provides an indication of exact output speed and enables control of load speeds within relatively close tolerances regardless of reasonable variations in load torque requirements. Slip-type adjustable speed drives without load speed control may experience fluctuations in load speed when torque requirements vary.

CONFIGURATIONS

The permanent magnet ASD can be incorporated as an integral part of the drive system, mounted between parallel shafts or connected to right-angle gearing. They are available in horizontal (Figure 3) and vertical (Figure 4) configurations. The drive can be shaft mounted, same as a conventional coupling, using hubs secured directly to the motor and load shafts, or the drive may be mounted inside an enclosure, similar to an eddy-current drive, with two shaft extensions that are flexibly coupled to the driver and pump. When mounted inside an enclosure, the permanent magnet ASD has an appearance very similar to an eddy-current coupling (Figure 5). The most economical selection is a shaft-mount drive unit. Horizontal machines up to 400 horsepower are frequently shaft-mount air-cooled and larger power sizes are usually mounted inside an enclosure. Vertical motors are

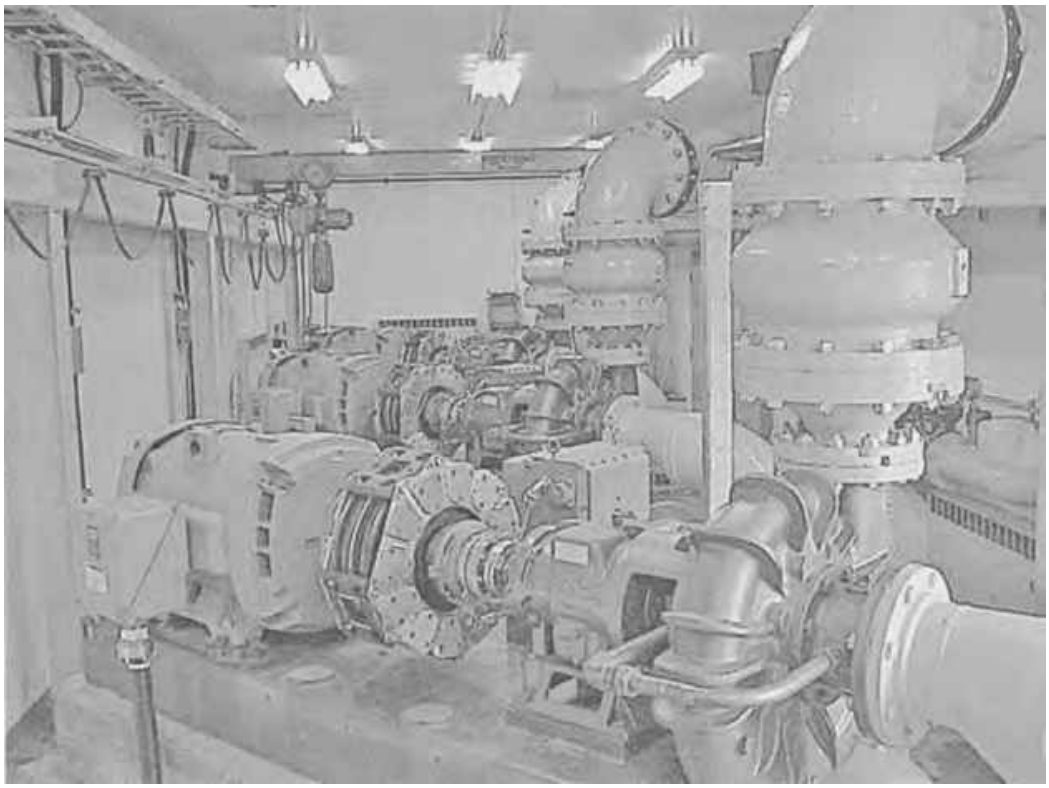


FIGURE 3 Horizontal centrifugal pump driven by 200-hp, 1780-rpm induction motor and permanent magnet ASD (MagnaDrive Corp.)

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FIGURE 4 Vertical centrifugal pump driven by 350-hp, 1180-rpm induction motor and permanent magnet ASD (MagnaDrive Corp.)

close-coupled to limit overall height and to prevent vibration problems. Pump hydraulic thrust requirements can be accommodated with thrust bearings, typically located in the bottom of the permanent magnet ASD.

Enclosures (housings) will vary depending on the size of the permanent magnet ASD and on the type of cooling required. Air-cooled installations in clean or dirty atmospheres need open or drip-proof enclosures where heat rejection is not a problem. Enclosures are weather protected and suitable for outdoor installations. Both horizontal and vertical installations above 500 horsepower require water-cooled, sealed enclosures (Figure 6). No special considerations are necessary except to provide a clean, pressurized water supply and employ caution in outdoor installations where freezing can occur. Cooling water may be discharged on site, otherwise a coolant recirculation system is required.

PERFORMANCE FUNDAMENTALS

The permanent magnet adjustable speed drive, like eddy-current couplings, is a constant torque device that operates on the slip principle and is classified as a torque transmitter. This means that input and output torque are essentially equal and speed change is achieved by modifying the amount of torque transferred through the device. Power is a

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FIGURE 5 Vertical centrifugal pumps driven by a permanent magnet ASD (front), an eddy-current coupling (center), and a direct coupled motor (back) (MagnaDrive Corp.)

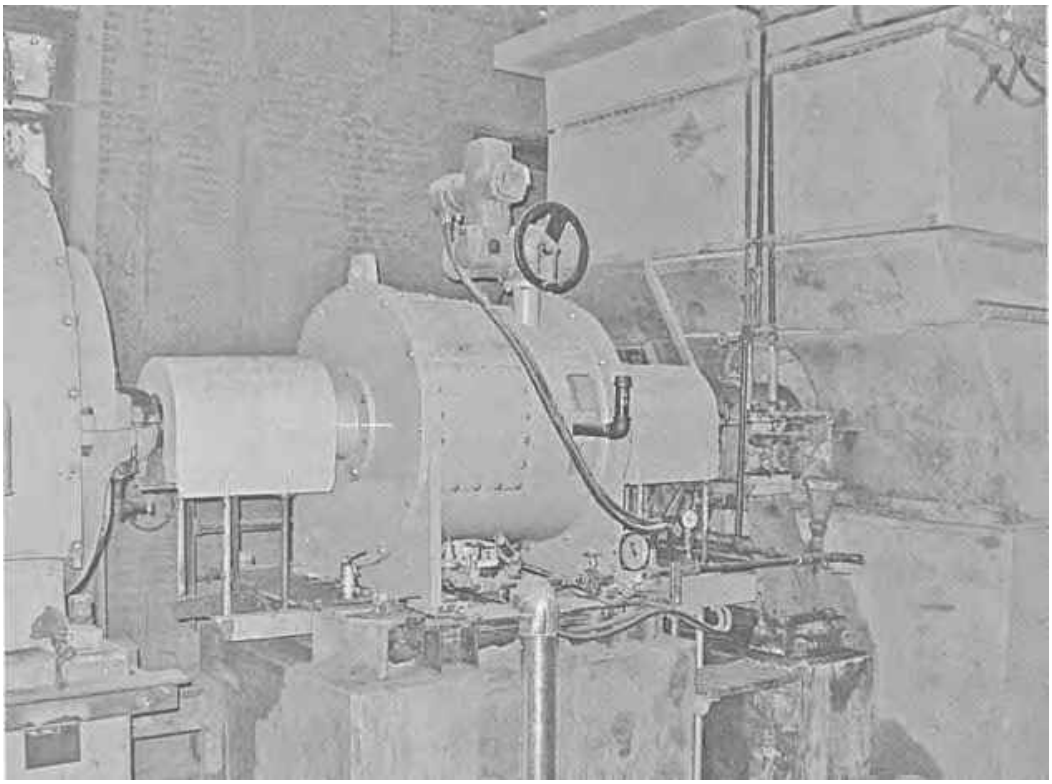


FIGURE 6 Horizontal induced draft fan driven by 700-hp, 1180-rpm induction motor and water-cooled ASD (MagnaDrive Corp.)

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function of speed and torque. The fundamental formula for power is given below for both USCS and metric units.

$$\text{hp} = \frac{\text{rpm} \times T}{5252}$$

$$\text{kw} = \frac{\text{rpm} \times T}{9545}$$

where rpm = shaft speed
T = torque, ft · lb (N · m)

A permanent magnet adjustable speed drive must slip in order to transmit torque. Applying the above power equation separately to the input shaft and the output shaft shows that there is a difference in power. This difference in power is called slip loss, and is in the form of generated heat. There are also windage and frictional losses, but they are usually small enough to be neglected. Common values of slip for a centrifugal pump application are from 1 to 4%. The power formula can be used to determine slip horsepower.

$$\text{hp}_{slip} = \frac{(\text{rpm}_1 - \text{rpm}_2) \times T}{5252}$$

$$\text{kw}_{slip} = \frac{(\text{rpm}_1 - \text{rpm}_2) \times T}{9545}$$

where rpm_1 = motor speed
rpm_2 = load speed
T = load torque, ft · lb (N · m)

Permanent magnet adjustable speed drives and eddy-current couplings typically control centrifugal equipment such as pumps and fans. This type of equipment has a variable torque load that increases approximately as the square of the speed. Load power increases approximately as the speed cubed. This type of load increase is specifically for systems where the total system head is developed from frictional resistance (dynamic losses). Examples of these systems include pumping between reservoirs having the same levels or in a closed loop. Figure 7 shows the load power curve as a function of motor speed for such a system and illustrates the motor power curve and slip loss, calculated from the equations above. It can also be seen that efficiency is approximately equal to the percentage slip.

Many centrifugal equipment systems must first reach a static head level before flow begins. Figure 8 is an example of such a system. Load power curve OAB illustrates the closed discharge power. At point A the static head is overcome, and the pump load then rises along curve AC. Typically, even large static head systems will use the same permanent magnet ASD or eddy-current coupling appropriately sized for the ideal system shown in Figure 7. However, note that the peak slip loss value is reduced.

Performance is typically rated for continuous operation with ample torque for start-up and over-speed based on the horsepower and speed of the driver. The coupling's ability to transmit torque is a function of slip speed. Figure 9 shows power transmission as a function of load speed—note that high load speeds equate to low slip speeds. The initial part of the curve at full speed (low slip speeds) is often approximated as linear and defined by a single "torque per rpm slip" factor that can be used for estimating slip at the maximum speed position. The peak torque is also called the "breakdown" torque value, as further increases in slip speed will cause a reduction in transmitted torque (and power).

Slip-type couplings are best suited for controlling speed of centrifugal equipment, as discussed above. The power capacity of a permanent magnet ASD is determined by both the torque capacity and the amount of heat that can be removed from the conductor plates. All slip couplings and drives generate heat that must be dissipated by air-cooling or, for

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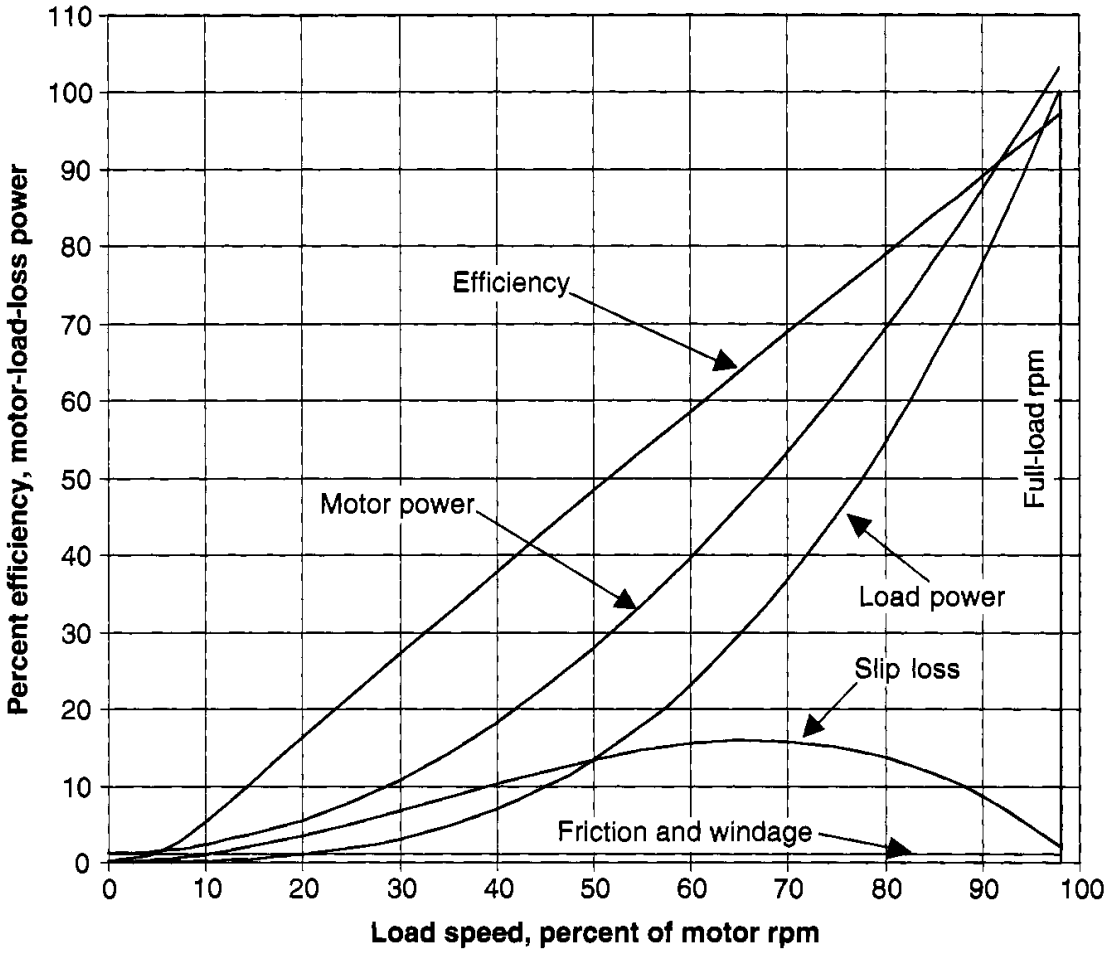


FIGURE 7 Load, power, and slip characteristics for a friction-only pumping system

high horsepower products, water-cooling. The heat is dissipated through the conductor plates (where eddy currents are developed) and must be removed to avoid overheating the magnets and/or causing thermal damage.

Two methods of heat dissipation are employed. Air-cooled units are primarily cooled by convection into the surrounding air, making their heat capacity a function of ambient air temperature and conductor rotating speed. At a given motor speed, each unit is rated for its heat capacity at 70°F ambient temperature, in horsepower. The coupling can dissipate its rated heat capacity in slip heat continuously. Water-cooled units rely on a system to supply coolant to the conductor plates. The water is pumped into an annulus that feeds water directly to the conductor to draw out the slip heat. A continuous supply of cool water must be available, or a circulating system with heat exchanger must be provided.

Slip loss can be calculated for each operating point from the torque required, using the slip hp equation. With a machine that has a cubic power relation to speed, the maximum heat generated (in power) is approximately 16% of rated power at full load speed, and is generated at 67% motor speed (Figure 7).

A permanent magnet adjustable speed drive controls speed by varying the air gap between the magnet and conductor rotors. Increasing the air gap will lower the torque transmitted by the coupling and reduce the load speed. A fully engaged coupling will operate at the intersection between the maximum power versus speed curve and the system curve as shown in Figure 9. As the air gap is increased (opened), the system will now operate at the intersection between the new power versus speed curve and the system curve. Minimum speed (turndown) is achieved when the air gap is fully opened as dictated by the

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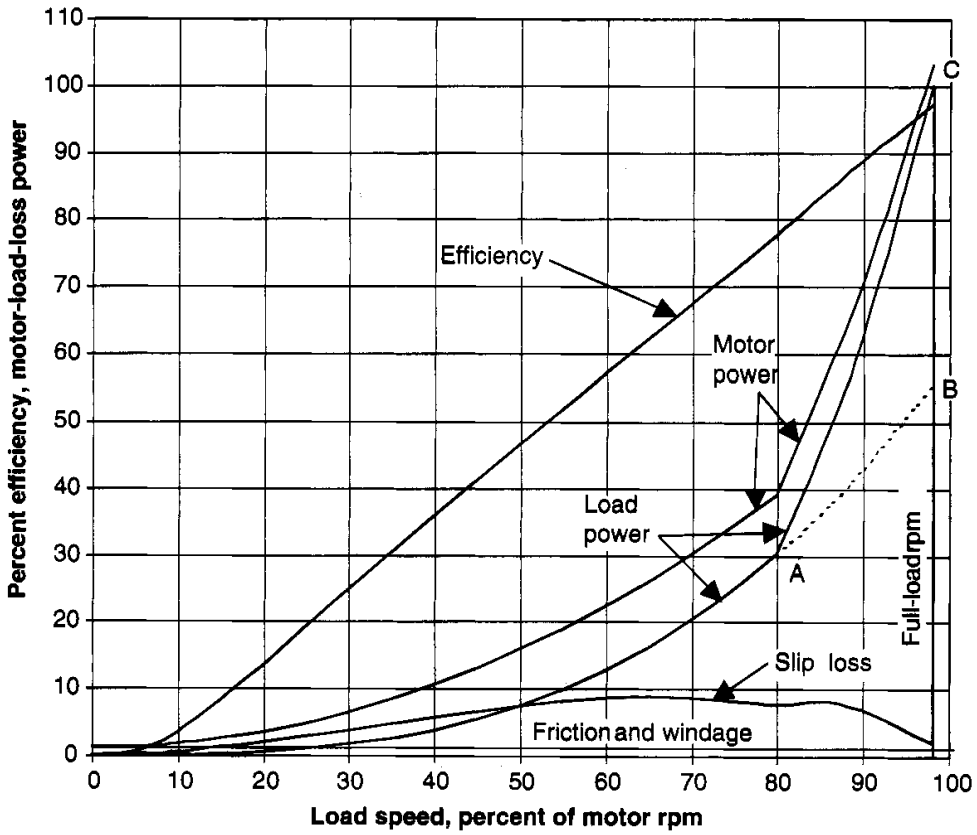


FIGURE 8 Load, power, and slip characteristics for a static head and friction pumping system

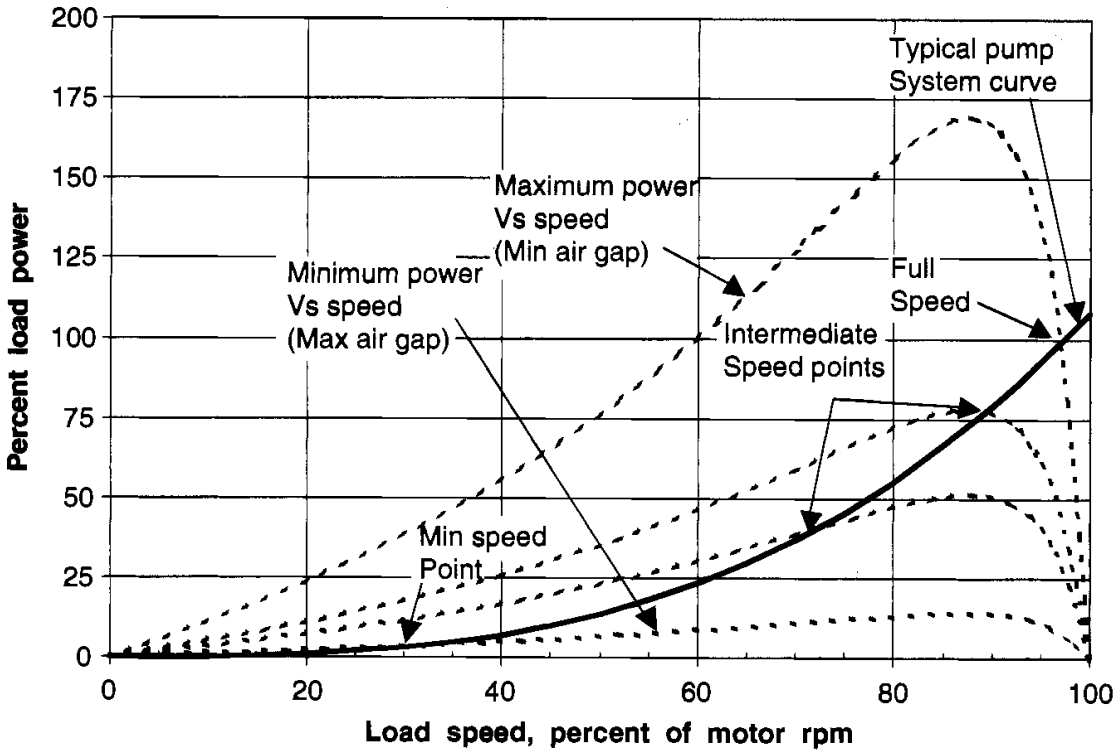


FIGURE 9 Typical load, power, and slip characteristics for a variable speed application showing intersections with the pump system curve

minimum power versus speed curve. Turndown speed ratios of 4:1 are typical and can adequately accommodate centrifugal pumps that will operate in their minimum flow region.

It is important for the system curve to have a steeper slope than the power versus speed curve. If the system curve is too flat or the power versus speed curve too steep, there could be multiple intersection points along the system curve. In certain speed ranges, the drive could provide more power than required by the system curve, causing the unit to accelerate through those ranges without providing control stability. Also note that constant torque applications must be carefully reviewed and may require special conductor materials to achieve desired control and turndown speed.

ENGINEERING CONSIDERATIONS

Permanent magnet adjustable speed drives are utilized to drive centrifugal pumps in many applications where variable flow and pressure is required. They provide vibration isolation between pump and driver because there is no mechanical connection between input and output members. For shaft-mount arrangements, this allows greater misalignment between equipment to accommodate thermal expansion, system fluctuations and pipe strain. When compared to flow control valves that restrict flow to desired levels, a permanent magnet ASD saves energy by reducing the rotational speed of the pump/load to match the variable process requirements. The motor operates near its synchronous speed, where maximum cooling efficiency is obtained, and output torque is lower so power consumption is reduced. Smooth transition between set points is critical to process control and by efficiently responding to any input signal through a mechanical drive mechanism, precise speed adjustment is attained. The permanent magnet ASD generates no electronic harmonic distortion and no voltage transients that are detrimental to the system and can disrupt electronic equipment. Likewise, they are not affected by lightning strikes or power fluctuations. They provide no-load or cushioned start/stop capability to the motor and pump, which reduces fatigue stress in the system, inrush current at start-up, and motor winding heat.

Drive systems should be given special consideration with regard to residual unbalance. In general, the major rotating assemblies of a permanent magnet ASD should be balanced to maintain the equipment within acceptable vibration levels. The type of balance and limits of imbalance vary with the component weight and radius, operating speed, and selected industry specifications. Generally, the rotating components are balanced to meet the requirements of ISO 1940-1 grade G6.3.

For an assembled drive unit, electromagnetic interference (EMI) is not a factor and is less than that typically found in an electric motor. However, for disassembled units, individual magnets and exposed magnet rotors require strict safety precautions and handling procedures—consult manufacturer for recommendations.

During maximum slip heat condition (~67% motor speed), conductor rotor surface temperatures can easily reach 250°F. Protective guards must provide sufficient airflow to allow adequate heat dissipation. Disconnected input and output shafts offer equipment protection. A failure or seizure of the equipment load will not result in shaft breakage or pump damage, but may result in unusually high slip speeds that can quickly overheat and damage the permanent magnet ASD. At elevated temperatures, magnets may become partially demagnetized and the conductor material may breakdown. In these instances, the drive unit should be instrumented with a noncontact optical or other type of temperature/speed sensor to provide alarm and shutdown circuitry.

Weight, length, and center of gravity should be considered for shaft-mount arrangements to evaluate overhung loads and estimate critical frequencies, especially with long, small diameter shafts. The foundation must be of adequate strength and stiffness to support the weight of the drive and its enclosure. Vertical configurations are much more flexible than most horizontal arrangements, making them more susceptible to resonance. The system critical (reed) frequency must always be reviewed to verify the pump discharge head has adequate stiffness to support the permanent magnet ASD and the motor without operating near the natural frequency of the installed equipment.

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CONTROLS

The movement needed to change the magnet rotor position and decrease/increase the output shaft speed is produced by an actuator. The type of actuator motion necessary to control the adjustable speed drive is either linear or rotary. A typical arrangement results from linear translating motion provided by a quarter-turn actuator working directly through linkage connected to the drive mechanism control arm.

The choice of actuator depends on the service, usually modulating. In modulating service, the actuator must adjust to changing speed demands of the system. Actuator torque, stroke speed, its ability to hold a speed control setting and provide stability under fluctuating loads are important factors when selecting an actuator. Actuators may be manually powered (gear box), pneumatic/hydraulic powered, or, most commonly, electric motor driven.

A closed feedback control loop can be added to the system that receives an input signal (usually air pressure or electric signal) from a controller or process source and adjusts the drive's output speed to a prearranged corresponding value. Instrumented process signals may originate from load speed, flow or pressure sensors. Typically, the controller contains a signal isolator to allow precise speed adjustment from an automatic 4 to 20 mA or 0 to 10 VDC control signal. Position accuracy is held within 0.1% of the set speed throughout the operating range. Most actuators include limit switches and the drive mechanism contains mechanical stops, both of which protect the unit from overpowering and over engaging.

INSTALLATION

The basic permanent magnet adjustable speed drive is factory assembled and balanced. They can be installed and maintained by millwrights (no electrician necessary). Regular and preventative maintenance techniques for a permanent magnet ASD are very similar to other types of rotating equipment. Shaft-mount arrangements connect directly to the driver and load shafts using keyed or shrink-disc coupling hubs, similar to a conventional coupling. Enclosed arrangements mount on their own pedestal and require conventional couplings to connect to the driver and load shafts. For enclosed units, the foundation and pedestals must be of adequate strength and stiffness to support the weight of the drive, to maintain correct alignment between connected machinery, and to minimize the effects of vibration. During field mounting, the unit should be aligned following manufacturers recommendations and securely bolted in place.

The permanent magnet ASD can be installed indoor or outdoor without special provisions. The drive is a nonenergized mechanical device; therefore, under normal operating conditions, it may be applied in hazardous services, and used in corrosive or explosive environments. Only the actuator needs to be rated for hazardous locations in order to place the drive within an explosive environment. Other hazardous and corrosive environments should be evaluated for materials of construction, but versatility is not affected. Since the permanent magnet ASD does not require an external power source, it can be used where electrical power is not available. All that is necessary is a driving shaft and a load shaft. A manual gear actuator can be used to control speed.

LIMITED END FLOAT, THRUST CAPABILITY AND LUBRICATION

Permanent magnet adjustable speed drives have limited axial end float capability—a result of the air gap between rotors. Many horizontal motor-driven pump systems utilize motors equipped with sleeve journal bearings and no thrust bearing. For horizontal shaft-mount configurations, both pieces of machinery being connected must have a thrust bearing to limit the total axial end float no greater than 0.03 in. Alternatively, horizontal enclosed configurations are equipped with a thrust bearing eliminating the concern with machinery end float limitations.

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Most vertical permanent magnet ASD are supplied with a thrust pot designed to handle the down-thrust and potential up-thrust generated by the pump/load and limit axial end float. A variety of external down-thrust load capabilities are available from the manufacturer. Bearing life is an important factor and most designs satisfy a minimum ABMA L_{10h} life of 40,000 h at an average output speed equivalent to the input speed. Bearings are angular-contact ball type or, where higher thrust values are encountered, spherical roller bearings may be used.

Bearing lubrication is generally accomplished either by grease or oil. Grease-lubricated bearings are the most common lubrication system for light to medium duty permanent magnet ASD. Oil lubrication is the common system for large horsepower water-cooled drives and vertical configurations with thrust pots. A typical system includes oil-lubricated bearing housing with a vented fill cap and an oil level sight glass to allow visual inspection of the oil level. Either too high or too low oil level can cause improper lubrication of the bearings, reducing their life.

APPLICATIONS

Permanent magnet adjustable speed drives are used extensively with centrifugal pump applications where the pump must deliver a wide range of flow over time. Examples of pumping systems which demand a range of flow include water and wastewater pumping in municipal installations, process pumping in a variable capacity process plant, boiler-feed and cooling-water pumping, circulating water and condensate pumping in power stations, stock pumping in paper mills, process water, reclamation and dewatering in mining and mineral processing, drainage, irrigation and flood control pumping, oil and gas processing, and HVAC chilled water and cooling water pumps in commercial and institutional buildings, and in oil and gas processing. Practically any pump application (or fan application), where cyclic constant-speed operation, fluctuations in process demand, valve throttling or other means of flow control are alternate considerations, can be conveniently solved with the use of a permanent magnet ASD as the variable-speed flow controlling device.

ENGINEERING SELECTION

Permanent magnet ASD are designed—and best suited—for driving equipment where power requirements vary exponentially with speed (rpm) -see Sections 2.1.1 and 2.1.3 for "similitude" and "affinity laws". For any centrifugal pump application, the engineer can make an initial size selection based upon the motor speed and nominal horsepower (Table 1). The user's system will further define the selection process dependent upon several factors, including power, rotational speed, shaft separation, equipment orientation (vertical/horizontal, shaft-mount or enclosure), system influences, reliability, type of permanent magnet ASD required, and any special specification requirements. A system-wide assessment is important during the evaluation/selection process and the total cost of ownership model is a helpful tool when evaluating product, operating, maintenance, and downtime expenses.

The following denotes the basic information required by the permanent magnet ASD manufacturer for selection purposes.

- Driver type, horsepower, speed, and shaft end details
- Driven machine type, power required, variable speed range, shaft end details, and how torque changes with speed
- Site conditions, environment (indoor/outdoor), ambient temperature, and elevation
- Cooling medium available
- Control type
- Accessories
- Specifications

TABLE 1 Permanent magnet ASD selection chart, horsepower output

Numbers in Table represent typical ASD model sizes
Example: 75 hp at 1800 rpm = 14.5 ASD
Centrifugal Applications Only

	Motor Speed (rpm)				
Motor hp	3600	1800	1200	900	600
10	6.5	6.5	8.5	10.5	10.5
20	6.5	8.5	10.5	12.5	12.5
30	8.5	10.5	12.5	14.5	14.5
40	8.5	12.5	12.5	14.5	16.5
50	8.5	12.5	14.5	16.5	18.5
60	10.5	12.5	14.5	16.5	18.5
75	10.5	14.5	16.5	18.5	20.5
100	12.5	14.5	18.5	20.5	22.5
125	12.5	16.5	18.5	22.5	24.5
150	14.5	18.5	20.5	24.5	26.5
200	14.5	18.5	22.5	26.5	31
250	16.5	20.5	24.5	26.5	31
300	16.5	22.5	26.5	31	31
350	18.5	24.5	31	31	31
500	20.5	26.5	31	31	33
750	31HS	31	31	33	33
1000	31HS	31	33	33	33
1500	31HS	33	33	33	35
2000	31HS	33	33	35	35
2500	–	33	35	35	35
3000	–	33	35	35	37
4000	–	–	35	37	37
5000	–	–	35	37	–
6000	–	–	–	37	–

Note: 1 hp = 0.746 kW

FURTHER READING

ABMA 9-1990, *Load Ratings and Fatigue Life for Ball Bearings.*
ISO 1940-1, *Mechanical Vibration—Balance Quality Requirements of Rigid Rotors—Part 1: Specification and Verification of Balance Tolerances.*
Volk, M. W. *Pump Characteristics and Applications,* Marcel Dekker, Inc., New York, 1996.

9.2.2 SINGLE-UNIT ADJUSTABLE-SPEED ELECTRIC DRIVES

E. O. POTTHOFF
J. R. HENDERSHOT

In past years, the electric motor industry has been dominated by the use of essentially single and constant speed motors when line fed. There were many mechanical methods used to achieve adjustable speed using a single-speed prime mover. There was also widespread use of dc motors for adjustable speed over the last century because only the voltage must be adjusted for speed change. Although operation at speeds well below synchronous can be accomplished through adjustment of the secondary resistance in wound-rotor induction motors, the ubiquitous squirrel-cage ac induction motor requires the adjustment of the frequency to change speed. This was not so easily accomplished until the recent utilization of pulse width modulation (PWM) inverters with insulated gate bipolar transistors (IGBTs) for power transistors plus microprocessors.

Electric drives are available featuring a single rotating element and some associated control for performing the adjustable-speed driving function. The use of a single rotating element differentiates this type of drive from the eddy-current coupling motor, hydraulic coupling motor combinations, and adjustable-speed belt drives, all of which are tandem drives.

The speed of the single-unit adjustable-speed drive is controlled through the interaction of the control and the motor. For this reason, one must consider these two elements as a drive and not consider the motor alone.

Section 9.1.1 described motors that will be further discussed in this section. Previous discussions of control were limited to that required for starting and protection; this section will also describe specific controls required for speed control. The discussion will cover the following types of drives:

- Ac adjustable-voltage drives
- Wound-rotor induction motors with several different types of secondary controls
- Adjustable-frequency drives

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- Modified Kraemer drives
- Dc motors with silicon-controlled rectifier (SCR) power supplies

All have physical features, operating characteristics, or prices that make them particularly valuable in some specific segment of the pump driver spectrum.

ALTERNATING-CURRENT ADJUSTABLE-VOLTAGE DRIVES

This drive consists primarily of an adjustable voltage, constant frequency control, and a motor (M) conveniently configured as shown in Figure 1. The motor must possess high slip characteristics and other characteristics that allow it to work successfully with the associated control. The motor is designed for operation and tested with its associated control. Because the motor has high slip characteristics, insulation is of Class F rating.

The fundamental parts of the control are a circuit protective device, such as a circuit breaker, an SCR assembly, and firing circuitry, identified as FC in Figure 1. Other parts are required to complete the control but serve as auxiliaries. The main function of this control is to provide a voltage to the motor at a level that will ensure a desired motor speed. The control also protects the motor under abnormal operating conditions and the motor cable under short-circuit conditions and provides a current-limit function so the motor draws a maximum of some preselected value, such as 150% of normal, under all operating conditions.

The SCR receives impressed voltage, usually at 60 Hz. The SCR can be turned on (or become conducting) by means of current pulses received from the firing circuits that energize the circuits at different points on the sinusoidal wave. After conductance starts, it continues until voltage disappears at the end of the half-cycle. The SCR must be pulsed again in order to become conducting. This occurs while the voltage is in the negative phase of its sinusoidal generation and is performed by an SCR connected in parallel to the first and having reverse polarity. The negative loop conductance continues until the voltage again returns to zero. The firing controls are designed to turn the SCR on repetitively sometime during each voltage half-cycle. Figure 1A shows the shape of the applied voltage between the SCR and the motor for the wave illustration shown. In this case, the SCR is turned on at the beginning of each half-cycle.

By delaying the firing, or turning on, of the SCR until later in voltage generation, shorter intervals of applied voltage and lower levels of voltage appear across the motor. The solid line of Figure 1B illustrates applied motor voltage when the voltage half-wave is half-completed. Figure 1C illustrates the motor-applied voltage when the half-waves are approximately 75% completed. Notice in these illustrations that the frequency of the voltage applied to the motor does not vary; only the voltage magnitude does.

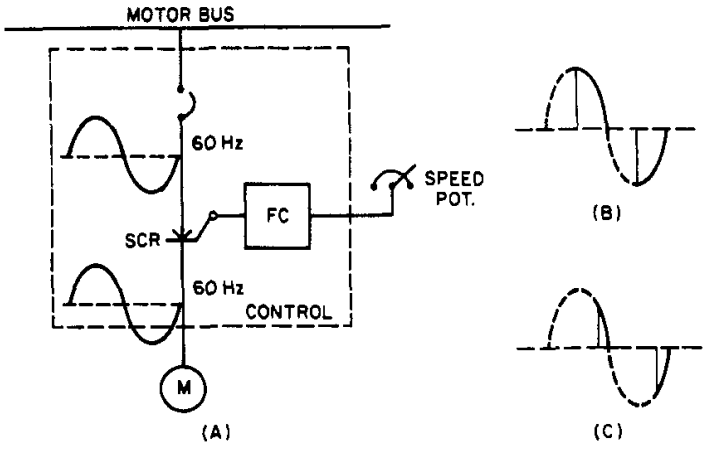


FIGURE 1A through C Block diagram for ac adjustable-voltage drive (General Electric)

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The exact point in the half-wave when firing occurs is controlled by a low-energy electronic signal that may come from a potentiometer, as in Figure 1A, or from some process instrument signal. By increasing the signal level, voltage applied to the motor increases; on the other hand, a decrease in signal level decreases the voltage level.

Figure 1A shows the SCR on the utility side of the motor as a convenience in illustration. In reality, the SCR is generally placed at the motor neutral point. This reduces voltage from the SCR to ground and allows the motor impedance to protect the SCR somewhat from damage caused by transient voltage spikes entering from the electric utility line. However, placing the SCR at the motor neutral point does require the use of six motor conductors instead of the conventional three.

Figure 2 illustrates representative motor speed-torque curves at rated and other voltages as well as a pump speed-torque curve varying as the square of the speed. The motor possesses approximately 10% slip under rated torque and voltage conditions. Note that motor torque does not break down at speeds lower than breakdown torque speed. These various characteristics help to identify the motor as one designed to operate specifically for this application. Note that by reducing applied motor voltage, motor torque decreases, causing the pump to operate at lower speed.

Ratings of these drives are limited essentially to low power levels. Figure 3 provides a guide to those available in open construction only. All ratings with maximum rated speeds existing within the solid envelope are available. Some, but very few, are available outside of this envelope. Totally enclosed motors are more restricted in supply than open motors and are limited to approximately 40 hp (30 kW) maximum. In addition to these restrictions, load torques must not exceed values varying as the square of the speed; thus this drive is not a candidate for driving a constant-torque load. Table 1 gives some pertinent application information.

WOUND-ROTOR INDUCTION MOTORS

Liquid Rheostat Controls This form of drive consists primarily of a full-voltage, nonreversing (FVNR) starter, a wound-rotor induction motor, and a liquid rheostat, all of which integrate into a configuration as illustrated in Figure 4. The FVNR starter switches power to and from the motor stator as well as provides generally accepted protection to

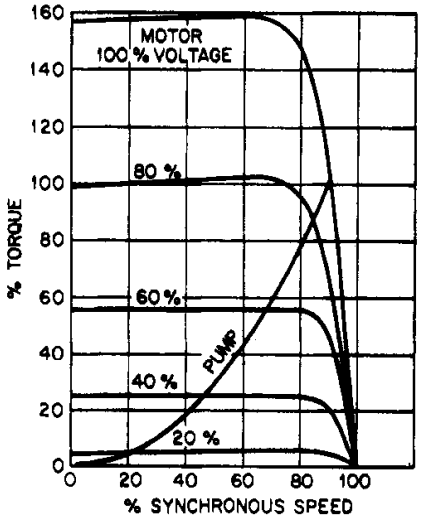


FIGURE 2 Torque versus speed for ac, adjustable-voltage drive (General Electric)

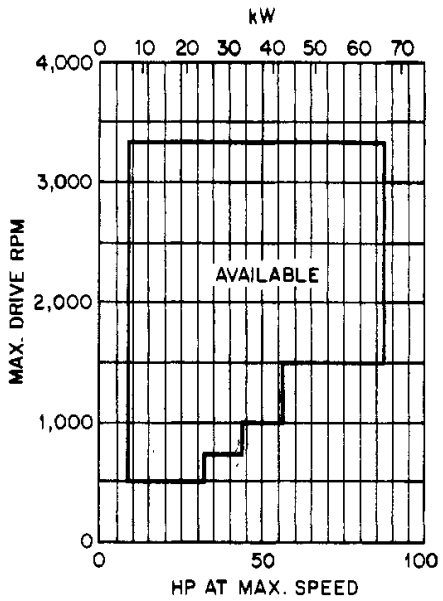


FIGURE 3 Availability of ac adjustable-voltage drives

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TABLE 1 Alternating-current adjustable-voltage drive data

Drive element	Power rating[a]	Voltage rating	Max rated speed, rpm	Speed range, %	Enclosure	Mounting	No. of speed points
Motor	See Figure 3 for open frame ratings; limited availability outside these limits	200 230 460	1640 1095 820 655 545 & others	100	Vertical: shielded drip-proof; TEFC not generally available	Vert. or horiz.	Infinite
	Totally enclosed ratings limited as shown under Enclosure				Horizontal: drip-proof, TEFC to 40 hp (30 kW) approx.		
Control	5–150 hp (3.7–112 kW)	200 230 460	...	...	NEMA 1 NEMA 12	Wall or floor	Infinite

[a]Drive is capable of operating with power varying as the cube of speed maximum. Specifically, these ratings are not suitable for constant-torque applications.

the motor from short circuits, overloads, and so on. Secondary cables connect the motor rotor and fixed electrodes in the liquid rheostat. The fixed electrodes exist in separate cells of the rheostat, one for each phase. Movable electrodes, one located in each cell, are suspended from a horizontal bar, and this bar, the vertical-electrode suspension bars, the movable electrodes, and the electrolyte filling the cells complete the Y, or common point in the external motor rotor circuit. When the upper electrodes are moved up or down, the secondary circuit resistance of the drive varies and this causes a change in the motor speed-torque characteristic.

As already pointed out, the motor starter provides the normal protective functions for the motor. Control circuitry allows the motor to be started with maximum secondary resistance in the rotor circuit, thus drawing minimum motor current from the power supply.

Figure 4 illustrates a motor (M3) operating a pulley that changes the position of the movable electrode. This motor could be controlled manually by a push button or automatically through a controller position providing either a raise-lower or modulated voltage signal. If desired, a pneumatic cylinder may substitute for motor M3 to provide power to move the electrodes.

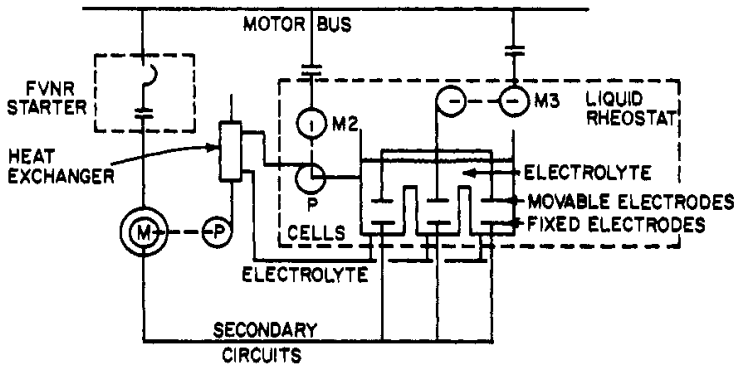


FIGURE 4 Block diagram for wound-rotor induction motor with liquid rheostat secondary power (General Electric)

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The electrolyte receives the drive slip losses dissipated in each of the cells, and of course this heat must be removed into some heat sink capable of dissipating it. In some installations, an electrolyte pump driven by motor M2 circulates the electrolyte through a heat exchanger before returning it to the individual cells. Returning cooled electrolyte reenters the cells under the fixed electrodes and passes through holes in the electrodes before passing vertically through the cells. The heat exchanger primary coolant can be tap water or mill water, which provides a good means of passing drive slip losses as heat directly out of the station, or it can be the pump discharge water, which conveys heat directly away from the building, as illustrated in Figure 4.

The liquid rheostat is factory-assembled with the rheostat, electrolyte pump, and its motor and electrode-positioning assembly packaged in a single steel enclosure suitable for control lineups except for ratings above 700 hp (522 kW). The heat exchanger may be of the sleeve (or wrap-around) type, as illustrated in Figure 4, and comes as a separate device to be attached to station effluent piping. Other forms of exchangers, such as liquid-to-air or shell- and-tube heat exchangers, are available.

Changing the position of the movable electrode results in a change in motor speed-torque relations, as shown in Figure 5. Each curve shown (except the pump curve) represents a motor characteristic for a discrete secondary resistance. Notice the number associated with each curve; it represents the percentage of secondary resistance external to the motor rotor; 100% ohms provide 100% motor torque at zero speed. In examining individual curves, note that as the percent resistance increases, the slope of the motor speed-torque curve decreases, thus allowing the motor to slide down the pump speed-torque curve and assume a lower speed.

This drive utilizes resistance only in the motor secondary circuit as a means of controlling speed. Reactance is available as a substitute for resistance in other forms of secondary controllers, but its use reduces the drive power factor and thereby increases motor current, reduces efficiency, and increases motor heating. As a further effect, the additional motor current may require the use of a larger motor frame to accommodate it.

Figures 6A, B, and D provide clues to the availability of enclosed vertical and horizontal wound-rotor induction motors. Figure 6A illustrates the point that vertical, totally enclosed wound-rotor induction motors are not generally available but may be available in random powers and speeds. Totally enclosed water-to-air-cooled (TEWAC) construction becomes available at some minimum point as illustrated. In large ratings, shown on the right, TEWAC construction becomes very important as it provides a convenient way of capturing motor losses and expelling them in cooling water external to the building. Both curves of Figure 6A terminate at 1800 rpm synchronous speed because higher speed ratings are generally unavailable.

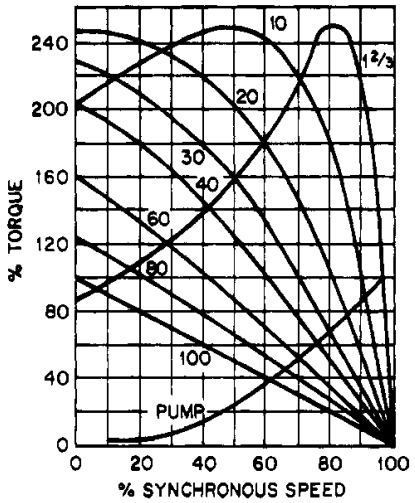


FIGURE 5 Torque versus speed for wound-rotor induction motor with variable secondary resistance (General Electric)

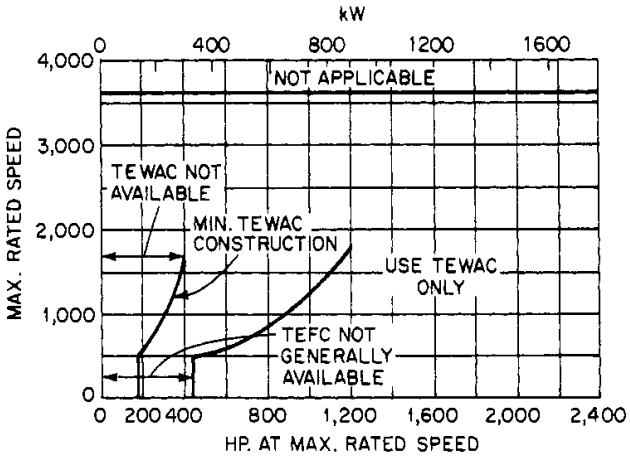


FIGURE 6A Approximate availability of totally enclosed vertical wound-rotor induction motors (General Electric)

Figure 6B delineates the left side, where totally enclosed fan-cooled (TEFC) construction should be used for horizontal wound-rotor induction motors because it is the only one available, the middle area, where TEFC construction should be used because it is less expensive than TEWAC construction, and, finally, the right side, where TEWAC construction is less expensive than TEFC construction. Again, motor speeds are limited to 1800 rpm synchronous.

Vertical wound-rotor induction motors are not available in explosion-proof construction, as their omission from Figure 6C implies.

Finally, the areas of availability and unavailability are delineated in Figure 6D for horizontal wound-rotor induction motors in explosion-proof construction. Again, the curve terminates at 1800 rpm synchronous speed for reasons already expressed.

Delineation of motor availability is an important criterion in drive selections. Table 2 presents significant data useful in drive selection.

Tirastat II* Secondary Controls This drive utilizes the same wound-rotor induction motor and FVNR starter as the preceding, but substitutes a Tirastat II controller for the

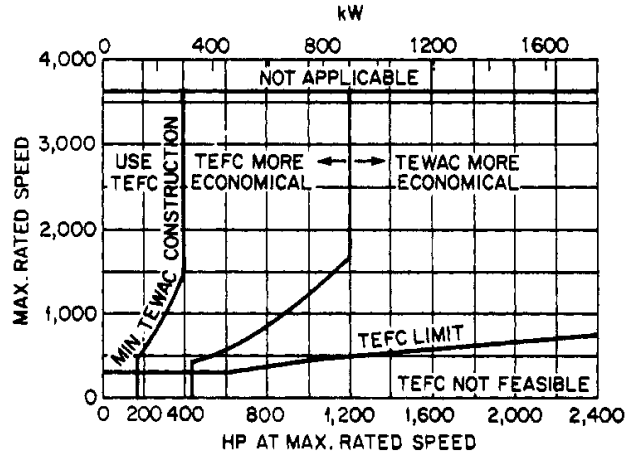


FIGURE 6B Approximate availability of totally enclosed horizontal and vertical squirrel-cage induction motors and horizontal wound-rotor induction motors. Wound-rotor induction motor speeds limited to 1800 rpm maximum (General Electric)

*Trademark of General Electric Co.