

Section 14

Fans, Pumps, and Compressors

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14.1 DISPLACEMENT PUMPS

by Terry L. Henshaw

REFERENCES: Buse, "Positive-Displacement Pumps," Marks' Handbook, 9th ed., McGraw-Hill. Henshaw, "Reciprocating Pumps," Van Nostrand Reinhold. "Hydraulic Institute Standards," 14 ed., Hydraulic Institute. Chesney, Water Injection—Pump Development, paper 68-Pet-11, ASME. Lees, Designing for High Pressures, paper 93-DE-9, ASME. "Positive Displacement Pumps—Reciprocating," API Standard 674, American Petroleum Institute.

NOTE: Much of the text and some of the illustrations are from the author's book: Henshaw, "Reciprocating Pumps," Van Nostrand Reinhold, 1987.

GENERAL

A **displacement pump** (also called **positive-displacement**, or just **p-d**) is a pump which imparts energy to the pumpage (the material pumped) by trapping a fixed volume at suction (inlet) conditions, compressing it to discharge pressure, then pushing it into the discharge (outlet) line. A displacement pump does not rely on velocity to achieve pumping action, as does a centrifugal pump or ejector.

Displacement pumps fall into two major classes: reciprocating and rotary, as illustrated in Fig. 14.1.1.

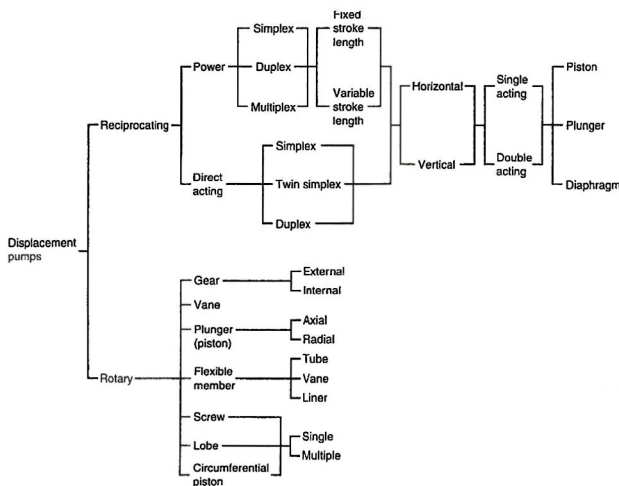


Fig. 14.1.1 Classification diagram of displacement pumps.

Uses and Applications

Displacement pumps serve primarily in applications of low capacity and high pressure, those mostly beyond the capabilities of centrifugal pumps. Some of these services could be performed by centrifugals, but not without an increase in power requirements and/or maintenance. Because displacement pumps achieve high pressures with low pumpage velocities, they are well-suited for abrasive-slurry and high-viscosity services. A reciprocating pump must have special fittings to be suitable for most slurries.

Net Positive Suction Head

Net positive suction head (NPSH), also called **net positive inlet pressure (NPIP)** and **net inlet pressure (NIP)**, is the difference between suction pressure and vapor pressure, at the pump suction nozzle, when the pump is running. In a reciprocating pump, NPSH is required to push the suction valve from its seat and to overcome the friction losses and acceleration head within the pump liquid end. In a rotary pump, NPSH

is required to push the pumpage into the cavities created by the pumping elements.

If sufficient NPSH is not provided by the system, the pumpage will begin to flash (boil) as it flows into the pump. The vapor will cause a deterioration of pump performance. As the vapor flows into regions of higher pressure in the pump, it condenses back to a liquid. This collapse of the vapor bubbles occurs with significant impact, impinging upon metal surfaces with enough energy to break out small pieces of the metal. This is called **cavitation** damage. The shock created by the bubble collapse may be severe enough to crack a fluid cylinder or break a crankshaft.

All pumps require the system to provide some NPSH. The NPSH provided by the system is called **NPSHA** (the A is for available). The NPSH required by the pump is called **NPSHR**. For displacement pumps, NPSHR is normally expressed in pressure units (lb/in² or kPa).

Since water usually contains dissolved air, the vapor pressure of the solution is higher than for deaerated water, but this is often overlooked when NPSH calculations are performed. The Hydraulic Institute recommends an NPSH margin of 3 lb/in² (20 kPa) for power pumps in systems where the pumpage has been exposed to a gas other than the liquid's own vapor. A liquid (such as propane) at its bubble point in the suction vessel requires no such margin.

RECIPROCATING PUMPS

A **reciprocating pump** is a displacement pump which reciprocates the pumping element (piston, plunger, or diaphragm). The capacity of a reciprocating pump is proportional to its speed, and is relatively independent of discharge pressure.

A **power pump** is one that reciprocates the pumping element with a crankshaft or camshaft (see Figs. 14.1.2 and 14.1.3). It requires a driver which has a rotating shaft, such as a motor, engine, or turbine. A constant-speed power pump will deliver essentially the same capacity at any pressure within the capability of the driver and the strength of the pump.

A **direct-acting** pump is a reciprocating pump driven by a fluid which has a differential pressure (see Fig. 14.1.4). The motive fluid pushes on

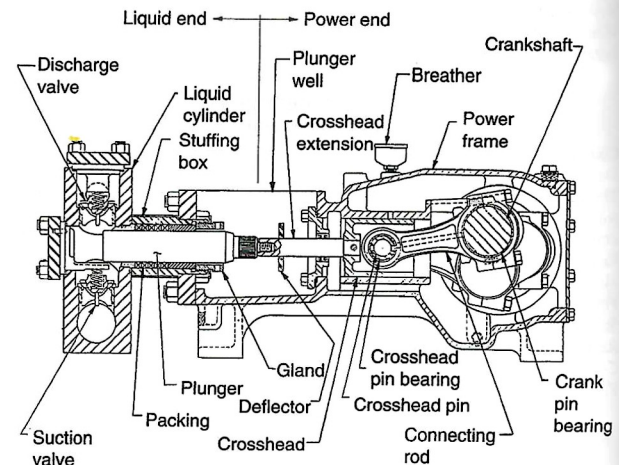


Fig. 14.1.2 Horizontal power pump.

piston (or diaphragm) which pushes the pumping element through a end (or directly on the pumpage).

Reciprocating pumps are classified by the following features:

Drive end (power or direct-acting)

Orientation of centerline of the pumping element (horizontal or vertical)

Number of discharge strokes per cycle of each drive rod (single-acting or double-acting)

Pumping element (piston, plunger, or diaphragm)

Number of drive rods (simplex, duplex, triplex, quintuplex, etc.)

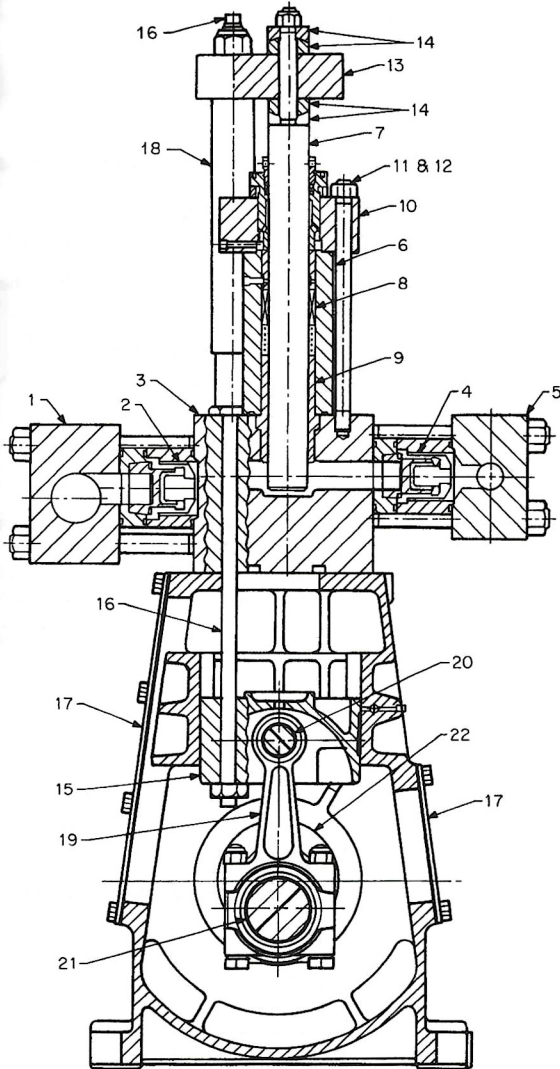


Fig. 14.1.3 Vertical power pump. (Ingersoll-Rand Company.)

Uses and Applications for Reciprocating Pumps

The justification for selecting a reciprocating pump instead of a centrifugal or rotary is cost, including costs of power and maintenance.

Reciprocating pumps are best suited for high-pressure/low-capacity services. Such services include high-pressure water-jet cleaning and cutting; glycol injection and charge, as well as amine and lean oil charge, in gas processing; ammonia and carbamate charging for fertilizer production; nuclear-reactor charging and standby control; oil-field

saltwater injection and disposal; oil well blowout preventers; pipelines (slurries and crude oil, and injection of ammonia and light hydrocarbons); steel and aluminum mill hydraulic systems; knockout drums in process plants; hydrostatic testing; process slurries; metering; food and chemical homogenizing; well-drilling mud; and car washes.

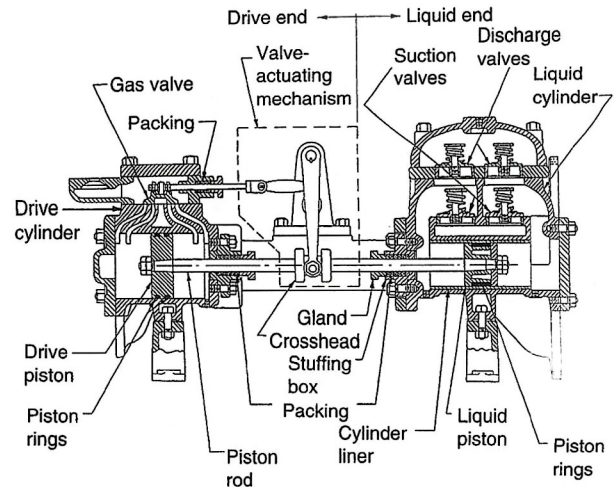


Fig. 14.1.4 Horizontal direct-acting gas-driven pump.

Applications that practically mandate reciprocating units are abrasive and/or viscous slurries above about 500 lb/in² (3500 kPa). Examples of these services range from powdered coal to peanut butter.

High-Pressure Applications

Hand-powered pumps, for pressures to 40,000 lb/in² (300 MPa), are used for small jacks, deadweight testers, and hydrostatic testing of small components. Small, air-driven direct-acting pumps are used for low-flow, high-pressure systems such as hydrostatic testing. The liquid pressure is approximately the air pressure multiplied ("intensified") by the ratio of the air piston area to the liquid plunger area. They are very simple and will stall and hold a fixed pressure without using power; but when pumping, they consume relatively large amounts of air. Available up to 10 hp (7 kW), the most popular sizes are up to 2 hp (1.5 kW). Pressures range to 100,000 lb/in² (700 MPa).

Hydraulically driven intensifiers operate on the same principle, but are more efficient, and because of the higher pressures commonly available from hydraulic systems, can produce high pressures with lower intensification ratios. Intensifiers are available to 300,000 lb/in² (2100 MPa) for laboratory-scale hydrostatic applications. Intensifier systems for water jetting are available from 30,000 to 60,000 lb/in² (210 MPa to 410 MPa) up to 200 hp (150 kW). They are powered by electric motors or diesel engines. Figure 14.1.5 shows the pumping end of an intensifier.

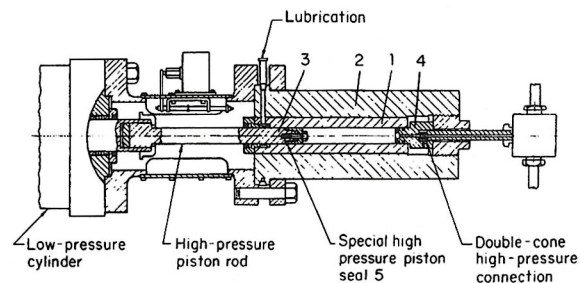


Fig. 14.1.5 Pumping end of a direct-acting, liquid-driven, "intensifier" pump. The check valves are in the tubing.

Power pumps (usually triplex or quintuplex) are available from about 5 to 150 hp (5 to 100 kW) with pressures to 40,000 lb/in² (280 MPa), and up to 2,000 hp (1,500 kW) at 20,000 lb/in² (140 MPa). Power pumps are efficient and mechanically simple. Figure 14.1.6 shows a power pump liquid end for a 36,000 lb/in² (250 MPa) water-jetting pump.

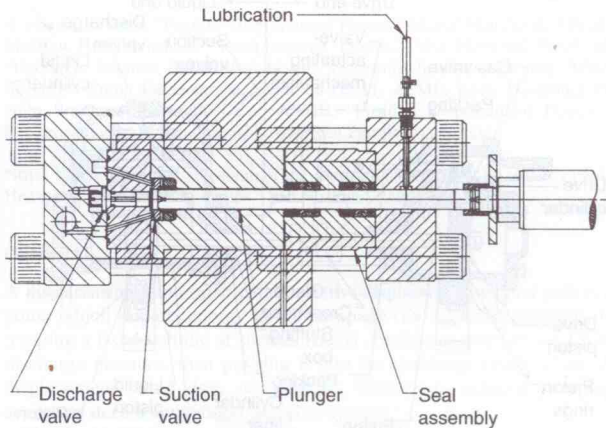


Fig. 14.1.6 Power pump liquid end for 36,000 lb/in² (250 MPa) water jetting. (NLB Corporation.)

Slurry Applications

The standard reciprocating pump is not designed to handle slurries. Modifications to standard designs, and in some cases special designs, are required to achieve satisfactory operation and component life.

To achieve satisfactory packing and plunger life, abrasive slurry must be prevented from entering the packing. Methods include a wiper ring between the pumpage and packing, a long throat bushing, injection of a clean liquid into the throat area, insertion of a diaphragm or floating piston between the plunger and the pumpage, and complete removal of the valve assembly from the stuffing box area. This last arrangement requires a liquid column between the valve assembly and the stuffing box, which increases the clearance volume and acceleration head within the pump.

Special pump valves are usually required for slurries. Depending on the nature of the solids, they can be ball, bell, bevel seat with elastomer insert, wing-guided with reduced seating area, or disk with special seating surfaces. Special construction can prevent the slurry from contacting the packing, but the valves cannot avoid contact with the slurry.

Problems and How to Avoid Them

Reciprocating pumps have some disadvantages, the most common being pulsating flow. Because of the pulsation, special consideration must be given to system design. Guidelines are provided later in this section.

In most applications, the initial and maintenance costs for a reciprocating pump will be greater than for a centrifugal or rotary pump. The packing in a typical power pump lasts about 2,500 hours, less than a mechanical seal on a rotating shaft.

Most problems with reciprocating pumps can be minimized by selecting pumps to operate at conservative speeds, by carefully designing the pumping system, by careful operating procedures, and by maintenance practices which preserve the alignment of the plunger (or rod) with the stuffing box.

Liquid-End Components

All reciprocating pumps contain one or more pumping elements (pistons, plungers, or diaphragms) that reciprocate into and out of the pumping chambers. (In reciprocating pump terminology, a **piston** is a cylindrical disk which mounts on a rod. A **plunger** is just a smooth rod.)

Each pumping chamber contains at least one suction and one discharge valve.

The liquid cylinder is the major pressure-retaining part of the liquid end, and forms the major portion of the pumping chamber.

A piston pump is normally equipped with a replaceable liner (sleeve) that absorbs the wear from the piston rings. Because a plunger contacts only stuffing-box components, plunger pumps do not require liners.

Sealing between the pumping chamber and atmosphere is accomplished by a **stuffing box** (Fig. 14.1.7). The stuffing box contains rings of packing that conform to and seal against the stuffing box bore and the rod (or plunger).

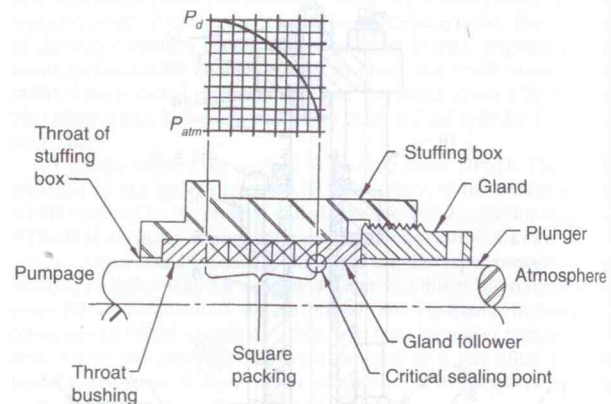


Fig. 14.1.7 Reciprocating pump stuffing box (nonlubricated) showing pressure gradient during discharge stroke.

The valves in a reciprocating pump are simply check valves which are opened by the liquid differential pressure. Most valves are springloaded. They have a variety of shapes, including spheres, hemispheres, and disks. Disks may be center-guided, outside-diameter-guided, or wing-guided. Figures 14.1.8 through 14.1.13 illustrate some of the valves used in reciprocating pumps.

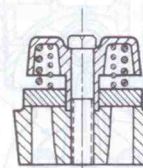


Fig. 14.1.8 Reciprocating pump disk valve. (Ingersoll-Rand Company.)

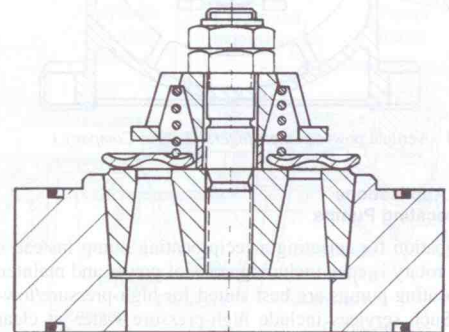


Fig. 14.1.9 Reciprocating pump disk valve. (Durabla Manufacturing Company.)

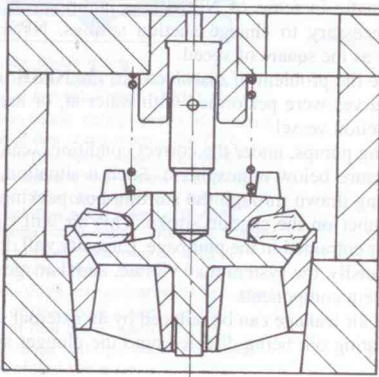


Fig. 14.1.10 Reciprocating pump elastomeric-insert valve. (TRW-Mission Manufacturing Co.)

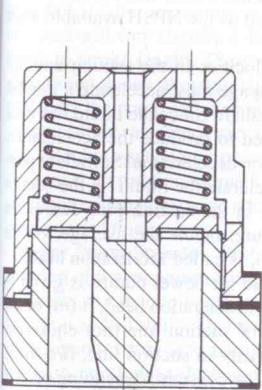


Fig. 14.1.11 Reciprocating pump double-ported disk valve.

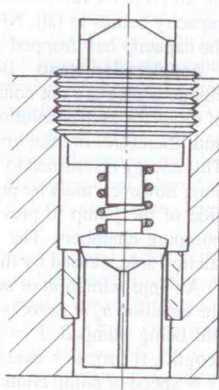


Fig. 14.1.12 Reciprocating pump conical-faced (bevel-seat) wing-guided valve.

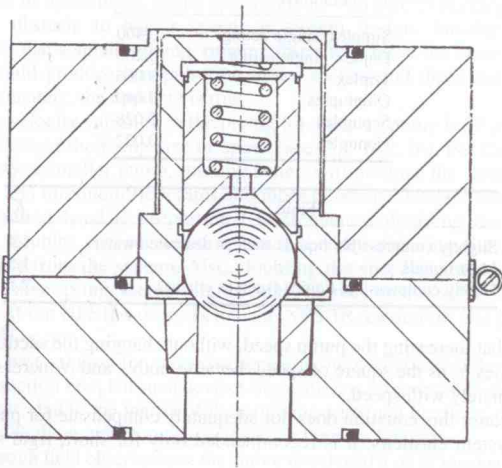


Fig. 14.1.13 Reciprocating pump ball valve. (Ingersoll-Rand Company.)

The Pumping Cycle

As the pumping element reverses from the discharge stroke to the suction stroke, liquid within the pumping chambers expands, and the pressure drops. Since most liquids are relatively incompressible, little movement of the element is required to reduce the pressure. When the

pressure drops sufficiently below suction pressure, the differential pressure (suction pressure minus chamber pressure) pushes the suction valve open. This occurs when the element is moving slowly, so that the valve opens slowly and smoothly as the velocity of the element increases. Liquid then flows through the valve and follows the element on its suction stroke. As the element decelerates near the end of the stroke, the suction valve returns to its seat. When the element stops, the suction valve closes.

The pumping element then reverses and starts its discharge stroke. The liquid trapped in the pumping chamber is compressed until chamber pressure exceeds discharge pressure by an amount sufficient to push the discharge valve from its seat. The action of the discharge valve, during the discharge stroke, is then the same as that of the suction valve during the suction stroke.

Power Pumps

The drive end of a power pump is called a **power end** (see Fig. 14.1.2). Its function is to convert rotary motion from a driver into reciprocating motion for the liquid end. The main component of the power end is the **power frame**, which supports all other power end parts and, usually, the liquid end. The second major component of the power end is the **crankshaft** (sometimes a camshaft). The crankshaft transmits power from the driver to the connecting rods.

The **connecting rod** is driven by a throw (journal) of the crankshaft on one end, and drives a crosshead on the other. The crankshaft moves in pure rotating motion, the crosshead in pure reciprocating motion. The connecting rod is the link between the two. **Main bearings** support the shaft in the power frame.

The **crosshead** transmits the force from the connecting rod into the pumping element. It is fastened directly to a plunger or piston rod, or to a rod called an **extension, stub, or pony rod**. The other end of this rod is fastened to the pumping element.

Power pump overall efficiencies normally range from 85 to 94 percent, higher than any other type of pump.

Selection Graphs Different manufacturers produce these graphs in different configurations. All graphs consist of straight lines since displacement is directly proportional to speed. Figure 14.1.14 shows one such graph for a triplex power pump. Displacement is plotted against crankshaft speed for the different plunger diameters available in the pump. The pressures listed are those at which each plunger size would operate to load the power end to its rated value (4,500 lbf or 20,000 N).

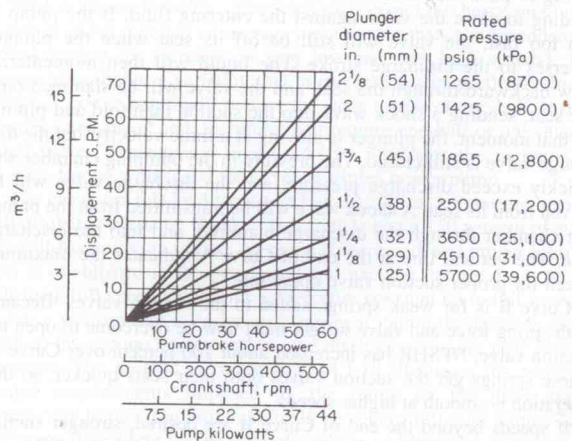


Fig. 14.1.14 Power pump selection graph. (Ingersoll-Rand Company.)

NPSH Curves Figure 14.1.15 provides insight into power pump NPSHR characteristics, valve action, and the effect of valve springs on pump performance. These curves are for a 3-in (76-mm) stroke, horizontal, triplex plunger power pump with 2 1/4-in-diameter (57-mm) plungers, and with suction valves that operate vertically. The valves are wing-guided, and have a diameter approximately equal to the plunger diameter.

Because the axis of the suction valve is vertical, the valve can operate without a spring if the pump speed is kept low. Curve A represents the NPSH requirements with no springs on the suction valves. NPSHR at 100 r/min is only 0.7 lb/in² (5 kPa) (1.6 ft or 0.5 m of water), less than for most centrifugals.

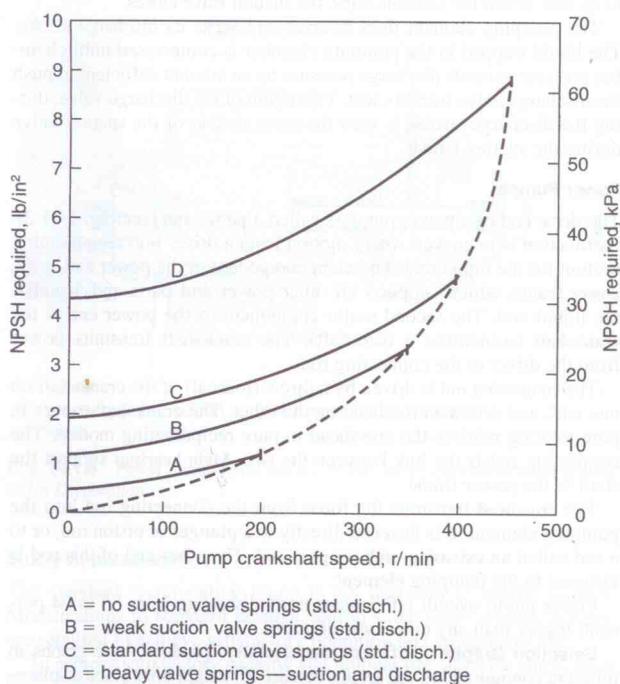


Fig. 14.1.15 NPSH curves for a 3-in. (75-mm-) stroke triplex power pump.

The speed of the pump in this configuration is limited by the ability of the suction valve to keep up with the plunger. Since there is no spring to push the valve back onto its seat, gravity is the only force tending to close the valve against the entering fluid. If the pump is run too fast, the valve will still be off its seat when the plunger reverses to the discharge stroke. The liquid will then momentarily flow backward through the seat, and the valve will be slammed onto the seat, sending a shock wave into the suction manifold and piping. At that moment, the plunger is moving at a finite velocity, but the discharge valve is still closed. The pressure in the pumping chamber will quickly exceed discharge pressure, and the discharge valve will be driven from its seat. A shock wave will be transmitted from the pumping chamber through the discharge manifold, and into the discharge line. The vertical line at the end of Curve A indicates the maximum speed for proper suction valve operation.

Curve B is for weak springs added to the suction valves. Because both spring force and valve weight must now be overcome to open the suction valve, NPSHR has increased about 100 percent over Curve A. These springs get the suction valves onto their seats quicker, so that operation is smooth at higher speeds.

If speeds beyond the end of Curve B are desired, stronger suction valve springs are required. Stronger springs allow operation to 400 r/min. NPSH requirements are about 3 times those of Curve A, ranging from about 2 to 5 lb/in² (14 to 35 kPa).

Curves A through C represent a pump equipped with the same (standard) discharge valve spring. Only the suction spring has been changed.

If operation is required at speeds exceeding the limits of Curve C, extra-strong springs are required on both suction and discharge valves. As shown by Curve D, NPSHR is about double that required for the standard springs, ranging from about 4.5 to 9 lb/in² (31 to 62 kPa).

The dashed line passing through the ends of the NPSH curves illustrates the dramatic increase of NPSHR as pump speed is increased. When it is necessary to change suction springs, NPSHR increases approximately as the square of speed.

To minimize the problem of dissolved air, the NPSH tests that produced these curves were performed with water at, or near, its boiling point in the suction vessel.

Reciprocating pumps, under the correct conditions, can operate with a suction pressure below atmospheric. Such a situation, though, can lead to air being drawn through the stuffing box packing and into the pumping chamber on the suction stroke. This air will cause as many problems as air entrained in the pumpage. Capacity will drop, the pump may operate noisily, the system may vibrate, and damage may occur to pump and system components.

This inward air leakage can be reduced by an external sealing liquid, such as lubricating oil, being directed onto the plunger surface or into the packing.

Test Criteria for NPSH Power pump NPSH tests are performed by holding the pump speed and discharge pressure constant, and varying the NPSH available (NPSHA) in the system. Capacity remains constant for all NPSHA values above a certain point. Below this NPSHA value, capacity begins to fall. NPSHR is defined as the NPSH available when the capacity has dropped 3 percent.

Acceleration Head Because the velocities in the suction and discharge piping are not constant, the pumpage must accelerate a number of times for each revolution of the crankshaft. Since the liquid has mass and, therefore, inertia, energy is required to produce the acceleration. This energy is returned to the system upon deceleration. Sufficient pressure, however, must be provided to accelerate the liquid on the suction side of the pump to prevent cavitation in the suction pipe and/or the pumping chambers. The drop in pressure, below the average, on the suction side, caused by this acceleration, is called **acceleration head**.

An approximation of acceleration head for power pumps is given by the equation $h_a = LvNC/gK$, where h_a = acceleration head, ft (m) of liquid being pumped; L = actual length of suction line (not equivalent length), ft (m); v = average liquid velocity in suction line, ft/s (m/s); N = speed of pump crankshaft, r/min; C = constant, depending on type of pump; g = gravitational constant, ft/s² (m/s²); and K = fluid compressibility correction factor. Values for the constants C and K are:

Pump type	C
Simplex, single-acting	0.400
Duplex, single-acting	0.200
Triplex	0.066
Quintuplex	0.040
Septuplex	0.028
Nonuplex	0.022

Fluid	K
Slightly compressible liquids such as deaerated water	1.4
Most liquids	1.5
Highly compressible liquids (such as ethane)	2.5

Note that increasing the pump speed, without changing the suction line, increases h_a as the square of speed, because both v and N increase proportionately with speed.

Because this equation does not adequately compensate for pumpage and system elasticity, it is recommended only for short, rigid suction lines.

To reduce acceleration head in a suction line, a bottle or suction stabilizer may be installed in the pipe, adjacent to the pump.

Figure 14.1.16 shows the theoretical flow rates for various types of power pumps during one revolution of the crankshaft. Pumps with an odd number of cranks have the same flow variations for both single-acting and double-acting liquid ends (except the simplex). Flow curves for septuplex and nonuplex pumps are similar to the quintuplex curve, with

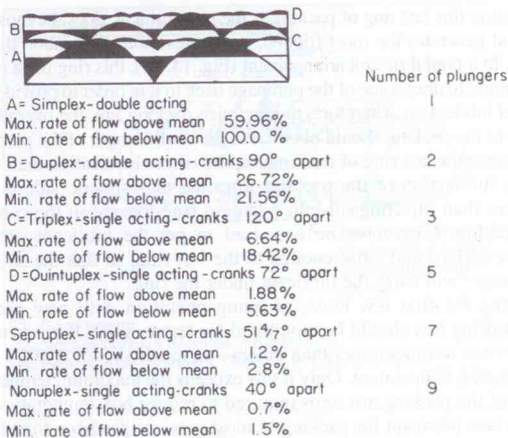


Fig. 14.1.16 Flow-rate variations in reciprocating power pumps.

more but smaller variations, as indicated in the tabulation. The values tabulated will vary slightly with variations in the ratio of connecting rod length to stroke length. These curves indicate flow-rate variations, not pressure pulsations. Pressure pulsations are a product of the acceleration created by the pump and the mass of the pumpage in the system.

Actual flow rates may vary considerably from these curves, particularly with high-speed pumps, where valve closing and opening may lag the crankshaft rotation as much as 15°, and at high pressures, where liquid compression may cause a further lag of 5° to 10° (or more for a large clearance volume).

Power Pump Speeds Probably the most controversial factor in the selection of power pumps is the maximum allowable speed. Some manufacturers, for example, offer 3-in. (76-mm-) stroke triplex pumps for operation at 500 r/min. One builder of portable water-jetting units routinely operates this size pump at 600 r/min. Most users of this size pump, in continuous-duty applications, prefer to operate it no faster than about 400 r/min.

Top speeds quoted by one manufacturer for the 3-in.-stroke triplex increased over a period of years from 150 to 520 r/min. Although improvements in power end and liquid end construction made the pump capable of operating at higher speeds, albeit with shorter packing life, a major obstacle to higher speeds is system design. Initially, some thought that a smaller pump, running faster to achieve the same capacity, would produce less pulsation in the suction and discharge pipes. Unfortunately, the opposite is true.

The velocity variation in the piping for a triplex pump is 25 percent, regardless of the pump size or speed (see Fig. 14.1.16). For the same capacity, a smaller pump, running faster, will produce the same maximum and minimum flow rates and more pulses per second. Since the acceleration head is proportional to frequency, doubling the pump speed doubles the acceleration head, thereby reducing the NPSH available from the system. Also, doubling the speed requires a much stiffer valve spring, thereby significantly increasing the NPSHR of the pump. If the NPSHA drops below the NPSHR, cavitation and pounding will occur.

The best solution for excessive acceleration is an effective dampener in the suction line; but such devices often are less effective at higher frequencies, due to the inertia of their moving parts and inertia of the liquid inside them, which must oscillate for them to function.

Through field observations, the author developed a set of **maximum recommended speeds** for plunger-type power pumps in continuous-duty services. These speeds, as adopted by API 674, are shown in Fig. 14.1.17, along with the Hydraulic Institute basic speeds. Intermittent and cyclic operation is often satisfactory above these speeds. Lower speeds may be dictated by factors such as pump construction, system design, low NPSHA, entrained gas, high temperature, entrained solids, high viscosity, and requirements for a low sound level.

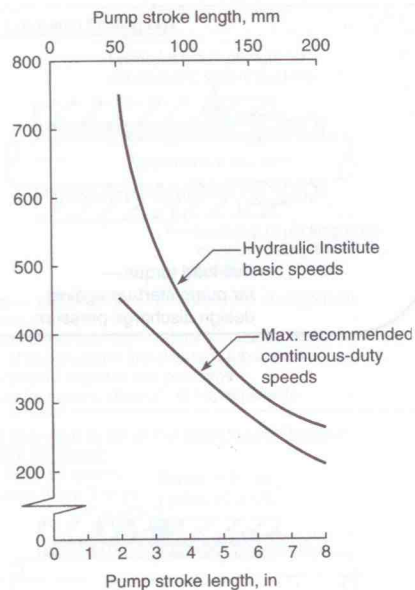


Fig. 14.1.17 Maximum recommended speeds for plunger-type power pumps in continuous-duty services.

The **minimum speed** of a power pump is determined by its ability to provide sufficient lubrication to all bearing surfaces in the power end. Some units can run satisfactorily at 20 r/min. Others must be maintained at 100 r/min or more.

Plunger load (also called **frame load** and **rod load**) is the force transmitted to the power end by one plunger. For a single-acting pump, the discharge plunger load is calculated by multiplying discharge pressure by the cross-sectional area of one plunger. Suction plunger load is the suction pressure multiplied by the plunger area.

A power pump is rated by the maximum discharge plunger load that the power end is capable of absorbing when the suction pressure is zero. Some units are rated for continuous operation, some for intermittent operation, and some are rated both ways.

Torque Characteristics For fixed suction and discharge pressures, a power pump requires an **average** input torque that is independent of speed (except for increases at very low and very high speeds). The torque actually has a variation that mirrors the discharge flow-velocity curve. A power pump will require the same torque at one-half or one-quarter rated speed and, therefore, will require one-half or one-quarter rated power, respectively. Figure 14.1.18 illustrates the variation of average torque with speed for a typical triplex power pump.

The curve for full-load torque in Fig. 14.1.18 is for starting against full discharge pressure. Breakaway torque is about 150 percent of average full-load running torque. As speed is increased, and proper lubrication is established in the power end and packing, torque drops to the full-load, full-speed value, and is thereafter constant up to full speed.

For a start-up that is easier on equipment, pump discharge is piped back to the suction vessel, making the discharge pressure near suction pressure. The no-load curve in Fig. 14.1.18 illustrates the resulting torque requirements imposed by the pump on the drive train. Breakaway torque is approximately 25 percent of full-load torque. This will vary, depending on the type of packing and bearings in the pump and the length of time the pump has been idle. As speed increases, the torque drops to less than 10 percent of the full-load torque.

Packing The component normally requiring the most maintenance on reciprocating pumps is packing. Although the life of packing in a power pump is typically about 2,500 h (3 months), some installations, with special stuffing box arrangements, have experienced lives of more than 18,000 h (2 years), at discharge pressures to 4,000 lb/in² (28 MPa).

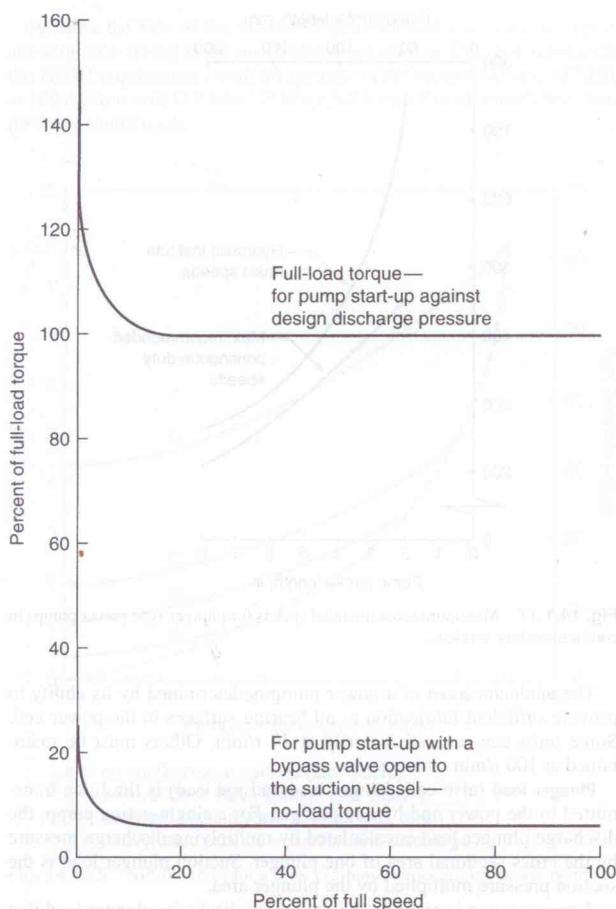


Fig. 14.1.18 Torque-speed curves for a typical triplex power pump.

Short packing life can result from any of the following conditions: (1) misalignment of plunger (or rod) with stuffing box; (2) worn plunger, rod, stuffing-box bushings, or stuffing-box bore; (3) improper packing for the application; (4) insufficient or excessive lubrication; (5) packing gland too tight or too loose; (6) excessive speed or pressure; (7) high or low temperature of pumpage; (8) excessive friction (too much packing); (9) packing running dry (pumping chamber gas-bound); (10) shock conditions caused by entrained gas or cavitation, broken or weak valve springs, or system problems; (11) solids from the pumpage, environment, or lubricant; (12) improper packing installation or break-in; and (13) icing caused by volatile liquids that refrigerate and form ice crystals on leakage to atmosphere, or by pumpage at temperatures below 32°F (0°C). As is evident from this list, short packing life can indicate problems elsewhere in the pump or system.

To achieve a low leakage rate, the clearance between the plunger (or rod) and packing must be essentially zero. This requires that the sealing rings be relatively soft and pliant. Because the packing is pliant, it tends to flow into the stuffing box clearances, especially between the plunger and follower bushing. If this bushing does not provide an effective barrier, the packing will extrude, and leakage will increase.

A set of square or V-type packing rings will experience a pressure gradient, during operation, as indicated in Fig. 14.1.7. The last ring of packing, adjacent to the gland-follower bushing, will experience the largest axial loading, resulting in greater deformation, tighter sealing and, therefore, the largest pressure drop. The gap between the plunger and the follower must be small enough to prevent packing extrusion. Most packing failures originate at this critical sealing point.

Because this last ring of packing is the most critical, does the most sealing, and generates the most friction, it requires more lubrication than the others. In a nonlubricated arrangement (Fig. 14.1.7), this ring must rely on the plunger to drag some of the pumpage back to it in order to provide cooling and lubrication. Therefore, to maximize packing life, the overall stack height of the packing should not exceed the stroke length of the pump.

Because the last ring of packing requires more lubrication than do the others, lubrication of the packing from the atmospheric side is more effective than injecting oil into a lantern ring located in the center of the packing. Care must be exercised to get the lubricant onto the plunger surface and close enough to the last ring, so that the stroke of the plunger will carry the lubricant under the ring.

During the first few hours of pump operation with new packing, each stuffing box should be monitored for temperature. It is normal for some boxes to run warmer than others—as much as 50°F (30°C) above the pumping temperature. Only if this exceeds the maximum temperature rating of the packing are steps required to reduce box temperature.

The best lubricant for packing in most services has been found to be steam cylinder oil.

The concepts that a higher discharge pressure requires more rings of packing and that a larger number of rings lasts longer are true for long-stroke, low-speed machines, but have been disproven in a number of moderate- to high-speed power-pump applications. Unless they are profusely lubricated, the larger number of rings create additional frictional heat and wipe lubricant from the plunger surface, thus depriving some rings of lubrication. On numerous saltwater injection pumps operating at pressures above 4,000 lb/in² (28 MPa), Chesney reported that packing life was only two weeks with twelve rings of packing in each stuffing box. With three rings in each box, packing life was approximately 6 months.

Plungers Next to packing, the plunger is the component of a power pump that typically requires most frequent replacement. The high speed of the plunger and the friction load of the packing tend to wear the plunger surface. For longer life, plungers are sometimes hardened, although it is more common to apply a hard coating. Such coatings are of chrome, various ceramics, nickel-based alloys, or cobalt-based alloys. Desired characteristics of the coatings include hardness, smoothness, high bond strength, low porosity, corrosion resistance, and low cost. No one coating optimizes all characteristics.

Ceramic coatings are harder than the metals, but are brittle, porous, and sometimes lower in bond strength. Porosity contributes to shorter packing life. Mixing of hard particles, such as tungsten carbide, into the less-hard nickel or cobalt alloys has resulted in longer plunger life at the expense of shorter packing life.

Stuffing Boxes Various stuffing box designs (various lubrication and bleed-off arrangements to minimize leakage and extend packing life) are shown in Fig. 14.1.19.

The most significant advance in packing arrangements in recent years has been spring loading. Spring loading is applied most to V-ring (chevron) packing but also works well with square packing. The spring must be located on the pressure side of the packing. Springs of various types can be used, including single-coil, multiple coil, wave-washer, belleville, and thick-rubber-washer. The force provided by the spring is small compared to the force imposed on the packing by discharge pressure. The major functions of the spring are to provide a small preload to help set the packing, and to hold all bushings and packing in place during operation.

Spring loading of packing has the following advantages:

Requires no adjustment of the gland. The gland is tightened until it bottoms, then is locked. This removes one of the biggest variables in packing life: personnel skill.

Alloys expansion. If the packing expands during the initial break-in, the spring allows for the expansion.

Adjusts for wear. As the packing wears, adjustment automatically occurs from inside the box. The problem of transmitting gland movement through the top packing ring is eliminated.

Provides a cavity. The spring cavity provides an annular space for the injection of a clean liquid for slurry applications.

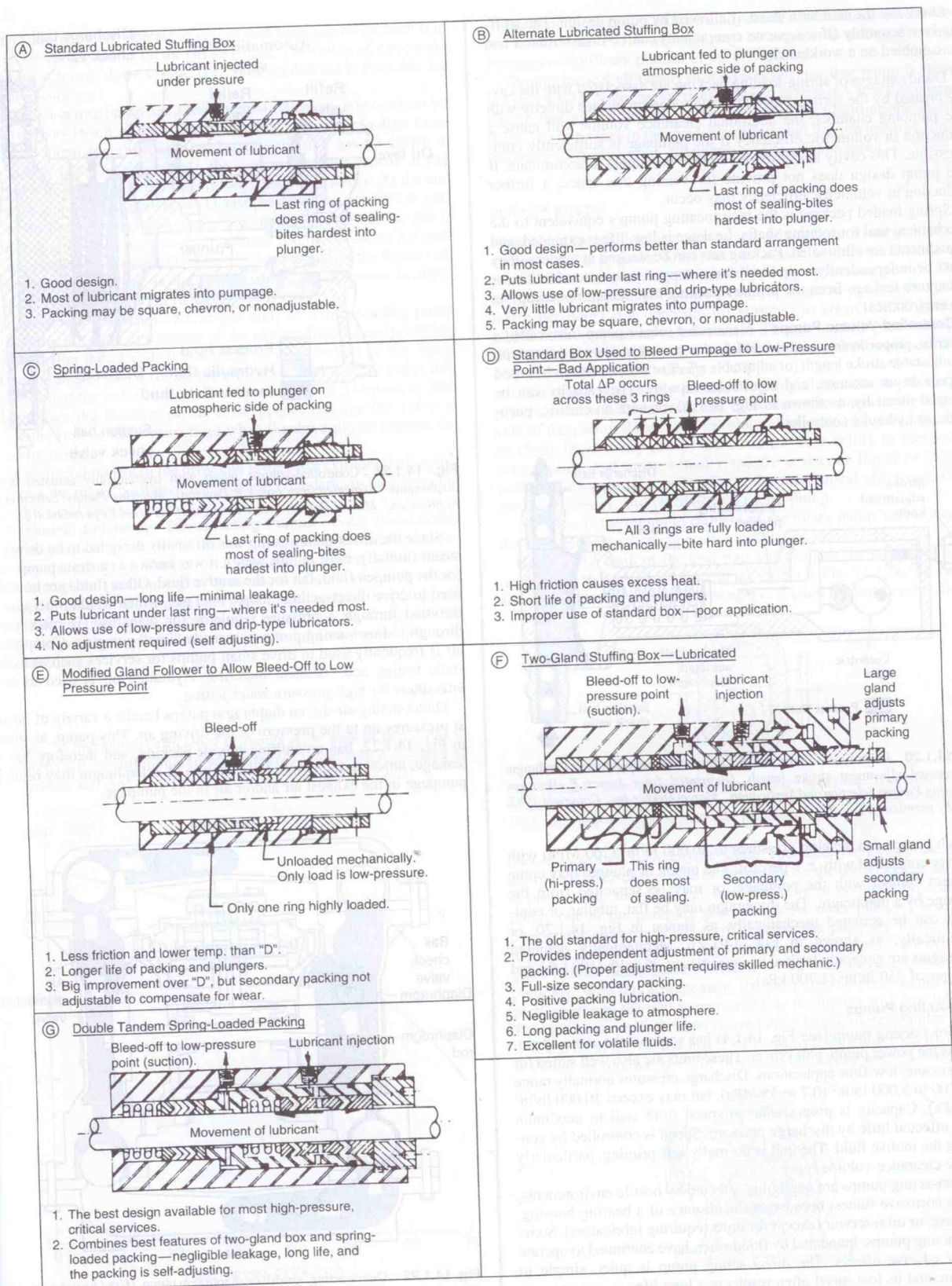


Fig. 14.1.19 Various stuffing box designs for reciprocating pumps.

Eliminates the need for a gland, if allowed by pump design. The stuffing box assembly (if a separate component) can be disassembled and reassembled on a workbench.

Disadvantages of spring-loading packing are associated with the cavity created by the spring. Since this cavity communicates directly with the pumping chamber, the additional clearance volume will cause a reduction in volumetric efficiency if the pumpage is sufficiently compressible. This cavity also provides a place for vapors to accumulate. If the pump design does not provide for venting this space, a further reduction in volumetric efficiency may occur.

Spring-loaded packing is the reciprocating pump's equivalent to the mechanical seal for rotating shafts. Leakage is low, life is extended, and adjustments are eliminated. Packing sets can be stacked in tandem (they must be independently supported) for a stepped pressure reduction, or to capture leakage from the primary packing that should not escape to the environment.

Controlled-Volume Pumps A controlled-volume pump (also called a metering, proportioning, or chemical-injection pump) is a power pump with an adjustable stroke length (or adjustable effective stroke length). It is used to provide an accurate, and adjustable, capacity. The capacity can be changed manually, as shown in Fig. 14.1.20, or with an electric, pneumatic, or hydraulic controller.

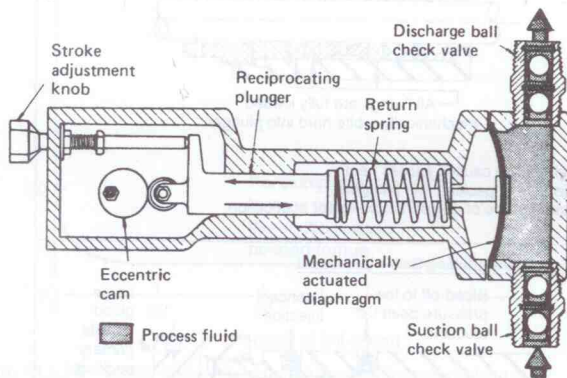


Fig. 14.1.20 Controlled-volume pump with mechanically actuated diaphragm and manual-adjustment stroke length. (Reprinted from James P. Poynton, "Metering Pumps: Selection and Application," Marcel Dekker, Inc., Copyright 1983. Used by permission.)

Such pumps can develop pressures to 30,000 lb/in² (200 MPa) with capacity controlled with ± 1 percent. The piston or plunger may come in direct contact with the pumpage, or may be separated from the pumpage by a diaphragm. The diaphragm may be flat, tubular, or conical. It can be actuated mechanically, as shown in Fig. 14.1.20, or hydraulically, as shown in Fig. 14.1.21. Mechanically actuated diaphragms are generally limited to capacities of 25 gal/h (100 L/h) and pressures of 250 lb/in² (1,700 kPa).

Direct-Acting Pumps

The direct-acting pump (see Fig. 14.1.4) has some of the same advantages as the power pump, plus others. These units are also well suited for high-pressure, low-flow applications. Discharge pressures normally range from 100 to 5,000 lb/in² (0.7 to 35 MPa), but may exceed 10,000 lb/in² (70 MPa). Capacity is proportional to speed from stall to maximum speed, affected little by discharge pressure. Speed is controlled by controlling the motive fluid. The unit is normally self-priming, particularly the low-clearance-volume type.

Direct-acting pumps are negligibly affected by hostile environments, such as corrosive fumes, because of the absence of a bearing housing, crankcase, or oil reservoir (except for units requiring lubricators). Some direct-acting pumps, inundated by floodwater, have continued to operate without adverse effects. The direct-acting pump is quiet, simple to maintain, and its low speed often results in a long life.

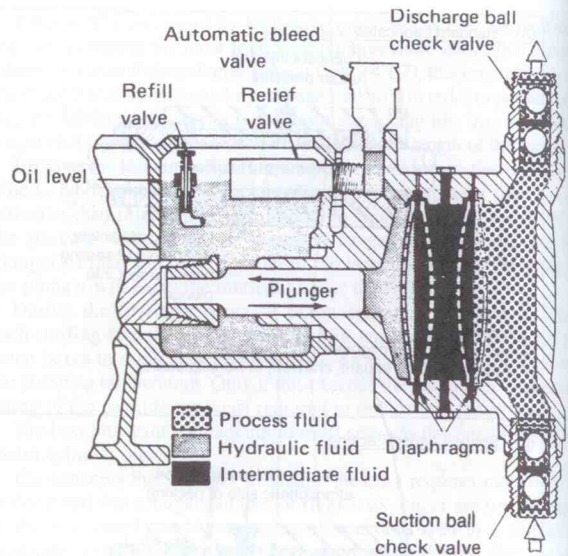


Fig. 14.1.21 Controlled-volume pump with hydraulically actuated double diaphragms. (Reprinted from James P. Poynton, "Metering Pumps: Selection and Application," Marcel Dekker, Inc., Copyright 1983. Used by permission.)

Since the direct-acting pump was originally designed to be driven by steam (initially, most pumps were), it was known as a **steam pump**—not for the pumped fluid, but for the motive fluid. Other fluids are now also used to drive direct-acting pumps. Fuel gas, which would otherwise be throttled through a pressure regulator for plant use, is often piped through a direct-acting pump to provide "free" pumping. Compressed air is frequently used to drive small pumps for services such as hydrostatic testing and chemical metering. Hydraulic oil is used to drive intensifiers for high-pressure water-jetting.

Direct-acting air-driven diaphragm pumps handle a variety of liquids at pressures up to the pressure of the driving air. This pump, as shown in Fig. 14.1.22, has no stuffing box or packing, and therefore has no leakage, unless a diaphragm fails. Failure of a diaphragm may result in pumpage in the exhaust air and/or air in the pumpage.

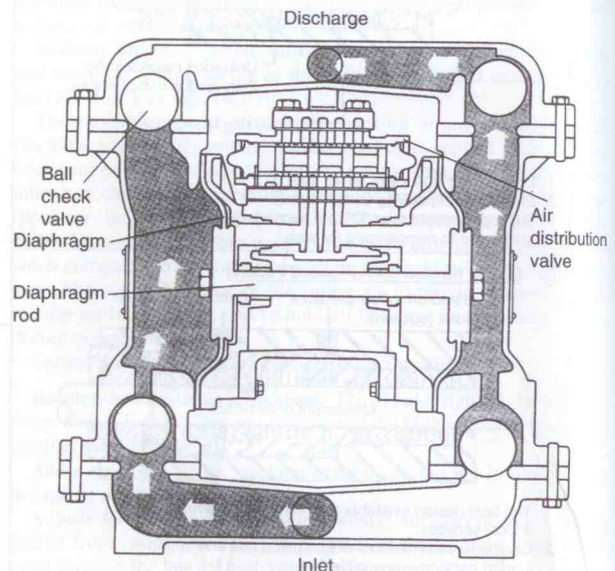


Fig. 14.1.22 Direct-acting, air-driven diaphragm pump. (Used by permission of Warren Rupp, Inc., a unit of IDEX Corp., Mansfield, OH.)

Direct-acting pumps operate at speeds that normally range from 0 to 50 cycles/min, depending on the stroke length. The ability of these units to operate at speeds down to zero (i.e., stall conditions) is desirable for some applications.

The direct-acting pump has a low thermal efficiency when driven by a gas (such as steam or air). The mechanical efficiency (output force divided by input force) is high; but because the unit has no device to store energy (such as a flywheel), the motive gas must remain at full inlet pressure in the drive cylinder through the entire stroke. At the end of the stroke, the gas expands to exhaust pressure, but no work is performed during this expansion. Hence, the thermal energy of the gas is lost to friction. Steam pumps consume about 100 lb/h of steam for each hydraulic horsepower (hhp) (60 kg/h · kW) developed on the liquid end. When natural gas or air is the motive fluid, consumption is about 3,500 std. ft³/h · hhp (130 m³/h · kW).

The **drive end** (or steam end, or gas end) of a direct-acting pump converts the differential pressure of the motive fluid to reciprocating motion for the liquid end. It is similar in construction to the liquid end, containing a piston (or diaphragm) and valving. The major difference is that the valve is mechanically actuated by a control system which senses the location of the drive piston, causing the valve to reverse the flow of the motive fluid when the drive piston reaches the end of its stroke.

The main component of the drive end is the drive cylinder. This cylinder forms the major portion of the pressure boundary, and supports the other drive end parts.

Mechanical Efficiency—Direct-Acting Pumps For a direct-acting pump, mechanical efficiency is the ratio of the force transmitted to the liquid by the piston (or plunger or diaphragm) to the force transmitted to the drive piston (or diaphragm) by the motive fluid. For double-acting pumps, the differential pressures are used at both ends of the pump, and since fluid friction losses in valves and ports must be charged to the pump, the pressures are as measured at the inlet and outlet ports. In equation form:

$$\eta_m = \frac{A_L \Delta p_L}{A_{dr} \Delta p_{dr}} \quad (14.1.1)$$

where η_m = mechanical efficiency of pump, A_L = area of liquid piston or plunger, Δp_L = differential pressure across liquid end, A_{dr} = area of drive piston, and Δp_{dr} = differential pressure across drive end.

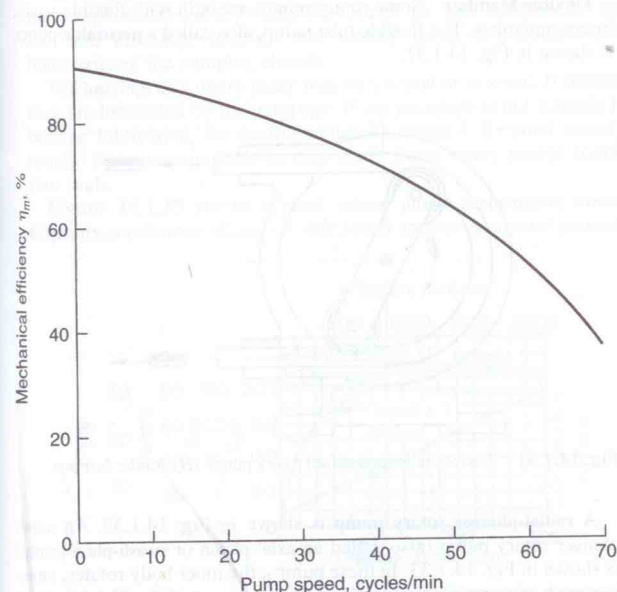


Fig. 14.1.23 Mechanical efficiency of a direct-acting pump.

The area of the piston rod is usually small relative to the piston area, and is often ignored. However, it must be considered when the rod area becomes a significant portion of the piston area.

The mechanical efficiency of a typical direct-acting pump is shown in Fig. 14.1.23. Note that η_m rises as speed is reduced. It is this characteristic that allows the unit to be controlled by throttling the motive fluid. The reduction in the available differential driving pressure forces the pump to operate more efficiently, i.e., at a lower speed.

ROTARY PUMPS

Rotary pumps are displacement pumps which have rotating pumping elements, such as gears, lobes, screws, vanes, or rollers. They do not contain inlet and outlet check valves, as do reciprocating pumps. They are built for capacities from a fraction of a gallon per minute (cubic metre per hour) (for domestic oil burners) to about 15,000 gal/min (3,000 m³/h) (for marine cargo service). Though used for pressures up to 20,000 lb/in² (140 MPa), their particular field is for pressures of 25 to 500 lb/in² (170 to 3,500 kPa).

Most rotary pumps rely on close running clearances to prevent the pumpage from leaking from the discharge side back to the suction side of the pump. Because of the close clearances, the pumpage must be clean. Most rotary pumps are noted for their ability to handle viscous liquids, and many actually *require* a viscous liquid to achieve peak performance. Viscosity affects the mechanical efficiency, volumetric efficiency, and NPSHR of a rotary pump.

The **rotor** is the pumping element of the rotary pump, and is usually the feature by which the pump is classified.

Gears The teeth of the gear trap and displace the pumpage. If the teeth are on the outside of the gear, as shown in Fig. 14.1.24, it is called an **external gear**. As the teeth disengage, suction pressure pushes liquid

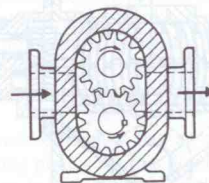


Fig. 14.1.24 External-gear rotary pump. (Reprinted from "Hydraulic Institute Standards for Centrifugal, Rotary and Reciprocating Pumps," 14th ed., copyright 1983, with the permission of Hydraulic Institute.)

into the cavities between the teeth. The teeth then carry the liquid around to the discharge side of the pump. As the teeth engage, the fluid is pushed into the discharge line. In the most common type of gear pump, one of the pumping gears drives the other pumping gear. This direct contact between the teeth requires the pumpage to be a good lubricant. In some units, though, the two pumping gears are driven by gears (called **timing gears**) mounted on the two shafts external to the pumpage. With this arrangement, the two pumping gears do not contact each other, and the unit is more suitable for liquids with low lubricity. This arrangement does, however, require a larger, more complex pump.

If the teeth are on the inside of a ring, as shown in Fig. 14.1.25, it is called an **internal gear**. In this unit, the larger internal-tooth gear

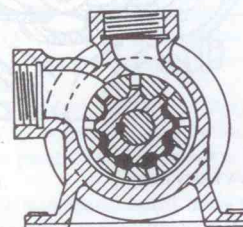


Fig. 14.1.25 Internal-gear rotary pump.

drives the smaller external-tooth gear. Both gears transfer liquid from the suction to the discharge.

Both external- and internal-gear rotary pumps are used in lubrication systems of engines, compressors, and larger pumps.

Lobe pumps are similar in construction and pumping action to external gear pumps, but one lobe does not drive the other. They do not even touch each other, and therefore must have their shafts independently driven by external timing gears. The lobes are often made of elastomers, and operate at low speeds. These units are used to transfer delicate items such as cherries and other foods—and even live fish. Figure 14.1.26 shows a rotary pump with three-lobe rotors.

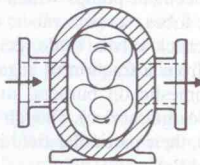


Fig. 14.1.26 Three-lobe rotary pump. (Reprinted from "Hydraulic Institute Standards for Centrifugal Rotary and Reciprocating Pumps," 14th ed., copyright 1983, with the permission of Hydraulic Institute.)

Screw pumps are constructed with one, two, or three screws. The single-screw pump is more commonly called a progressing-cavity pump, and is discussed separately below. The two-screw pump, as illustrated by Fig. 14.1.27, is used to handle liquids with viscosities to greater than 1,000,000 SSU (200,000 mm²/s), with capacities to about 15,000 gal/min (3,000 m³/h), pressures to 3,500 lb/in² (25 MPa), and temperatures to 600°F (300°C). It is particularly suited for high-viscosity liquids.

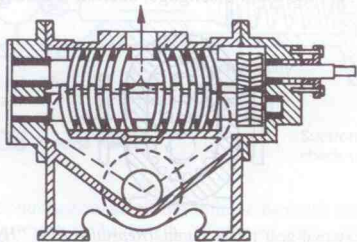


Fig. 14.1.27 Two-screw rotary pump.

The three-screw rotary pump, as illustrated in Fig. 14.1.28, is available, in the smaller sizes, for operation up to 8,000 r/min for use in lubrication systems on turbines, compressors, and centrifugal pumps. Larger units are found in such services as crude oil pipelines.

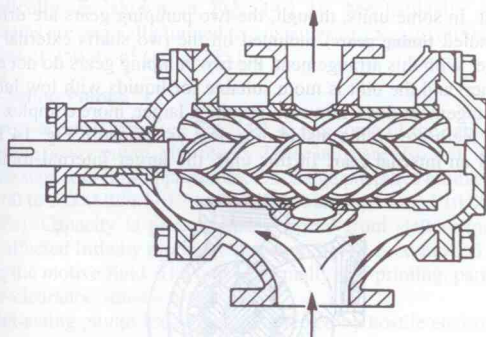


Fig. 14.1.28 Three-screw rotary pump.

A **progressing-cavity pump** is illustrated in Fig. 14.1.29. The rotor is made of polished steel, and in the shape of a single-lead screw. The stator,

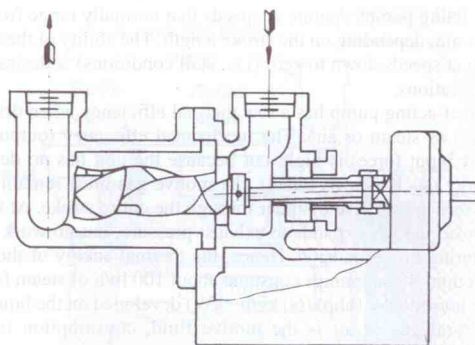


Fig. 14.1.29 Progressing-cavity (single-screw) rotary pump. (Hydraulic Institute.)

usually made of elastomer, is in the shape of a double-lead screw. As the rotor turns, its axis prescribes a circle. This characteristic requires the rotor to be driven by an internal universal joint (or equivalent). The pumpage moves through the pump in a spiral. The rubber stator makes this unit well-suited for abrasive services.

Figure 14.1.30 illustrates a **sliding vane rotary pump**. The single rotor contains multiple vanes which slide in radial slots. The rotor and casing are eccentric. The vanes maintain contact with the casing by centrifugal force and pressure. These units are typically available for capacities to 1,000 gal/min (200 m³/h), viscosities to 500,000 SSU (100,000 mm²/s), and pressures to 350 lb/in² (2.5 MPa). Some sliding-vane pumps are suitable for low-lubricity liquids such as light hydrocarbons.

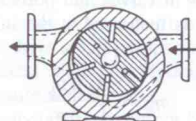


Fig. 14.1.30 Sliding-vane rotary pump. (Reprinted from "Hydraulic Institute Standards for Centrifugal, Rotary and Reciprocating Pumps," 14th ed., copyright 1983, with the permission of Hydraulic Institute.)

Flexible Member Some rotary pumps are built with flexible vanes, liners, and tubes. The flexible tube pump, also called a **peristaltic pump**, is shown in Fig. 14.1.31.

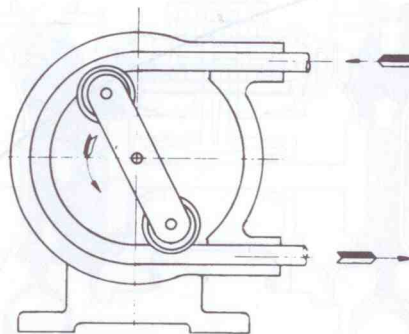


Fig. 14.1.31 Flexible-tube (peristaltic) rotary pump. (Hydraulic Institute.)

A **radial-plunger rotary pump** is shown in Fig. 14.1.32. An **axial-plunger rotary pump** (also called an **axial-piston** or **swash-plate pump**) is shown in Fig. 14.1.33. In these pumps, the inner body rotates, causing each plunger to alternately accept pumpage from the inlet and deliver it to the discharge port. In the radial pump, the length of stroke

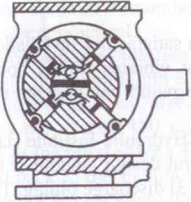


Fig. 14.1.32 Radial-plunger rotary pump.

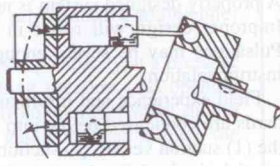


Fig. 14.1.33 Axial-plunger (swash-plate) rotary pump.

of each plunger is established by the eccentricity of the rotor. In the axial pump, the stroke length is established by the angle of the plate or shaft. On some pumps, the eccentricity (or angle) is adjustable, providing variable displacement. Rotary plunger pumps are used in hydraulic systems to provide power to hydraulic motors and cylinders.

Circumferential-Piston Pump Although sometimes considered a lobe pump, this unit (shown in Fig. 14.1.34) differs from the lobe unit in that there is no close clearance between the two rotors. Close clearance does exist, though, between each rotor and adjacent stationary parts.

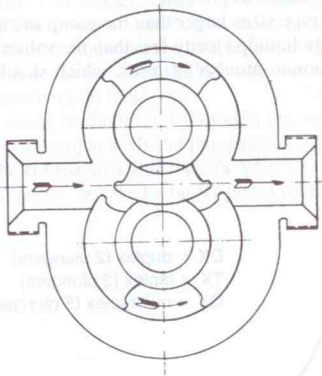


Fig. 14.1.34 Circumferential-piston rotary pump. (Hydraulic Institute.)

The **casing** (also called a **body** or **housing**) is the main pressure-retaining component of a rotary pump. It supports the rotor, contains the suction and discharge nozzles, and usually forms a portion of the boundaries of the pumping chambers.

The **bearings** of a rotary pump may be internal or external. If internal, they are lubricated by the pumpage. If the pumpage is not suitable for bearing lubrication, the bearings must be external. External bearings require two pumpage seals on each shaft. Some rotary pumps contain four seals.

Figure 14.1.35 shows typical **rotary pump performance curves**. Capacity, mechanical efficiency, and power are plotted against pressure.

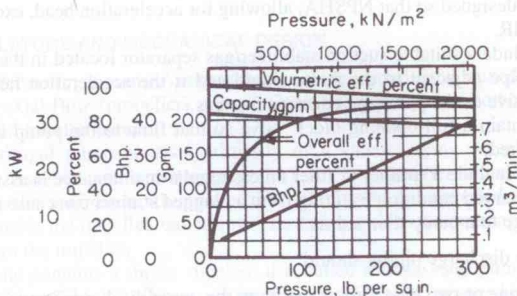


Fig. 14.1.35 Rotary pump performance curves for a herringbone gear pump at 600 r/min with a 400-SSU (90-mm²/s) liquid.

The difference between displacement and capacity is called **slip**. Power input is the sum of power output and power losses (due to friction and slip).

As defined by the Hydraulic Institute, the NPSH required by a rotary pump is that value of NPSHA when cavitation noise, a sharp drop in capacity, or a 5 percent reduction in capacity occurs, whichever occurs first.

Increased Running Clearances An increase in the clearances between the rotating parts, and/or between the rotating and stationary parts, of a rotary pump will cause the slip to increase. An increase in slip causes the volumetric efficiency to drop, the mechanical efficiency to drop, and the NPSHR to rise. If the unit is running at a constant speed, the lower volumetric efficiency (lower capacity) will result in a lower discharge pressure if the system head varies with capacity.

VOLUMETRIC EFFICIENCY

Volumetric efficiency η_v of a displacement pump is the ratio of capacity Q to displacement D : $\eta_v = Q/D$. Capacity is the volume flow rate in the suction pipe. Displacement is the volume swept by the pumping elements per unit time. For a single-acting triplex plunger pump, displacement may be calculated as follows: $D = 3(\pi/4)d^2LN$, where D = displacement; d = diameter of plunger; L = stroke length; and N = crankshaft speed, r/min.

To predict the capacity of a reciprocating pump, in the selection process, it is necessary to predict the volumetric efficiency. Volumetric efficiency may be calculated with the following equation:

$$\eta_v = 1 - \frac{S}{D'} - \frac{C}{D'} \left(\frac{v_s}{v_d} - 1 \right) \quad (14.1.2)$$

where η_v = volumetric efficiency; S = slip through both suction and discharge valves, typically about 3 percent of D' ; D' = displacement of one pumping element during one stroke; C = clearance volume (dead space) in the pumping chamber (when the pumping element is at the end of the discharge stroke); v_s = specific volume of pumpage at suction conditions; and v_d = specific volume of the pumpage at discharge conditions.

Although volumetric efficiency has an effect on mechanical efficiency, in reciprocating pumps the two do not necessarily move in unison. It is possible to have a high η_m with a low η_v (due to liquid compressibility), or a low η_m with a high η_v (such as would occur in a high-suction-pressure, low-differential-pressure application).

The same equation can be applied to rotary pumps, but usually $v_s = v_d$, so that Eq. (14.1.2) reduces to $\eta_v = 1 - S/D'$. For a rotary pump, S = leakage of pumpage from the discharge side of the pump to the suction side. Because rotary pumps have no check valves, they must rely on close clearances to restrict slip. Slip is a function of the differential pressure, the viscosity of the pumpage, and the clearance between the parts. Slip in a rotary pump is usually a higher fraction of displacement than in a reciprocating pump.

POWER OUTPUT AND INPUT

The **power output** (also called **hydraulic power**, **hydraulic horsepower**, and **water horsepower**) of any pump is the usable power it imparts to the pumpage. It is the product of capacity and differential pressure.

In U.S. units, $P_o = Q \times \Delta p / 1,714 = Q(p_d - p_s) / 1,714$, where P_o = power output, hp; Q = pump capacity, gal/min; Δp = pump differential pressure, lb/in²; p_d = pump discharge pressure, lb/in²; and p_s = pump suction pressure, lb/in².

In metric units, $P_o = Q \times \Delta p / 3,600 = Q(p_d - p_s) / 3,600$, where P_o = power output, kW; Q = pump capacity, m³/h; Δp = pump differential pressure, kPa; p_d = pump discharge pressure, kPa; and p_s = pump suction pressure, kPa.

The **power input** (sometimes called **brake horsepower**) is the power required to drive power and rotary pumps. Normally, the power input includes the losses in an integral gear unit, but not the losses in a separate

gear unit or variable-speed drive. V-belt losses may be included. If the mechanical efficiency is known, power input may be calculated by dividing the power output by the mechanical efficiency.

MECHANICAL EFFICIENCY—POWER AND ROTARY PUMPS

Mechanical efficiency (also called **pump efficiency** and **overall efficiency**) of a power-driven pump is the ratio of output power P_o to input power P_i : $\eta_m = P_o/P_i$.

As shown in Fig. 14.1.36, mechanical efficiency drops as the frame load is reduced, because the power output falls faster than friction losses, and becomes a smaller part of the power input. Mechanical efficiency is zero when the differential pressure is zero.

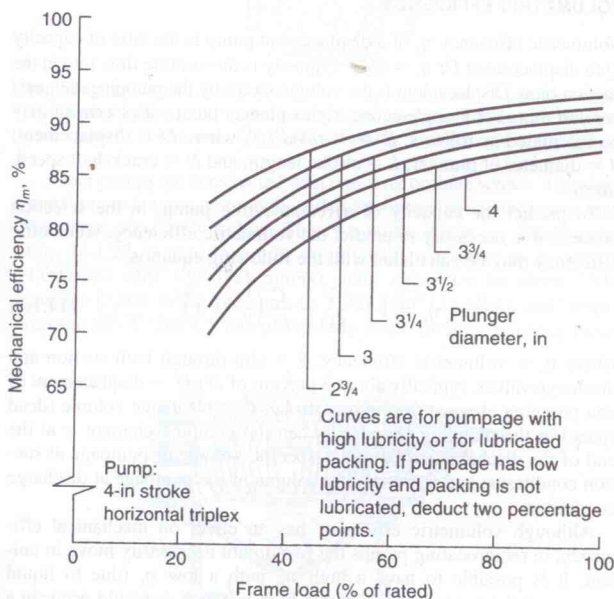


Fig. 14.1.36 Mechanical (overall) efficiency of a triplex power pump.

PULSATION DAMPENERS

A **pulsation dampener** is a device which reduces the pressure pulsations created, in a suction or discharge pipe, by the interaction of a pump and a system. The dampener performs this function by reducing the velocity variations of the pumpage in the pipe.

Most reciprocating pumps, and many rotary pumps, create a variation in the velocity of the pumpage in the piping. Unless the suction and discharge vessels are very close to the pump, these velocity variations will be converted by the system into pressure variations. To reduce these pulses, a small vessel can be installed in the suction and/or discharge line. Such a vessel mounted in the discharge line is called a **pulsation dampener**. If the suction pressure is high enough, this same type of device can be used in the suction line, but a special dampener, called a **suction stabilizer** will perform the dual function of reducing pulses and separating free gas from the pumpage.

For optimum effectiveness, each dampener must be installed close to the pipe and adjacent to the pump. There must be minimal restriction between the dampener and pipe. For a low suction pressure, a simple air-filled standpipe is often adequate. For the discharge side, and for high suction pressures, the dampener often contains a diaphragm or bladder that is precharged with nitrogen to about 70 percent of system pressure.

SYSTEM DESIGN

A properly designed system is necessary for a satisfactory installation. Improper design will result in a system that vibrates and is noisy. Pulsations may be severe enough to damage pump components and instrumentation.

Field experience and information from the Hydraulic Institute standards are condensed below into a list of general design guidelines for the (1) suction vessel, (2) suction piping, and (3) discharge piping. The asterisks in the following lists show features that *do not* apply to rotary pumps that produce a uniform flow rate (constant velocity).

The suction vessel should:

Be large enough to provide sufficient retention time to allow free gas to rise to the liquid surface.

Have the feed and return lines enter below the minimum liquid level.

Include a vortex breaker over the outlet (pump suction) line and/or provide sufficient submergence to preclude vortex formation.

Contain a weir plate to force gas bubbles toward the surface. The top of the weir must be sufficiently below the minimum tank level to avoid disturbance.

The suction piping should:

Be as short and direct as possible.

Be one or two pipe sizes larger than the pump suction connection.

Have an average liquid velocity less than the values in Fig. 14.1.37.*

Contain a minimum number of turns, which should be long-radius elbows or laterals.*

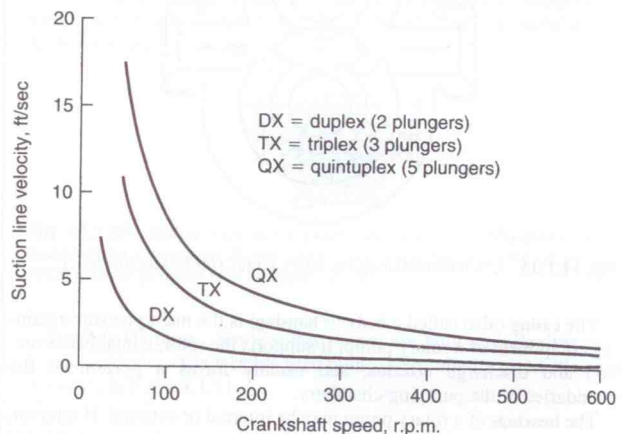


Fig. 14.1.37 Suction line velocities for constant acceleration head, for reciprocating power pumps.

Be designed to preclude the collection of vapor in the piping. (There should be no high points, unless vented. The reducer at the pump should be the eccentric type, installed with the straight side up.)

Be designed so that NPSHA, allowing for acceleration head, exceeds NPSHR.

Include a suitable suction stabilizer/gas separator located in the suction pipe adjacent to the pump liquid end if the acceleration head is excessive or the pumpage contains free gas.

Contain a full-opening block valve so that flow to the pump is not restricted.

Not include a strainer or filter unless regular maintenance is assured. (The starved condition resulting from a plugged strainer can cause more damage to a pump than solids.)

The discharge piping should:

Be one or two pipe sizes larger than the pump discharge connection.

Have an average velocity less than 3 times the maximum suction line velocity.

Contain a minimum number of turns, using long-radius elbows and laterals where practical.

Include a suitable pulsation dampener (or provisions for adding one) adjacent to the pump liquid end."

Contain a relief valve, sized to pass full pump capacity at a pressure that does not exceed 110 percent of cracking pressure (the opening pressure of the relief valve). The discharge from the relief valve should be piped back to the suction vessel so that gases liberated through the valve are not fed back into the pump.

Contain a bypass line and valve so that the pump may be started against negligible discharge pressure.

Contain a check valve to prevent the imposition of system pressure on the pump during start-up.

The features of a good system are illustrated in Fig. 14.1.38.

Remedies for Low NPSHA/High NPSHR

In designing the suction system for a displacement pump, if NPSHA is found to be less than NPSHR, a remedy may be found in the following list:

Increase the diameter of the suction line.

Reduce the length of the suction line by providing a more direct route, or by moving the pump closer to the suction vessel.

Install a suction bottle or stabilizer adjacent to the pump liquid end. A fabricated bottle has often been successfully used at pressures below about 50 lb/in² (300 kPa), but maintenance of a liquid level is required.

A section of rubber hose in the suction line, adjacent to the pump, will often reduce the acceleration head.

Elevate the suction vessel or the level of liquid in the suction vessel.

Reduce temperature of pumpage. (If the pumpage is at bubble point in the vessel, it must be cooled after it leaves the vessel.)

Reduce speed of the pump, or select a larger pump running slower.

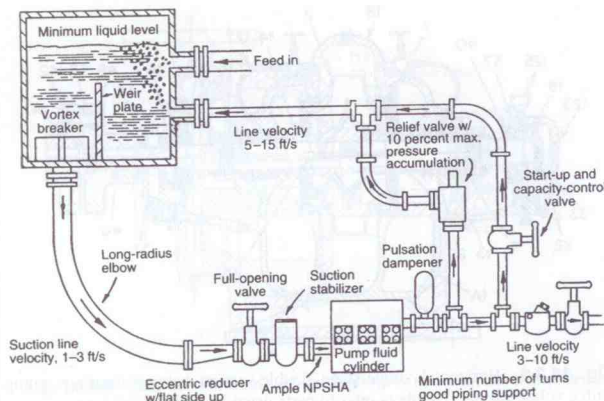


Fig. 14.1.38 A good system for a reciprocating pump.

At a lower speed, it may be possible to operate a reciprocating pump with light suction valve springs, or none at all.)

If the above steps are insufficient, impractical, or impossible, a booster pump must be provided. A booster for a displacement pump is normally a centrifugal, although direct-acting reciprocating and rotary pumps are sometimes used. The NPSHR of the booster must be less than that available from the system. The head of the booster should exceed the main pump NPSHR plus suction-line losses by about 20 percent. The booster should be installed near the suction vessel, and a pulsation bottle or stabilizer should be installed in the suction line, adjacent to the main pump, to protect the booster from pulsating flow.

14.2 CENTRIFUGAL PUMPS

by Terry L. Henshaw

REFERENCES: Stepanoff, "Centrifugal and Axial Flow Pumps," Wiley. Wislicenus, "Fluid Mechanics of Turbomachinery," McGraw-Hill, and Marks' Handbook, 10th ed., McGraw-Hill. Pfleiderer, "Die Kreiselpumpen," Springer. Spannake, "Centrifugal Pumps, Turbines and Propellers," Technology Press. Hicks, "Pump Selection and Application" and "Pump Operation and Maintenance," McGraw-Hill. Standards of the Hydraulic Institute. Karassik, "Centrifugal Pump Clinic," Marcel Dekker.

NOMENCLATURE AND MECHANICAL DESIGN

The term "centrifugal pumps" as used here includes radial-flow, mixed-flow, and axial-flow (propeller) pumps. Centrifugal pumps are sometimes called **impeller pumps**.

A centrifugal pump consists primarily of a vane-carrying wheel, called the **impeller**, and a **casing** (Fig. 14.2.1). The impeller imparts velocity to the **pumpage**, with most of that velocity being converted to pressure inside the impeller and casing. The casing guides the pumpage and from the impeller.

If a pump contains a single impeller, it is called a **single-stage** pump (Figs. 14.2.1 and 14.2.2). When two or more impellers are arranged in series (for higher pressure), it is called a **multistage** pump (Figs. 14.2.9 and 14.2.12).

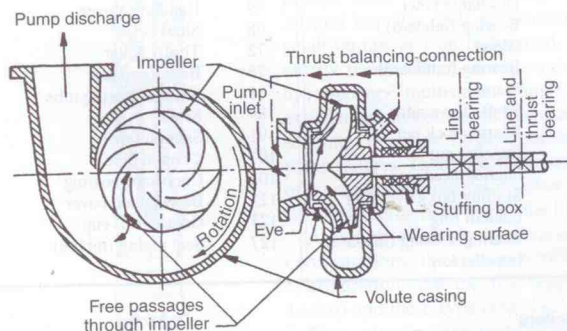


Fig. 14.2.1 Single-stage single-suction overhung pump with volute casing.

Size designations of centrifugal pumps usually contain the nominal sizes of the discharge and suction nozzles, and sometimes the approximate maximum diameter of the impeller—or the number of stages. For example, in U.S. units, a 2 × 3 × 8 pump would have a 2-in discharge