# SECTION 3.3

# STEAM PUMPS

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## BASIC THEORY

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

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

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

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

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

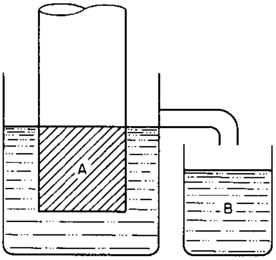

**FIGURE 1** The volume of liquid displaced by a solid equals the volume of the solid

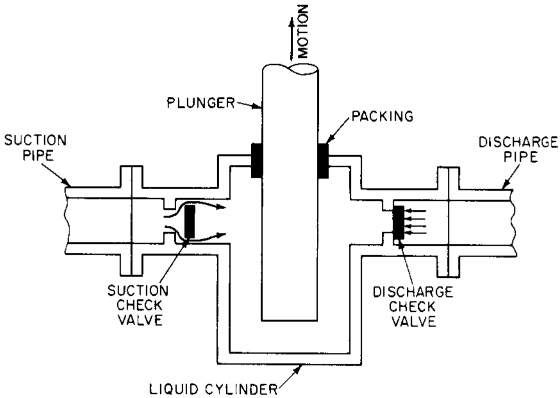

**FIGURE 2** Schematic of the liquid end of reciprocating pump during the suction stroke

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

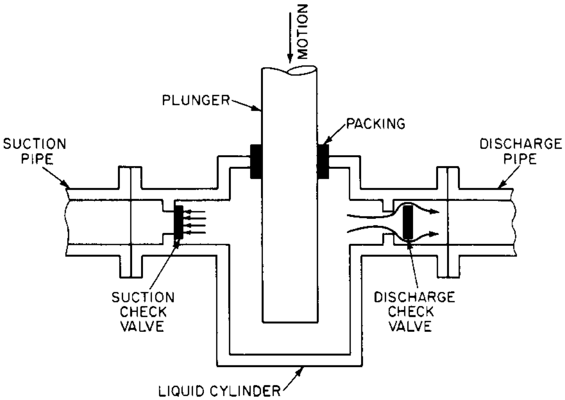

**FIGURE 3** Schematic of the liquid end of a reciprocating pump during the discharge stroke

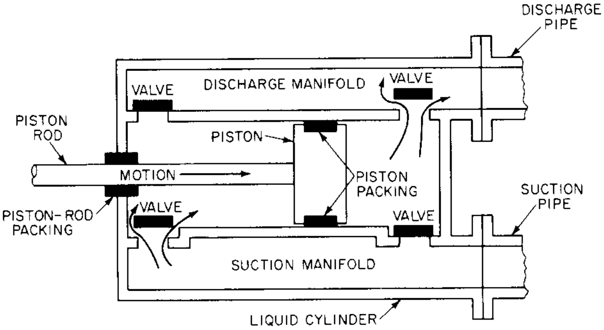

**FIGURE 4** Schematic of a double-acting liquid end

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.

## 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:

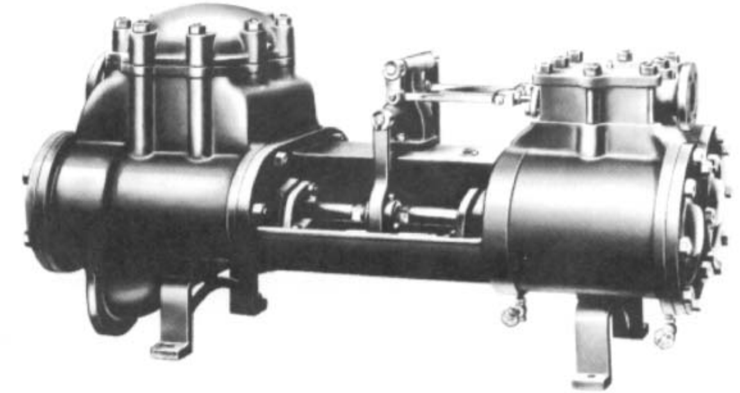

**FIGURE 5** Duplex steam pump

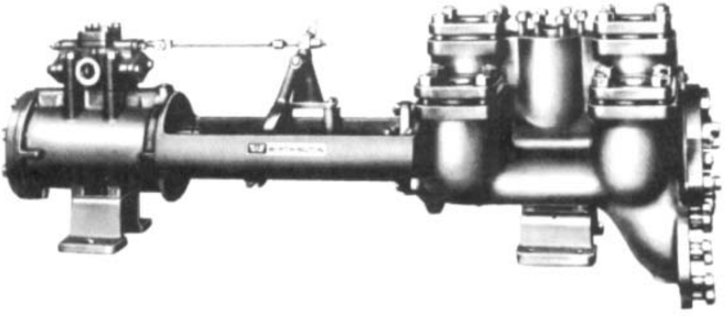

**FIGURE 6** Simplex steam pump (Flowserve Corporation)

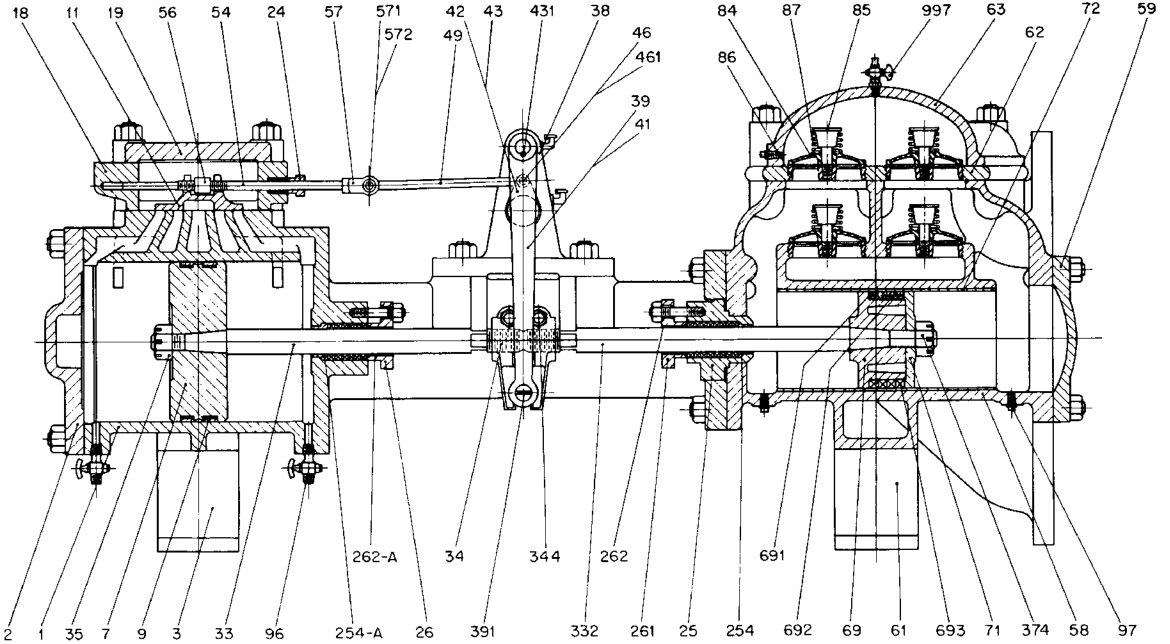

**FIGURE 7** Typical section of duplex steam pump: (1) Steam cylinder with cradle, (2) steam cylinder head, (3) steam cylinder foot, (7) steam piston, (9) steam piston rings, (11) slide valve, (18) steam chest, (19) steam chest cover, (24) valve rod stuffing box gland, (25) piston rod stuffing box, liquid,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

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

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

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

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

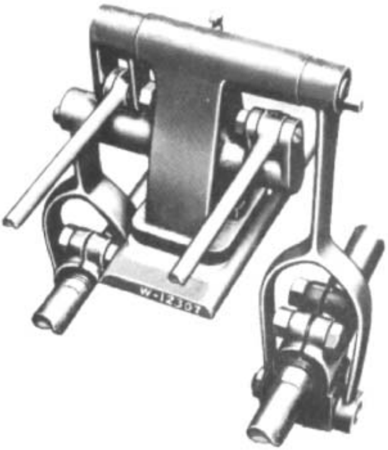

**FIGURE 8** Steam valve actuating mechanism or a duplex pump (Flowserve Corporation)

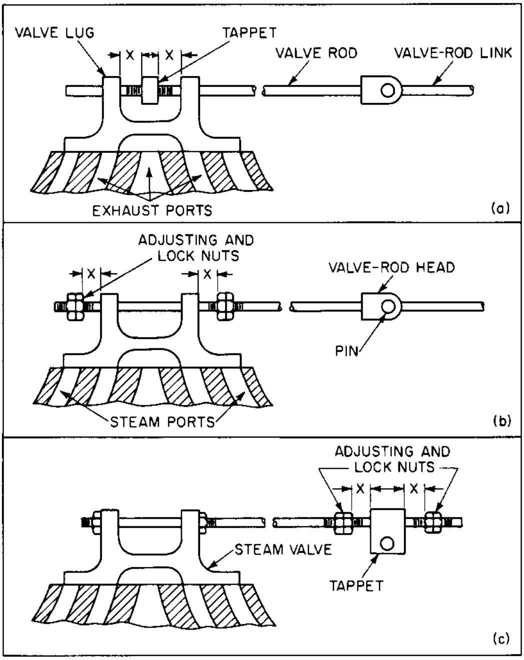

**FIGURE 9A through C** Duplex pump steam valve lost-motion arrangements

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

***Flat Slide Steam Valves*** Steam enters the pump from the steam pipe into the steam chest on top of the steam cylinder. Exhaust steam leaves the pump through the center port of five ports, as shown in Figure 9. Most duplex pumps use a flat slide valve that is held against its seat by steam pressure acting upon its entire top area; this is called an *unbalanced* valve. The flat or D valve, as it is often called, is satisfactory for steam pressures up to approximately 250 lb/in$^2$ (17 bar[1]) 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.

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[1] 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 vol-

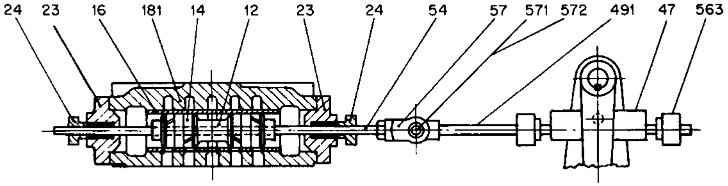

**FIGURE 10** Balanced piston steam valve: (12) piston valve, (181) steam chest, (23) valve rod stuffing box, (24) valve rod stuffing box gland, (54) valve rod, complete, (57) valve rod head, (14) piston valve ring, (16) piston valve lining, (47) lost-motion block tappet, (491) valve rod link, (563)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

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

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

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

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

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

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

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

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

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

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

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

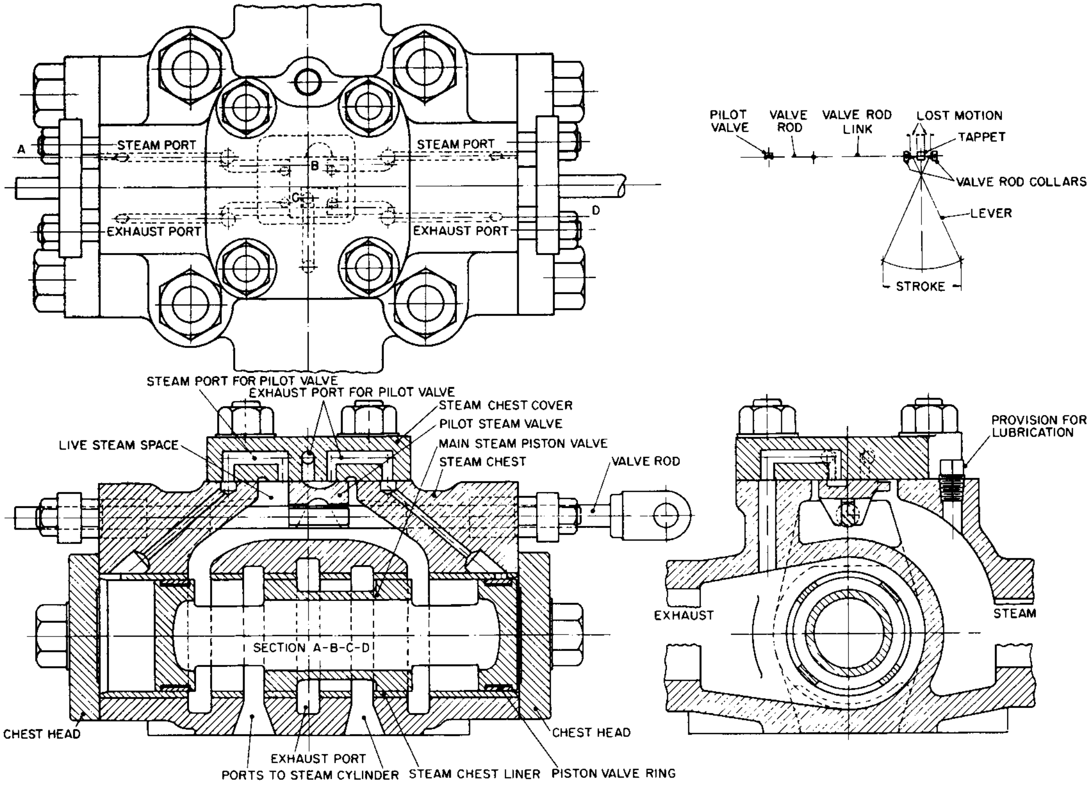

**FIGURE 11** Simplex-type steam valve (Flowserve Corporation)

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

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

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

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

## *LIQUID END CONSTRUCTION*

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

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

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

***Piston-Type Liquid Ends*** The most generally used piston pump is the cap-and-valve plate design, illustrated in Figure 7. This is usually built for low pressures and temperatures, although some designs are used at up to 350 lb/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.

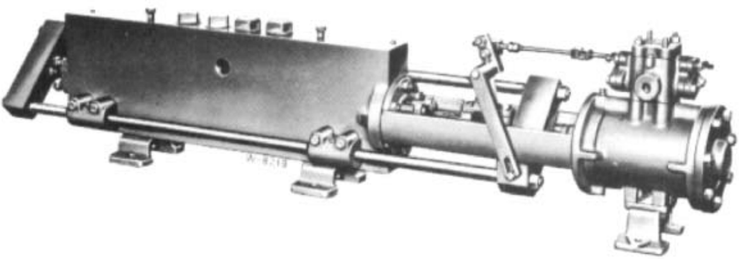

**FIGURE 12** Simplex-type plunger pump (Flowserve Corporation)

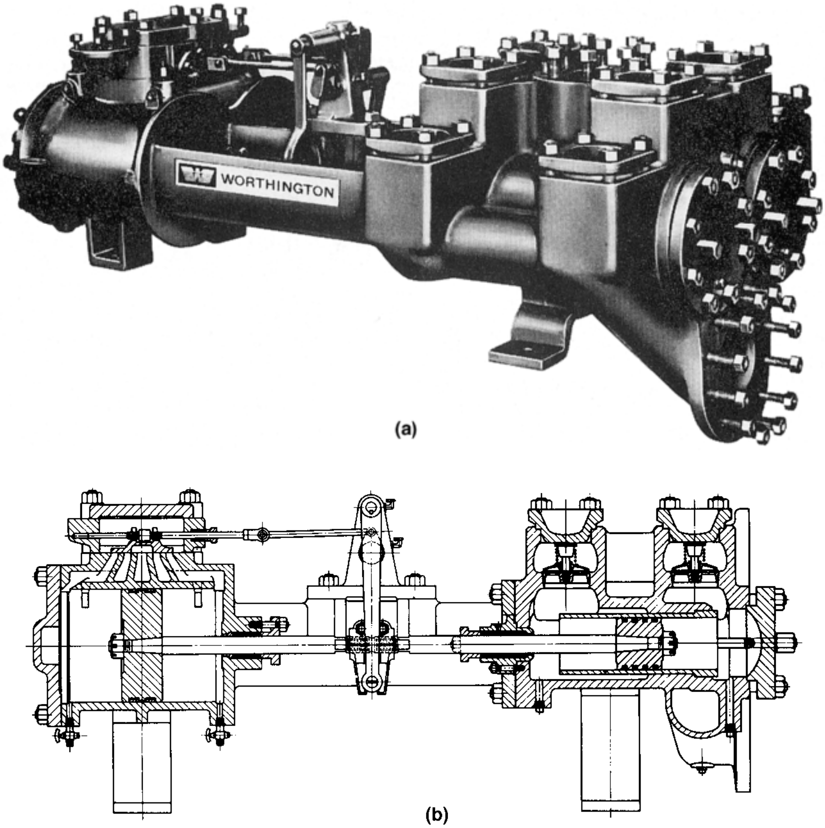

**FIGURE 13A and B** Side-pot piston pump (Flowserve Corporation)

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 discharge valve will compress and absorb the entire displacement of the pump. When this occurs, the dis-

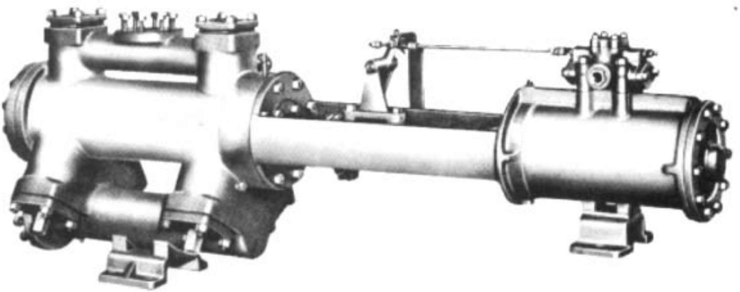

**FIGURE 14** Close-clearance liquid end pump (Flowserve Corporation)

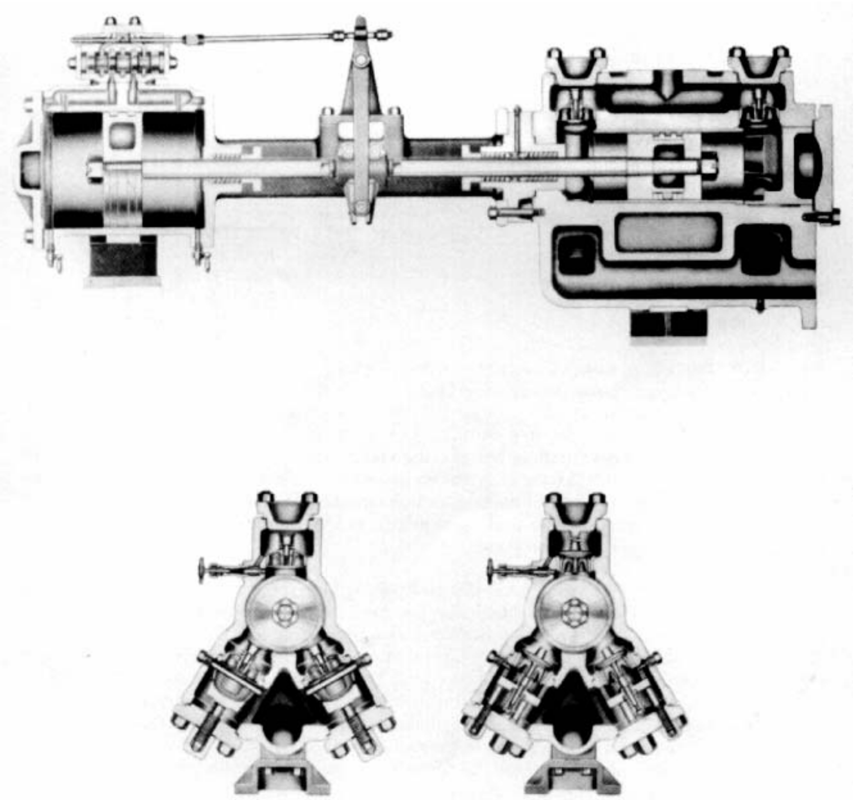

**FIGURE 15** End views of close-clearance liquid end pump, showing disk valve assembly and wing valve assembly (Flowserve Corporation)

charge 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.

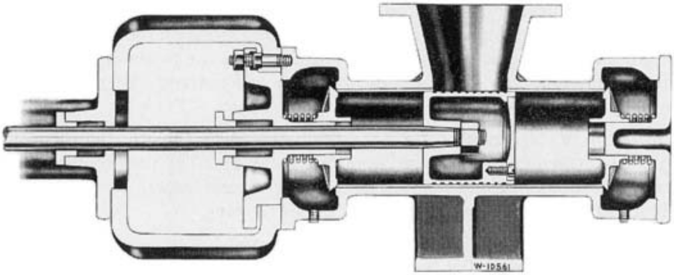

**FIGURE 16** A pump for handling viscous liquid (Flowserve Corporation)

**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 |

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.

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

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

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

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

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

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

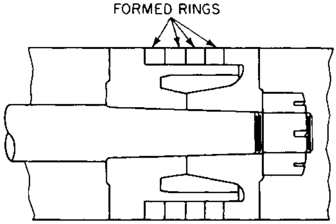

**FIGURE 17** Body-and-follower piston

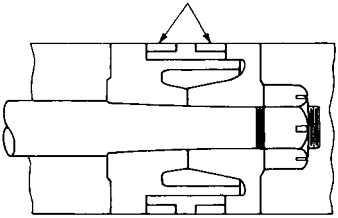

**FIGURE 18** Body-and-follower piston with snap rings

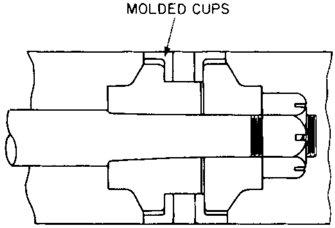

**FIGURE 19** Body-and-follower piston with molded cups

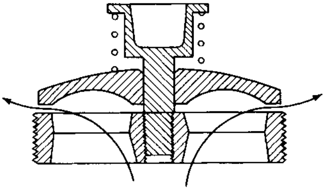

**FIGURE 20** Stem-guided disk valve

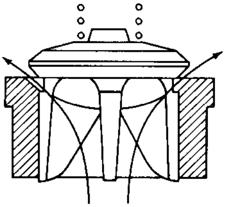

**FIGURE 21** Wing-guided valve

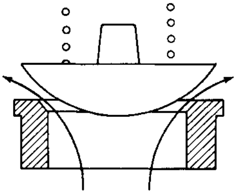

**FIGURE 22** Semispherical valve

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 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

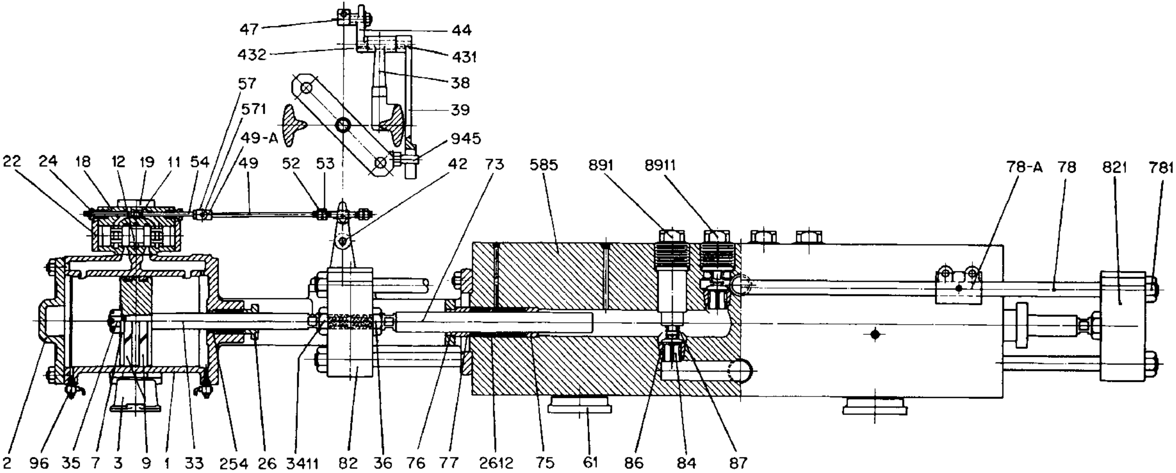

**FIGURE 23** Simplex plunger pump with forged steel cylinder: (1) steam cylinder with cradle, (2) steam cylinder head, (3) steam cylinder foot, (7) steam piston, (9) steam piston rings, (11) slide valve, (12) piston valve, (18) steam chest, (19) steam chest cover, (22) steam chest head, (24) valve rod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

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

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

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

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.

**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)

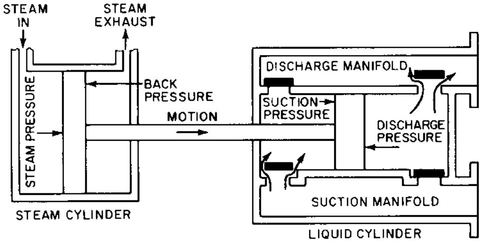

**FIGURE 24** Schematic showing direction of forces on pistons

**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)

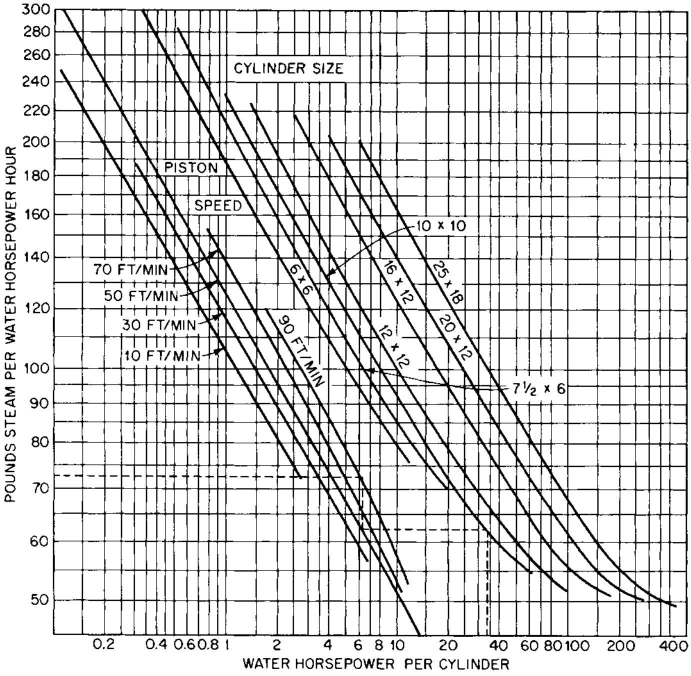

**FIGURE 25** Approximate steam consumption for steam pumps (pounds of steam per water horsepower-hour = 1.644 kg of steam per water kilowatt-hour; water horsepower per cylinder = 1.341 wkW per cylinder; ft/s = 0.3047 m/s; in = 25.44 mm) (Hydraulic Institute Standards, 12th Edition, 1969—out of print)

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

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

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,

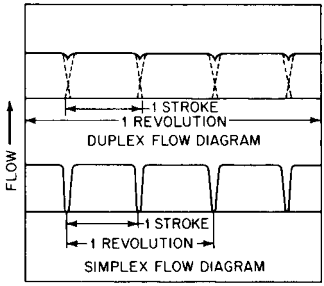

**FIGURE 26** Flow characteristics of simplex and duplex pumps

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

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

## *FURTHER READING*

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