Reciprocating pumps

4

Nomenclature

A	plunger or piston cross-sectional area, mm^2 (in^2)
a	piston rod cross-sectional area, mm^2 (in^2)
B	bulk modulus of fluid, kPa (psi)
BHP	brake horsepower, kW (hp)
C	constant (for type of pump)
D	pump displacement, m^3/h (gpm)
d	displacement per pumping chamber, m^3 (gal)
d'	pump piston or plunger diameter, mm (in.)
d_w	diameter of wrist pin, mm (in.)
E_M	pump mechanical efficiency
f_p	pump pulsation frequency, cycles/s
g	acceleration due to gravity, 9.81 m/s^2 (32.2 ft/s^2)
H_A	the head on the surface of the liquid supply level, m (ft)
H_{AC}	acceleration head, m (ft)
H_{PH}	potential head, m (ft)
HSH	vapor pressure head, m (ft)
H_{VH}	velocity head, m (ft)
H_{VPA}	static pressure head, m (ft)
H_f	pipe friction loss, m (ft)
H_p	total head required for pump, m (ft)
HHP	hydraulic horsepower, kW (hp)
K	A factor based on fluid compressibility
L	length of suction line, m (ft)
l	length of wrist pin under load, mm (in.)
m	number of pistons, plungers, or diaphragms
NPSH	net positive suction head, m (ft)
$NPSH_A$	net positive suction head available, m (ft)
$NPSH_R$	net positive suction head required, m (ft)
n	stroke rate or crank revolutions per min, rps (rpm)
P	bladder precharge pressure, kPa (psi)
P_c	plunger load, N (lb)
PL	pressure increase, kPa (psi)
ΔP	static pressure, kPa (psi)
Q	flow rate, m^3/h (ft^3/s)
Q'	flow rate, m^3/h (BPD)
q	flow rate, m^3/h (gpm)
S	stress, kPa (psi)
S'	valve slip, %
s	stroke length, mm (in.)
(SG)	specific gravity of liquid relative to water
V	velocity in suction line, m/s (ft/s)
Vol	volume of surge tank, m^3 (ft^3)

Surface Production Operations. https://doi.org/10.1016/B978-0-12-809895-0.00004-1

$(Vol)_g$	required gas volume, m^3 (ft^3)
Z	elevation above or below pump centerline datum, m (ft)
ρ	density of fluid, kg/m^3 (lb/ft^3)

4.1 Engineering principles

4.1.1 Background

Positive displacement (PD) pumps were developed long before centrifugal pumps. Fluid is positively displaced from a fixed volume container. PD pumps are

- Capable of developing high pressures while operating at low suction pressures
- Self-priming
- Commonly referred to as constant volume pumps

Unlike centrifugal pumps, a PD pump's capacity is not affected by the pressure against which it operates. PD pumps' flow rate is usually regulated by varying the speed of the pump. PD pumps are generally used in

- High head
- Low flow applications
- Where high maintenance costs are more than offset by their higher efficiency

4.1.2 Pumping action

In a PD pump the volume containing the liquid is decreased until the resulting liquid pressure is equal to the pressure in the discharge system. That is, the liquid is compressed mechanically causing a direct rise in potential energy. As shown in Fig. 4.1, PD pumps are classified as either reciprocating or rotary pumps. Most PD pumps are classified as reciprocating pumps where the displacement is accomplished by the linear motion of a piston or plunger is a cylinder. Rotary pumps' displacement is accomplished by a circular motion. Rotary pumps are discussed in Chapter 5.

PD pumps, unlike centrifugal pumps, will in theory produce the same flow at a given speed no matter what the discharge pressure. Thus PD pumps are "constant flow machines." However due to a slight increase in internal leakage as the pressure increases, a truly constant flow rate cannot be achieved. A PD pump must not be operated against a closed valve on the discharge side of the pump, because it has no shutoff head like centrifugal pumps. A PD pump operating against a closed discharge valve will continue to produce flow and the pressure in the discharge line will increase, until the line bursts or the pump is severely damaged, or both. Since the pump will attempt to match whatever system pressure is required, it is necessary that a relief valve be installed on the pump discharge. This procedure ensures that the pump will not overpressure itself or the discharge pipe.

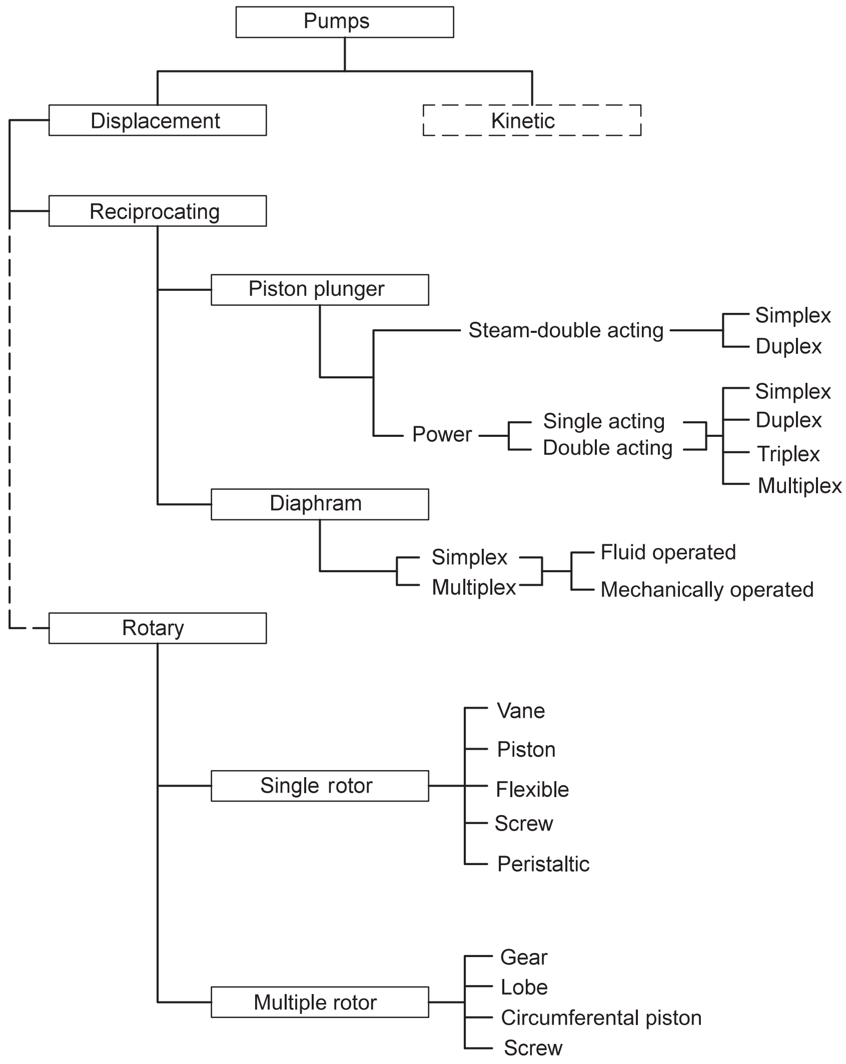


Fig. 4.1 Classification of positive displacement pumps.
Courtesy of the Hydraulic Institute.

A reciprocating pump traps a fixed volume of liquid at near-suction conditions, compresses it to discharge pressure, and pushes it out of the discharge nozzle. Energy is added to the fluid intermittently as one or more boundaries is moved linearly with a piston, plunger, or diaphragm in one or more fluid-containing volumes. Liquid is moved by means of a constant back-and-forth motion of a piston, plunger, or diaphragm within a fixed volume or cylinder.

4.1.3 Pump types

4.1.3.1 Plunger and piston pump (Fig. 4.2)

In a plunger pump, the plunger moves through a stationary packed seal and is pushed into the fluid. In a piston pump, the packed seal on the piston pushes the fluid from the cylinder. Movement of either the plunger/piston creates an alternating increase and decrease of flow. As shown in Fig. 4.3, as the piston, or plunger, moves backward (suction stroke), the available volume in the cylinder increases and a suction valve opens to allow the fluid to enter the cylinder. As shown in Fig. 4.4, as the piston, or plunger, moves forward (discharge stroke), the volume available in the cylinder

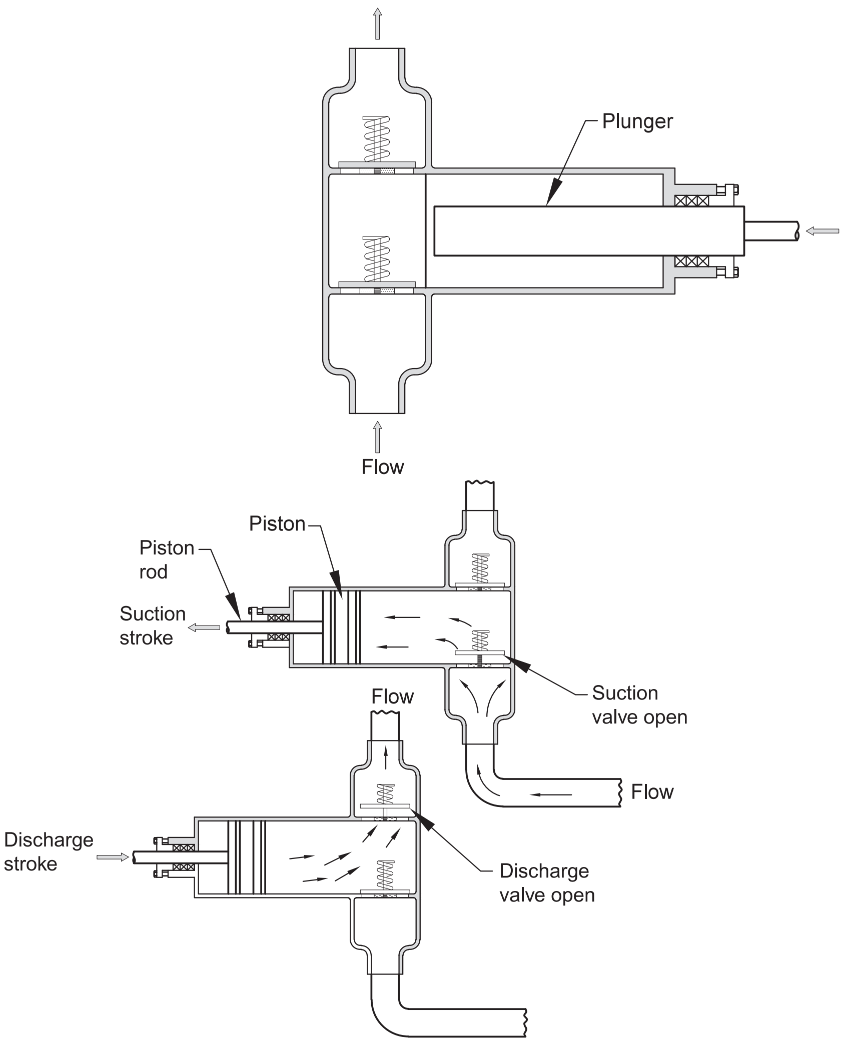


Fig. 4.2 Schematic diagrams illustrating a plunger (top) and piston (bottom) pump.

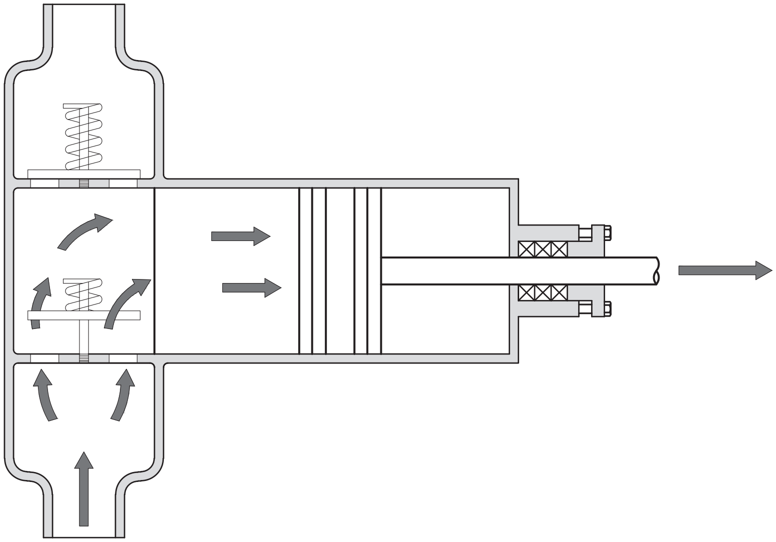


Fig. 4.3 Schematic illustrating the backward "suction" stroke of a single-acting pump.

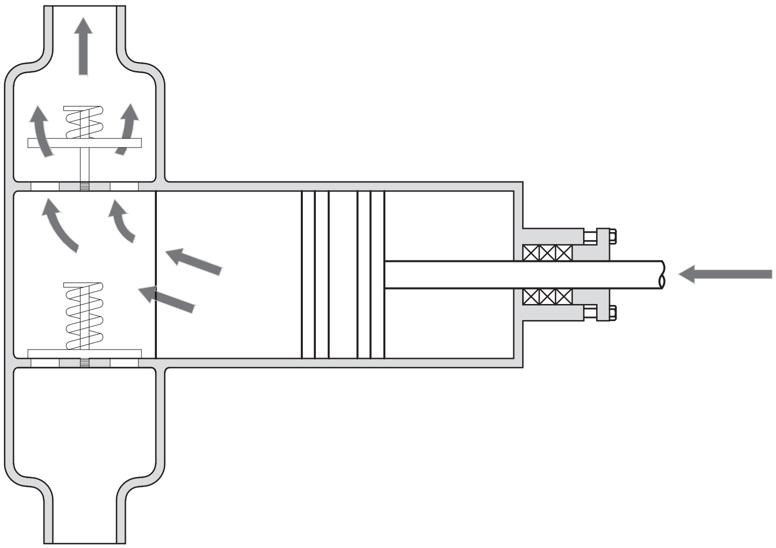


Fig. 4.4 Schematic illustrating the forward "discharge" stroke of a single-acting pump.

decreases, the pressure of the fluid increases, and the fluid is forced out through a one-way discharge valve.

If liquid is pumped during linear movement in one direction only (Fig. 4.5), the pump is classified as "single-acting." If the liquid is pumped during movement in both directions (Fig. 4.6), it is classified as "double-acting."

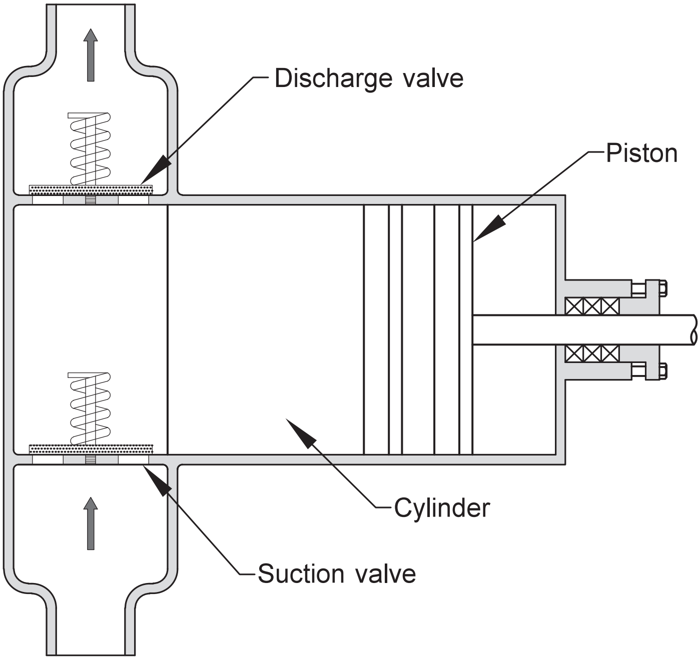


Fig. 4.5 Schematic diagram of a “single-acting” piston pump.

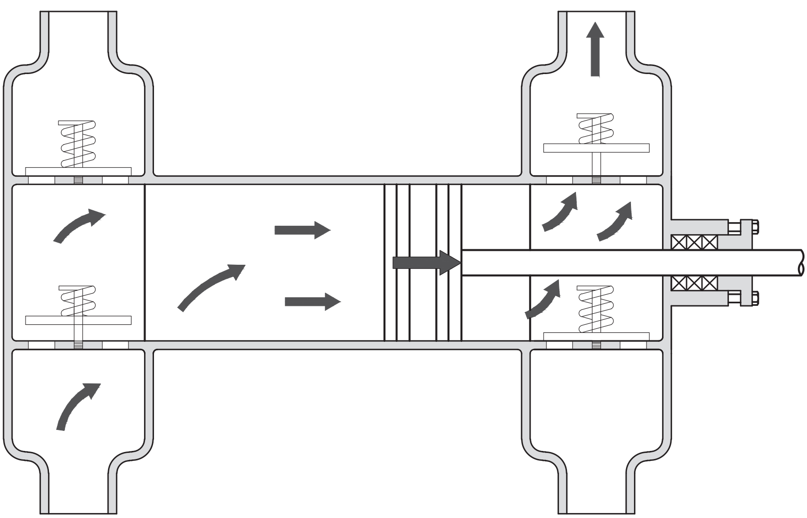


Fig. 4.6 Schematic diagram of a “double-acting” piston pump.

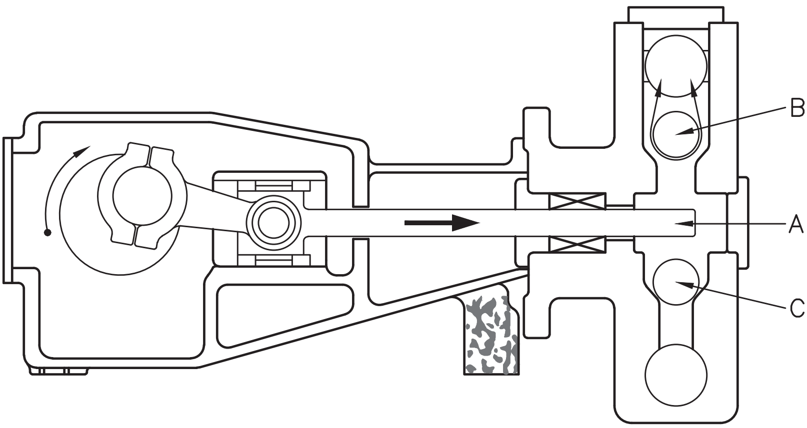


Fig. 4.7 Schematic diagram of a "single-acting" plunger pump (A—plunger; B—discharge check valve; C—suction check valve).

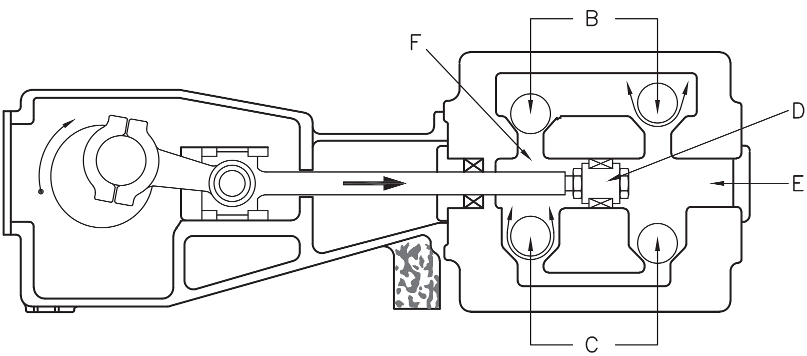


Fig. 4.8 Schematic diagram of a "double-acting" piston pump (B—discharge check valve; C—suction check valve; D—piston; E—discharge pressure; and F—suction pressure).

Figs. 4.7 and 4.8 show a single-acting and double-acting pump. As the plunger (*point A*) moves to the right in the single-acting pump, the fluid is compressed until its pressure exceeds the discharge pressure, and the discharge check valve (*point B*) opens. The continued movement of the plunger to the right pushes liquid into the discharge pipe. As the plunger begins to move to the left, the pressure in the cylinder becomes less than that in the discharge pipe, and the discharge valve (*point B*) closes. Further movement to the left causes the pressure in the cylinder to continue to decline until it is below suction pressure. At this point the suction check valve (*point C*) opens. As the plunger continues to move to the left, the cylinder fills with liquid from the suction. As soon as the plunger begins to move to the right, it compresses the liquid to a high enough pressure to close the suction valve (*point C*), and the cycle is repeated. Thus liquid is discharged only when the plunger moves to the right.

In a double-acting pump, the plunger is replaced by a piston (*point D*). When the piston moves to the right, the liquid in the cylinder to the right of the piston (*point E*) is discharged, and the cylinder to the left of the piston (*point F*) is filled. When the direction of the piston is reversed, the liquid in "*F*" is discharged, and the cylinder at "*E*" is filled with suction fluid. Thus liquid is pumped when the piston moves in either direction.

Reciprocating pumps are also classified by the number of cylinders they have. If the liquid is contained in one cylinder it is called a simplex pump, two cylinders a duplex, three cylinders a triplex, five cylinders a quintuplex, seven cylinders a setuplex, and so forth. Reciprocating pumps normally have odd numbers of cylinders to reduce the rocking coupling forces.

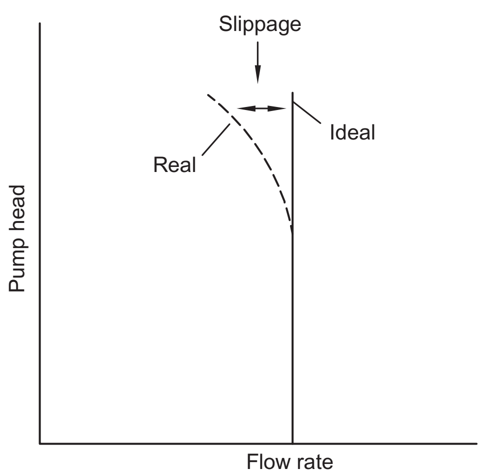
As shown in Fig. 4.9, the head-flow rate curve is a nearly straight vertical line. That is, no matter how high a head is required, the plunger will displace a given volume of liquid for each rotation. The flow rate through the pump can be varied only by changing the pump speed. A throttling valve that changes the system head-flow rate curve will have no effects on the flow rate through the pump.

4.1.3.1.1 Advantages

The advantages of reciprocating pumps are as follows:

- High efficiency regardless of changes in required head; efficiencies on the order of 85% to 95% are common.
- The efficiency remains high regardless of pump speeds, although it tends to decrease slightly with increasing speed.
- Reciprocating pumps' operating speeds are much lower than those of centrifugal pumps. As a result, reciprocating pumps are better suited for handling viscous fluids.
- For a given speed the flow rate is constant regardless of head. The pump is limited only by the power of the prime mover and the strength of the pump parts.
- Capable of producing high pressures and large capacities and are self-priming (Fig. 4.10)

Fig. 4.9 Positive displacement reciprocating pump characteristic curve.



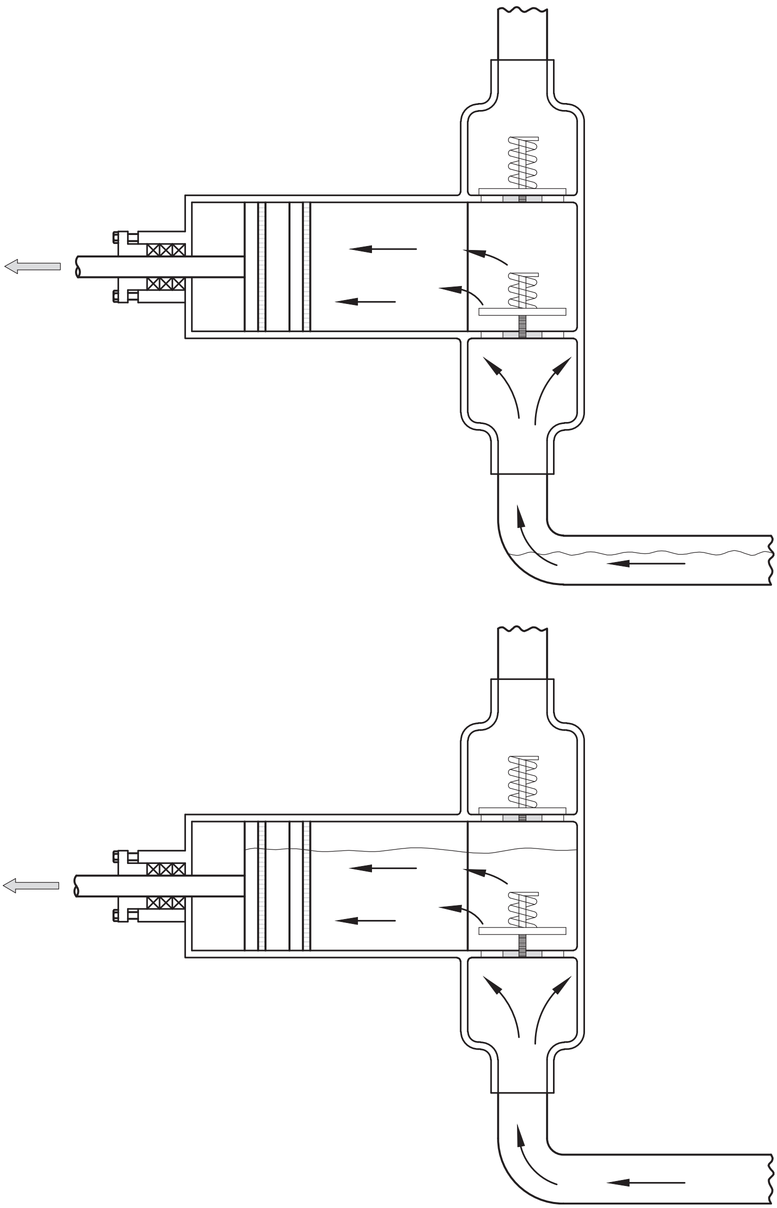


Fig. 4.10 Schematic showing "self-priming" of a reciprocating pump.

4.1.3.1.2 Limitations

Reciprocating pumps, because of the nature of their construction, have the following limitations when compared to centrifugal pumps:

- They have higher maintenance cost and lower availability because of the pulsating flow and large number of moving parts.

- They are poorer at handling liquids containing solids that tend to erode valves and seats.
- Because of the pulsating flow and pressure drop through the valves, they require larger suction pressures ($NPSH_R$) at the suction flange to avoid cavitation.
- They are heavier in weight and require more space.
- Pulsating flow requires special attention to suction and discharge piping design to avoid both acoustical pulsations and mechanical vibrations.

4.1.3.1.3 Vapor-locking

Vapor-locking is the condition where the cylinder becomes filled with high-pressure gas. As the piston moves forward, it compresses the gas into a smaller volume. The pressure of the gas in the cylinder may not rise as high as the discharge line pressure thus preventing the discharge valve from opening. This condition causes excess heat and packing failure (Fig. 4.11).

4.1.3.2 Diaphragm pump

Diaphragm pumps are a special type of reciprocating pump that uses the action of a diaphragm moving back and forth within a fixed chamber. Fig. 4.12 shows a typical diaphragm pump where the flexure of the diaphragm creates the pumping action. Diaphragm pumps are also known as controlled volume, proportioning, metering, and chemical injection.

The principle of operation is similar to that of the plunger/piston pump except that, instead of a cylinder, there is a flexible pulsating diaphragm. When gas pressure is applied against either diaphragm it forces liquid out. When the gas is relieved the diaphragm flexes under the pressure in the suction line and allows liquid to enter.

Diaphragm pumps can be fluid or mechanically driven at either constant or variable speeds. They are limited to small flow rates and moderate temperatures. They usually handle a small discharge volume (typically between 1 and 10 gpm) and high discharge

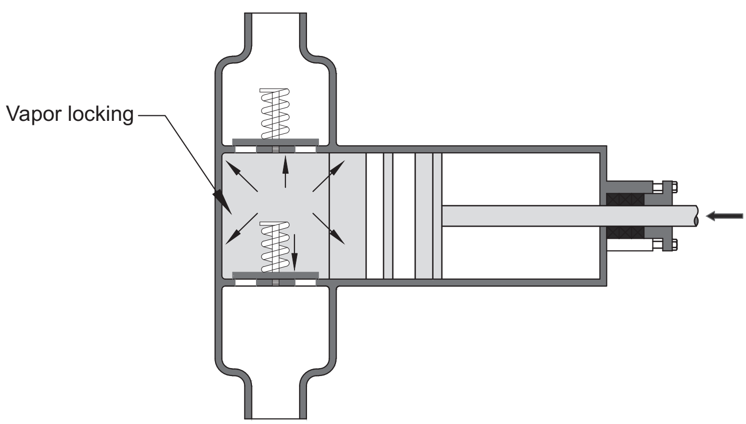


Fig. 4.11 Schematic illustrating the concept of "vapor-locking."

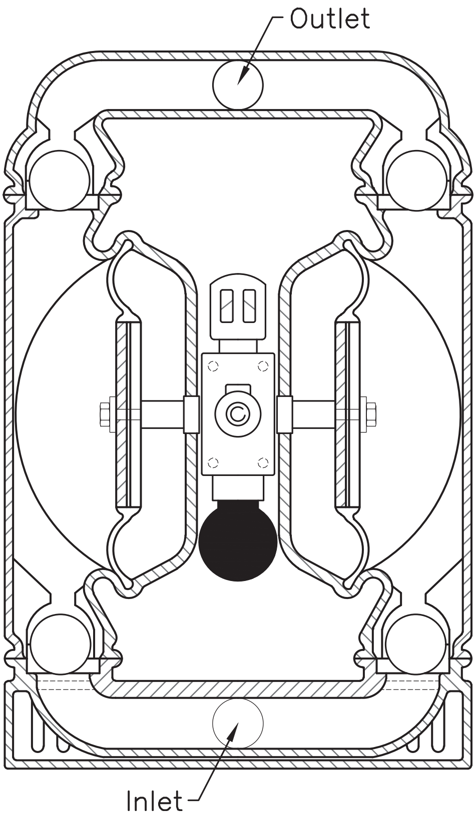


Fig. 4.12 Schematic diagram of a diaphragm pump.

pressure (up to 30,000 psig) (206,843 kPa). The volume must be infinitely controlled between limits and virtually independent of discharge pressure.

Diaphragm pumps are driven by fixed-speed electric motor through an integral reduction gear. Capacity control is achieved by adjusting the pump stroke. Air-driven are also commonly used in upstream operations. Proportioning mechanisms are usually integrated with the drive mechanism.

Diaphragm pumps provide an effective solution to leakage problems and, to a lesser extent, abrasive problems. There are two kinds of diaphragm pumps.

- Mechanical diaphragm drive (Fig. 4.13).
- Hydraulic diaphragm drive (Fig. 4.14).

Although mechanical drives are both simple and cheap, they have a short diaphragm life and are only suitable for very light duty (automobile fuel pumps) applications and will not be discussed further in this section.

Hydraulic drives may have a single or double diaphragm. A single diaphragm is the most common and suitable for most services (Fig. 4.14). However, a double diaphragm may be necessary for extremely toxic services.

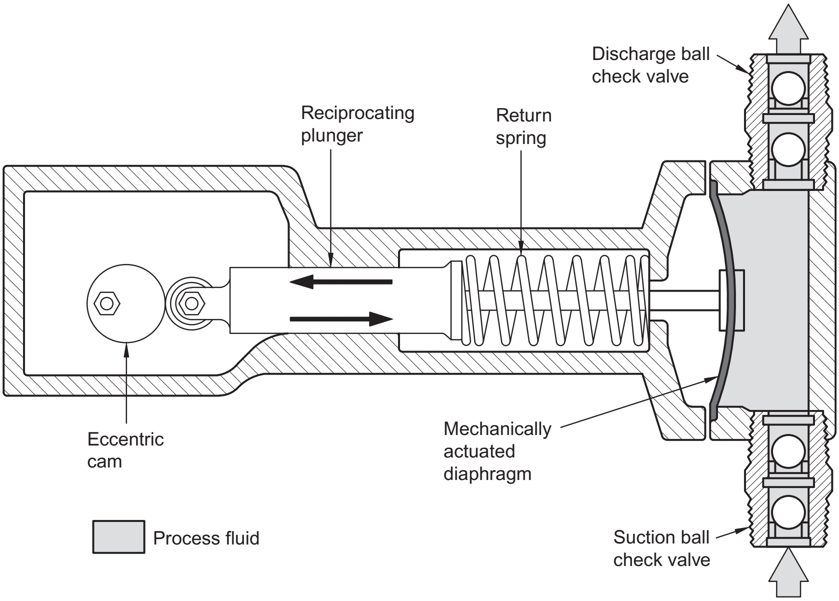


Fig. 4.13 Schematic diagram of a mechanical diaphragm driver. Courtesy of Marcel Dekker, Inc.

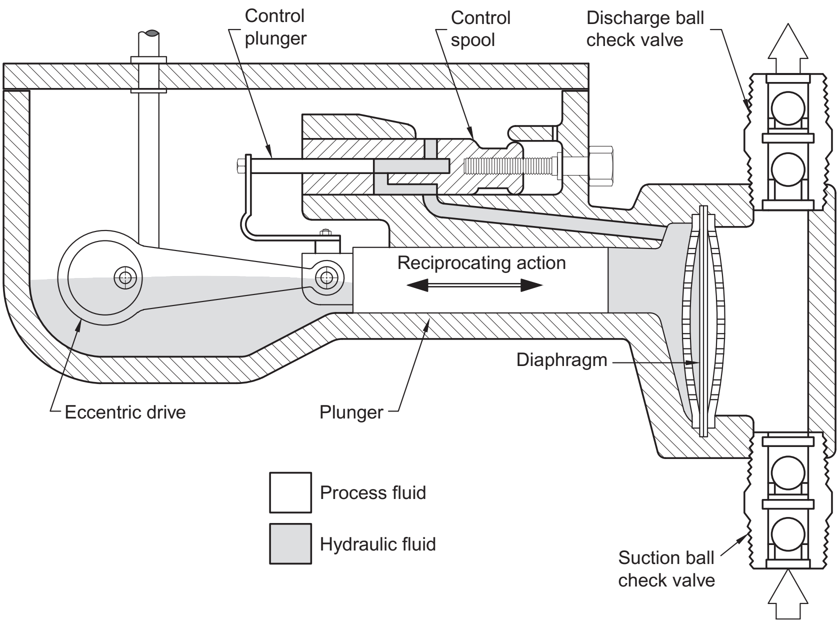


Fig. 4.14 Schematic diagram of a hydraulic diaphragm driver. Courtesy of Marcel Dekker, Inc.

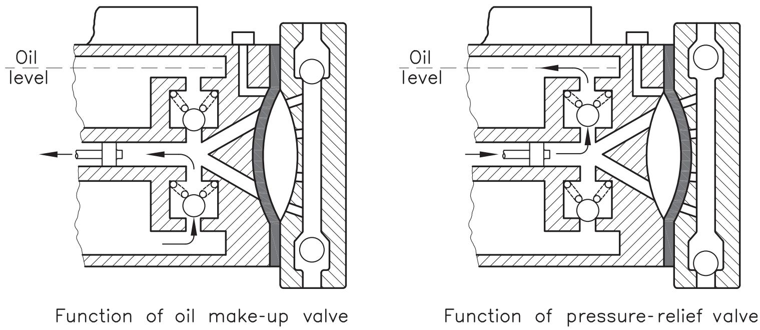


Fig. 4.15 Schematic diagram illustrating relief valve principles used in a diaphragm pump.

Diaphragms are usually made of Teflon or stainless steel, but elastomers or elastomer-coated steel diaphragms are also available. The double diaphragm provides positive isolation between the process fluid and the drive fluid (hydraulic oil). The diaphragm interspace may be designed with alarms to alert personnel to contamination by process fluid, for example, conductivity, which may indicate outer diaphragm failure.

To prevent diaphragm overstress, relief valves are incorporated into the drive system. Fig. 4.15 illustrates relief valve principles.

The hydraulic drive system looks similar to a packed plunger pump. However, it has a number of advantages over the plunger pump:

- The plunger works in an ideal fluid, that is, good lubricity, clean, and so on.
- The hydraulic drive uses relief valves to avoid diaphragm overstress. This feature is the equivalent of a discharge pressure shut off.
- It pumps corrosive and abrasive materials with much lower wear rates and better reliability than packed plunger pumps.
- Field repairs can be made quickly.

4.1.3.2.1 Advantages

Diaphragm pumps have the following advantages:

- Capable of pumping fluids that are viscous, erosive, corrosive, or contain large amounts of suspended solids.
- Long life and are inexpensive to repair since they have no stuffing box and have few moving parts.
- Can handle low flow rates inexpensively.
- Self-priming and can run periodically without any liquid.

4.1.3.2.2 Limitations

Diaphragm pumps have the following limitations:

- Limited to small flow rates (90 gpm) (21 m^2/h), moderate discharge pressures [1000 ft (305 m) of head], and moderate temperatures.
- Require frequent maintenance because they are reciprocating pumps.

- Diaphragm has a tendency to *fatigue* with time (diaphragm life is usually more than several hundred thousand but <2 to 3 million cycles).
- Diaphragm leakage can cause a hazard by mixing power gas with the process fluid.

Sometimes air or natural gas is used to power a diaphragm, which in turn drives a reciprocating pump. In this situation it is possible to handle large discharge pressures, but only if the flow rate is very small. Typically, this type of pump is used as sump pumps or used for chemical injection in field gathering and treating operations.

4.1.3.2.3 Application guidelines

Pumps may be arranged with hydraulic ends in parallel and/or series. Parallel operation helps achieve desired volumes. One major advantage of parallel operation is that the pulsating flow characteristic of a single unit can effectively "smooth" by carefully phasing the drive for each liquid end.

Series pumping is rare but required for high pressure [>1000 psi (68 bar)].

Fig. 4.16 summarizes diaphragm pump application guidelines.

4.2 Performance considerations

4.2.1 Pressure

Reciprocating pumps are constant volume pumps. Variations in discharge pressure do not affect flow rate. Since these pumps continue to deliver the same capacity, any attempt to throttle the discharge flow may overpressure the pump casing and/or discharge piping. Thus no reciprocating pump should ever be started or operated with the discharge block valve closed. Flow is regulated by speed. In rare cases that require a discharge throttle valve, an automatic bypass valve that is piped back to the suction source must be provided.

Fig. 4.17 is a schematic diagram illustrating how a force of 10,000 pounds (44,482,216 N) exerted on a piston with an area of 50 in^2 (323 cm^2) yields a discharge pressure of 200 psi (1379 kPa)

4.2.2 Capacity

The capacity (q) of a reciprocating pump is the total volume of fluid actually delivered per unit of time. The volume includes liquid and any dissolved or entrained gas at the stated operating conditions. Pump capacity is calculated from the following

$$q = D(1 - S) \tag{4.1}$$

Field units

$$q = dnm(1 - S) \tag{4.2a}$$

SI units

$$q = 3600dnm(1 - S) \tag{4.2b}$$

Metering pump application criteria		
Liquid end type	**Recommended for**	**Not recommended for**
Packed plunger	• Very high discharge pressure	• Corrosive fluids
	• Temperature over 250°F	• Abrasive fluids
	• Low vapor pressure fluids	• Applications whose trace contamination of pumpage with packing lubricants is not permitted
Single diaphragm (hydraulic drive)	• Corrosive fluids • Applications requiring high reliability	• High discharge pressure (>1500 psi) • Temperatures over 175°F (elastomer diaphragm) • Over 250°F (teflon diaphragm)
Double diaphragm	• Very corrosive or hazardous fluids	• High discharge pressure (>1000 psi)
	• Abrasive fluids	
	• Duties requiring guaranteed isolation from drive oil	
	• Applications requiring early warning of diaphragm failure	

(A)

Comparison of metering pumps		
Pump type	**Advantage(s)**	**Disadvantage(s)**
Mechanical diaphragm	• Least expensive • Can handle most fluids • Glandless	• Accuracy not as good as plunger and piston-diaphragm pumps • Diaphragm subject to fatigue failure • Limited to low pressure deliveries
Piston diaphragm	• High accuracy • Good diaphragm life • Can handle most fluids • Glandless • Readily rendered in double diaphragm form for fail-safe characteristics	• Diaphragm may be subject to wear if pumped fluids include particulate matter
Electromagnetic-driven diaphragm	• Particularly suitable for micro-metering with precise pulse operation • Glandless	• Limited capacity • More complex control system (needs digital signal input)

(B)

Fig. 4.16 (A) Diaphragm pump application guidelines and (B) comparison of diaphragm pumps.

where

q = pump capacity, gpm (m^3/h), delivered by all pistons, plungers, or diaphragms
D = pump displacement, gpm (m^3/h)
S = slip, fraction
d = displacement per pumping chamber, gal (m^3)
m = number of pistons, plungers, or diaphragms
n = pump speed, rpm (rps)

The displacement (D) of a reciprocating pump cylinder or chamber is the calculated volume displaced per chamber per single stroke of the piston, plunger, or diaphragm

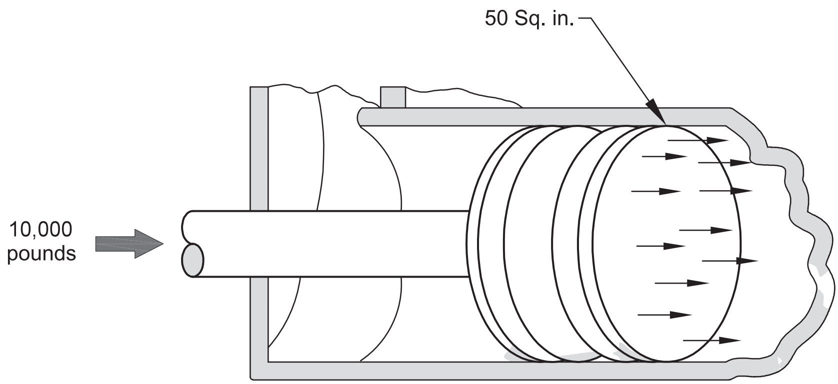


Fig. 4.17 Schematic diagram illustrating a force of 10,000 pounds (44,482,216 N) acting on a piston with area of 50 in^2 (323 cm^2) yields a discharge pressure of 200 psi (1379 kPa).

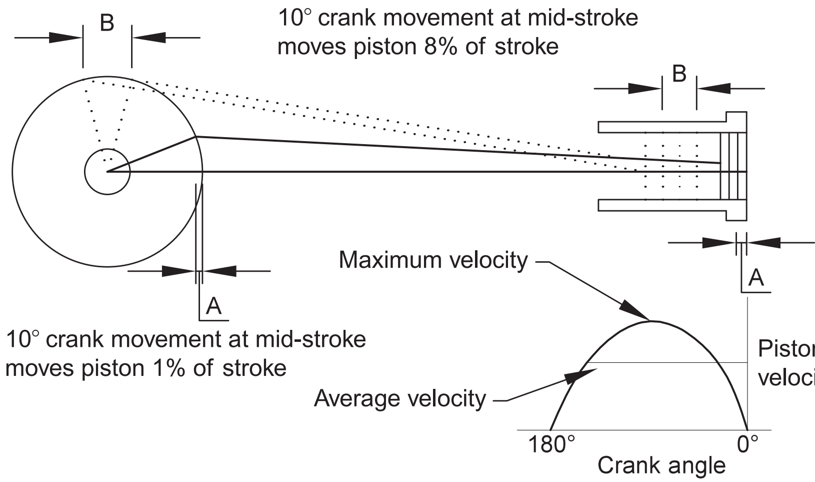


Fig. 4.18 Schematic diagram illustrating "cylinder displacement."

(Fig. 4.18), with no losses due to slip or fluid compressibility. Displacement of an entire pump is the total displacement of all the chambers when the pump is operating at a specified speed. Deductions for piston rod volume should be made on double-acting piston pumps when displacement is calculated.

Cylinder displacement for a *single-acting* cylinder is given by

Field units

$$D = \frac{ASNm}{231} \tag{4.3a}$$

SI units

$$D = \frac{ASNm}{277{,}808} \tag{4.3b}$$

Cylinder displacement for a *double-acting* cylinder is given by
Field units

$$D = \frac{(2A - a)SNm}{231} \tag{4.4a}$$

SI units

$$D = \frac{(2A - a)SNm}{277{,}808} \tag{4.4b}$$

where
D = cylinder displacement, gpm (m^3/h)
A = plunger or piston area, in^2 (mm^2)
a = piston rod cross-sectional area, in^2 (mm^2)
S = stroke length, in (mm)
N = speed, rpm (rps)
m = number of pistons or plungers

The stroke of a reciprocating pump is the number of complete pumping cycles of the plunger, piston, or diaphragm per minute (stokes/min).

Slip is the loss of capacity as a percentage of the cylinder displacement due to

- volumetric efficiency
- stuffing box losses
- valve losses

Volumetric efficiency (not to be confused with mechanical efficiency) for pistons is normally 95% to 97% and lower for plungers. Efficiency is also reduced when pumping a light hydrocarbon that has some degree of compressibility.

Pump capacity can be determined from the following:

$$Q = \frac{D(100 - Slip)\%}{100} \tag{4.5}$$

Table 4.1 presents typical values of slip for various viscosities.

4.2.3 Speed

Speed is the primary factor in the determination of the capacity of a reciprocating pump. The primary constraint in determining a pump's speed is the average velocity (V_{AVG}) of the reciprocating elements, which can be determined by

Table 4.1 Pump efficiency reductions due to high-viscosity fluids

Centistokes	Viscosity (SSU)	Slip (%)
215	1000	90
430	2000	80
865	4000	70
1300	6000	62
1725	8000	55
2160	10,000	50

$$V_{AVG} = N\left(\frac{S}{6}\right) \tag{4.6}$$

where

V_{AVG} = average velocity, ft/min (m/s)

Running at high speeds shortens packing life. Table 4.2, based on field tests and observations, lists recommended maximum continuous service speeds and velocities.

It is permissible to interpolate between the values given in Table 4.2. Higher velocity values may be used, but only for intermittent and cyclic duty. Lower speeds should be used when the NPSH margin is <2 ft.

4.2.4 Viscosity

In general, reciprocating pumps are capable of handling liquids of up to about 100 centistokes (475 SSU) without any reduction in speed and capacity, although the efficiency will be slightly lower than with water. At higher viscosities, efficiency is likely to fall still further because of increased resistance to flow through ports and valves. It is therefore usual to operate reciprocating pumps at reduced speed when handling high-viscosity fluids.

Table 4.2 Recommended maximum continuous service speeds and velocities

Stroke (S), in. (mm)	Speed (N), rpm (rps)	(V_{AVG}), fpm (m/s)
1 (25)	500 (30,000)	83 (25)
2 (51)	450 (27,000)	150 (45)
3 (76)	400 (24,000)	200 (61)
4 (102)	350 (21,000)	233 (71)
5 (127)	310 (18,600)	258 (79)
6 (152)	270 (16,200)	270 (82)
7 (178)	240 (14,400)	280 (85)
8 (203)	210 (12,600)	280 (85)

Table 4.3 Pump efficiency reductions due to high-viscosity fluids

Viscosity, SSU (centistokes)	% of Maximum speed
<300 (<65)	100
300–1000 (65–220)	90
1000–2000 (220–440)	80
2000–4000 (440–880)	70
4000–6000(880–1320)	60
6000–8000(1320–1760)	55
8000–10,000(1760–2200)	50

Courtesy of Hydraulic Institute.

High viscosity also reduces the slip parameter, S, reducing capacity as presented in Table 4.3. These values are illustrative only, as much depends on the size and detailed design of the pump.

Operating below maximum "rated" speed may be advantageous when:

- Pump is operated unattended
- Pump has no spares and no standby
- There is a high penalty for down time
- Unit is remotely located
- Unit maintenance is poor
- Long life is desired
- NPSH margin is low

Operating at the maximum "rated" speeds requires

- Clean, cool fluids
- Excellent piping layout with rigidly fixed piping
- Good operating environment and solid foundation
- Adequate NPSH margin
- Well-designed suction stabilizers and discharge dampeners
- Good maintenance

Whenever it becomes necessary to operate above the maximum "rated" speed, very close attention should be given to all design, operation, and maintenance details.

4.2.5 Net positive suction head ($NPSH_R$)

For a reciprocating pump, energy is required to push the suction valve from its seat and to overcome the friction losses and acceleration head. Since these losses can be significant (especially at low speeds), $NPSH_R$ is expressed in terms of pressure (psi) rather than head (feet).

$NPSH_R$ is a function of many factors including

- Liquid density
- Vapor pressure
- Viscosity

- Dissolved air/gas
- Pump speed
- Piston diameter
- Valve type and flow area
- Valve spring load and spring rate
- Liquid passage configuration
- Stuffing box leakage

The $NPSH_A$ is the total suction head at the pump suction flange less the absolute vapor pressure of the liquid being pumped. The NPSH margin is defined as the difference between $NPSH_A$ less $NPSH_R$. $NPSH_A$ must exceed $NPSH_R$. $NPSH_R$ is affected by changes in

- Plunger size
- Pump speed
- Flow rate
- Suction pressure

A minimum of 7-ft (2.1 m) of NPSH margin is recommended. Additional NPSH margin may be required when handling:

- high vapor pressure fluids
- fluids with dissolved air or gases, or
- fluids with suspended solids

If a reciprocating pumping system exhibits inadequate NPSH margin and cannot be redesigned, it may be necessary to install a suction stabilizer or stand pipe near the pump inlet, reduce the operating speed or increase the source tank liquid level or operating pressure.

Cavitation occurs in a pump when the pressure of the liquid is reduced to a value equal to or below its vapor pressure and small vapor bubbles or pockets begin to form. As these vapor bubbles are compressed to a higher pressure they rapidly collapse. The forces during the collapse are generally high enough to cause fatigue failure of pump valves and under severe conditions can cause serious pitting damage to the pump internals.

The accompanying valve damage and pressure pulsations in the piping system are the most easily recognized signs of cavitation. Cavitation also results in reduced capacity due to the vapor present in the pump. Vibration and mechanical damage to the pump and piping system can also occur as a result of operating in cavitation.

The only way to prevent the undesirable effects of cavitation is to ensure that the $NPSH_A$ in the system is greater than the $NPSH_R$ by the pump. $NPSH_A$ for a reciprocating pump application is calculated in the same manner as for a centrifugal pump, with the exception that, in determining the $NPSH_R$ for a reciprocating pump, some additional allowance must be made for the reciprocating actin of the pump. This additional requirement is termed "acceleration head," which is defined as the head required to accelerate the liquid column on each suction stroke so that there will be no separation due to liquid inertia in the pump or suction line.

If this minimum condition is not met, the pump will experience a fluid knock caused when the liquid separates and then overtakes the receding plunger. This knock occurs approximately two-thirds of the way through the suction stroke. If sufficient head is provided for the liquid to completely follow the motion of the receding face of the plunger, this knock will disappear. NPSH must always have a positive value and can be calculated by the following equations:

For suction lift (liquid supply level below pump centerline)

$$NPSH_A = H_{PA} - H_{VPA} - H_{VH} - H_{F1} - H_1 - H_A \tag{4.7}$$

For "flooded" suction head (liquid supply level above pump centerline)

$$NPSH_A = H_{PA} - H_{VPA} - H_{VH} - H_{F1} + H_1 - H_A \tag{4.8}$$

where

$NPSH_A$ = net positive suction head available, ft (m)
H_{PA} = absolute head on the surface liquid. This will be barometric pressure if suction is from an open or the pressure existing in a closed tank or vessel, ft (m)
H_{VPA} = absolute vapor pressure head at the temperature being pumped, ft (m)
H_{VH} = velocity head in the pump suction piping minus the velocity head in the suction supply, ft (m)
H_{F1} = friction head in the suction line, including entrance losses from the fluid storage vessel to the piping and friction losses through pipe, valves, and fittings, and so on, ft (m)
H_1 = static head that the liquid supply level is above or below the pump centerline, ft (m)
H_A = acceleration head, the head required to accelerate the liquid column on each stroke so that there will be no separation of this liquid in the pump or suction line, ft (m). Always subtracted from the $NPSH_A$

When pumping from a production vessel the liquid is normally in equilibrium with the gas in the vapor space and thus:

$$H_{PA} = H_{VPA} \text{ and } H_{PA} - H_{VPA} = 0 \tag{4.9}$$

The $NPSH_R$ is determined by the pump manufacturer and will depend on many factors, including losses in the suction valves of the pump, head needed to propel stationary fluid into motion at the average liquid velocity of the pump inlet port, and so on. The $NPSH_R$ of a reciprocating pump is determined with water at 60°F (15.6°C). The $NPSH_A$ is lowered with the pump at constant speed and throughout until there is a 3% reduction in throughput. The $NPSH_A$ at this point is the specified $NPSH_R$ for this speed. Tests are repeated at different speeds to draw a curve of $NPSH_R$ versus speed.

The $NPSH_A$ depends on the system layout and must always be equal to or greater than the $NPSH_R$. On an existing installation the $NPSH_A$ is the reading of a gauge at the suction flange converted to height of liquid absolute and corrected to the pump centerline elevation less the sum of the vapor pressure of the liquid in height absolute and the velocity head in height of liquid at the point of gauge attachment and the acceleration head.

Field experience and laboratory tests have confirmed that pumps handling gas-free hydrocarbon fluids and water at elevated temperatures will operate satisfactory with little or no cavitation at lower $NPSH_A$ than is required for cold water. Fig. 4.19 shows NPSH reductions that may be considered for hot water and gas-free pure hydrocarbon liquids.

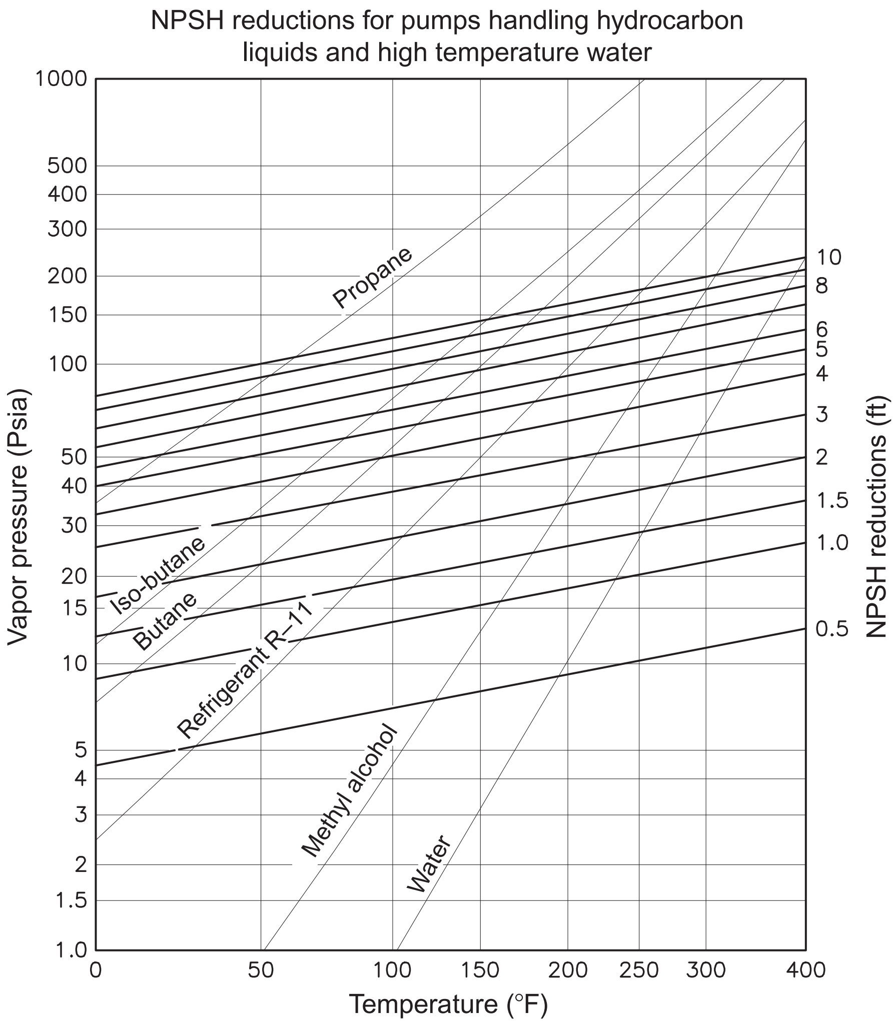


Fig. 4.19 NPSH reductions for pumps handling hydrocarbon liquids and high-temperature water.
Courtesy of Hydraulic Institute.

The use and application of Fig. 4.19 is subject to the following limitations:

(1) The NPSH reductions shown are based on laboratory test data at steady-state suction conditions and on the gas-free pure hydrocarbon liquids shown. Its application to other liquids must be considered experimental and is not recommended.

(2) No NPSH reduction should exceed 50% of the $NPSH_R$ for cold water or 10 ft (3 m), whichever is smaller.

(3) In the absence of test data demonstrating NPSH reductions >10 ft (3 m), the chart has been limited to that extent; extrapolation beyond that is not recommended.

(4) Vapor pressure for the liquid is true vapor pressure at bubble point and not Reid vapor pressure.
(5) Do not use the chart for liquids having entrained air or other noncondensable gases which may be released as the absolute pressure is lowered in the pump suction manifold, in which case additional NPSH may be required for satisfactory operation.
(6) In the use of the chart for high-temperature liquids, particularly with water, due consideration must be given to the susceptibility of the suction system to transient changes in temperature and absolute pressure which might require additional NPSH to provide a margin of safety, far exceeding the reduction otherwise permitted for steady-state operation.

Subject to the earlier limitations, which should be reviewed with the manufacturer, the procedure in using the chart is illustrated as follows:

Assume a pump with a specified $NPSH_R$ of 16 ft (5 m) as the design capacity is to handle pure propane at 55°F (12.8°C). From Fig. 4.19, propane at 55°F (12.8°C) has a vapor pressure of ~100 psia (689.5 kPa), and a reduction of 9.5 ft (2.9 m) could be considered. This reduction, however, is greater than one half the cold water $NPSH_R$. Thus the corrected value of the $NPSH_R$ is one half the cold water $NPSH_R$ or 8 ft (2.4 m). If this same pump has an application to handle propane 14°F (10°C), where its vapor pressure is 50 psia (345 kPa), a reduction of 6 ft (2 m) could be considered as shown in Fig. 4.19. This is less than one half of the cold water $NPSH_R$. The corrected value of $NPSH_R$ is, therefore, 16 ft less 6 ft, or 10 ft (5 m less 2 m, or 3 m) for this second case.

4.2.6 Acceleration head

Fluid in the piping has to accelerate and decelerate a number of times for each revolution of the crankshaft. Sufficient additional energy must be provided on the suction side of the pump to prevent cavitation. This energy takes away from the energy already existing at the pump suction. The head required to accelerate the fluid column is a function of the suction line, the average velocity in this line, the rotating speed, the type of pump, and the relative elasticity of the fluid. Thus the amount of $NPSH_A$, calculated previously, is reduced by an amount equal to

Field units

$$H_a = \frac{LVNC}{kg} \tag{4.10a}$$

SI units

$$H_A = \frac{60LVNC}{kg} \tag{4.10b}$$

where

H_a = acceleration head, ft (m)
L = actual length of suction line, ft (m)
V = velocity in suction line, ft/s (m/s)
N = speed, rpm (rps)
C = constant for the type of pump (from Table 4.4)
k = constant for relative fluid compressibility (from Table 4.5)
g = acceleration of gravity, 32.3 ft/s^2 (9.81 m/s^2)

Table 4.4 Constant "C" used for determining acceleration head

Pump type	**"C" constant**
Simplex, double-acting	0.200
Duplex, single-acting	0.200
Duplex, double-acting	0.115
Triplex	0.066
Quintuplex	0.040
Septuplex	0.028
Nonuplex	0.022

Table 4.5 Constant "k" used in determining acceleration head

Fluid(s)	**"k"**
Amine, glycol, water	1.5
Most hydrocarbons	2.0
Hot oils, ethane	2.5

The factor "C" depends upon the type of pump used as follows

The value of "k" can be *estimated* from the following:

The acceleration head, where L is ≤ 5 ft (1.5 m) is slightly conservative. When L exceeds 10 ft (3 m) the acceleration head is moderately conservative and becomes overly conservative when L exceeds 50 ft (15 m). The equation assumes piping is rigid and not elastic. The equation does not account for pressure valve velocities.

Example 4.1. Determination of acceleration head (field units)

Given:

A triplex reciprocating pump is required to inject 6000 barrels of salt water per day. The pump operates at 360 rpm. The suction line is 10 ft. of 4-in., Schedule 40 pipe.

Determine:

Calculate the head required to accelerate the fluid through the suction pipe (assume $g = 32.17\,\text{ft/s}^2$).

Solution:

(1) Determine the water displaced by the pump in gpm.

$$q = \frac{(6000)(42)}{(60)(24)}$$
$$= 175\,\text{gpm}$$

(2) Determine the average velocity in the suction piping.

$$v = \frac{(0.4085)(q)}{d^2}$$
$$= \frac{(0.4085)(175)}{4.026^2}$$
$$= 4.41\,fps$$

(3) Determine the acceleration head.

$$H_a = \frac{LVNC}{kg} = \frac{(10)(4.41)(360)(0.066)}{(1.5)(32.17)} = 21.7\,\text{ft}$$

Thus 21.7 ft must be subtracted from the value of calculated $NPSH_A$. The NPSH margin must be provided unless the piping system is accurately defined.

4.2.7 Power and efficiency

The hydraulic power is calculated in a similar fashion as centrifugal pumps. The differences are

- pressure rather than head
- zero suction pressure
- higher efficiencies

The hydraulic power that must be developed by the pump is given by the following equations:
Field units

$$HHP = \frac{(TDH)(\rho)(Q)}{550} \tag{4.11a}$$

SI units

$$HHP = \frac{(TDH)(\rho)(Q)}{367{,}000} \tag{4.11b}$$

where

HHP = hydraulic horsepower, hp (kW), (1 hp = 550 ft-lb/s)
TDH = pump head, ft (m)
ρ = density of liquid, lb/ft^3 (kg/m^3)
Q = flow rate, ft^3/s (m^3/s)

By making the appropriate unit conversions Eqs. (4.11a, 4.11b) may be expressed as:
Field units

$$HHP = \frac{(TDH)(SG)q}{3960} \tag{4.12a}$$

SI units

$$HHP = \frac{(TDH)(SG)q}{367.6} \tag{4.12b}$$

Field units

$$HHP = \frac{q\Delta p}{1714e} \tag{4.13a}$$

SI units

$$HHP = \frac{q(\Delta P)}{3600} \tag{4.13b}$$

Field units

$$HHP = \frac{Q(\Delta P)}{58,766} \tag{4.14a}$$

SI units

$$HHP = \frac{q(\Delta P)}{3600} \tag{4.14b}$$

where

q = capacity, gpm (m^3/h)
Q = capacity, bpd (m^3/h)
ΔP = maximum discharge pressure of pump, psig (kPa) (for driver sizing, pump MAWP is used)
SG = specific gravity relative to water

For design, a conservative assumption is that the driver must be sized for the MAWP of the pump case. This is done in the event of accidental closing of the pump discharge valve.

The input power to the shaft of the pump is called the brake horsepower and is given by

$$BHP = \frac{HHP}{E_m} \tag{4.15}$$

where

BHP = brake horsepower, hp (kW)
E_M = pump mechanical efficiency (%)

Tables 4.6 and 4.7 provide approximate values to be used for mechanical efficiency based on either pump speed or developed pressure.

4.3 Pump classification

There are two types or reciprocating pumps: power pumps and direct-acting pumps. The driver for a power pump has a rotating shaft such as a motor, engine, or turbine. Power pumps reciprocate the pumping element with a crank or camshaft.

Table 4.6 Effect of speed on mechanical energy at constant developed pressure

Percent of full speed	44	50	73	100
E_M (%)	93.3	92.5	92.5	92.5

Table 4.7 Effect of pressure on mechanical efficiency at constant speed

Percent of full speed developed pressure	20	40	60	80	100
E_M (%)	82	88	90.5	92	92.5

Direct-acting pumps are driven by pressure from a motive gas. Direct-acting pumps were originally known as steam pumps because steam was the motive fluid.

Reciprocating pumps are typically classified by

Type of drive

- direct-acting, gas driven (Figs. 4.5 and 4.7)
- crank driven (power pump) (Figs. 4.6 and 4.8)

Cylinder orientation

- horizontal
- vertical (used for higher horsepower requirements)

Liquid end arrangement

- plunger (outside packing)
- piston (inside piston rings and packing on the piston rod)
  - drives the pumping element through a crank or camshaft
  - power sources include *motors and engines*

Number of pistons or plungers

- simples
- duplex
- triplex
- quintuplex, etc.

Type of action

- single-acting (delivers on either forward or backward stroke, not both)
- double-acting (delivers on both forward and backward strokes)

Fig. 4.20 illustrates the classifications. Pumps using reciprocating motion, such as rotary or metering pumps, are not discussed here.

4.3.1 Single- and double-acting pumps

Single-acting pumps discharge on either the forward or return stroke of the piston or plunger; every cycle of the pump displaces only one volume of liquid. In double-acting pumps, liquid is discharged on both the forward and return stroke of the piston. Plunger pumps are only single-acting; piston pumps can be either single- or double-acting. Fig. 4.21 illustrates these pumps' actions.

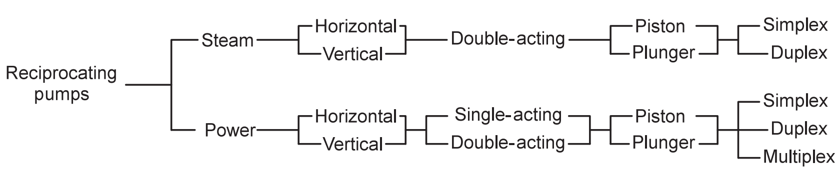


Fig. 4.20 Reciprocating pump types.

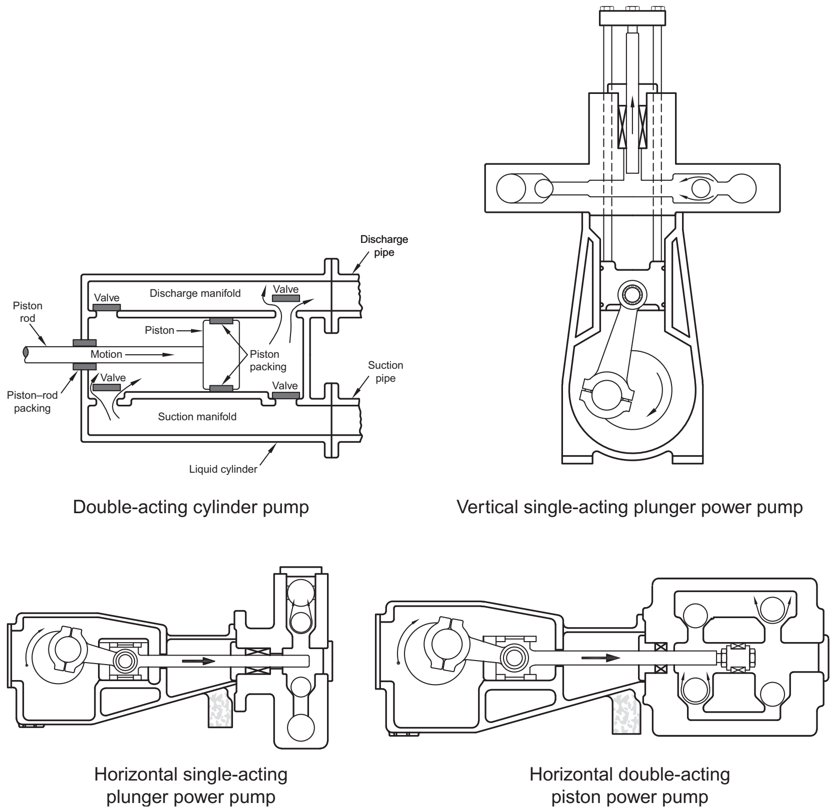


Fig. 4.21 Typical single- and double-acting pumps.
Courtesy of the Hydraulics Institute.

Simplex, duplex, and multiplex refer to the number of piston and rod assemblies in a pump. Simplex pumps have one piston and rod assembly; duplex pumps have two; multiplex pumps have three or more.

Unlike the centrifugal pump, which is a kinetic energy machine, the reciprocating pump does not require velocity to achieve pressure. This is one of the reciprocating pump's advantages, particularly for abrasive, slurry, and high-viscosity applications. High pressures can be obtained at low velocities, and fluid capacity varies directly with pump speed.

The discharge pressure of a reciprocating pump is only that required to force the desired volume of liquid through the discharge system. Within the constraints of pump construction, the maximum pressure developed for gas-driven pumps is limited only by the differential gas pressure available; for crank-driven pumps, the driver torque is the only limit.

The flow of liquid from a reciprocating pump pulsates, varying both in flow rate and pressure. As the piston or plunger moves back and forth in the cylinder, alternatively opening and closing the suction and discharge valves, a cyclic pulsation is set up in the suction and discharge lines of the pump. Fig. 4.22 shows the changes in flow rate

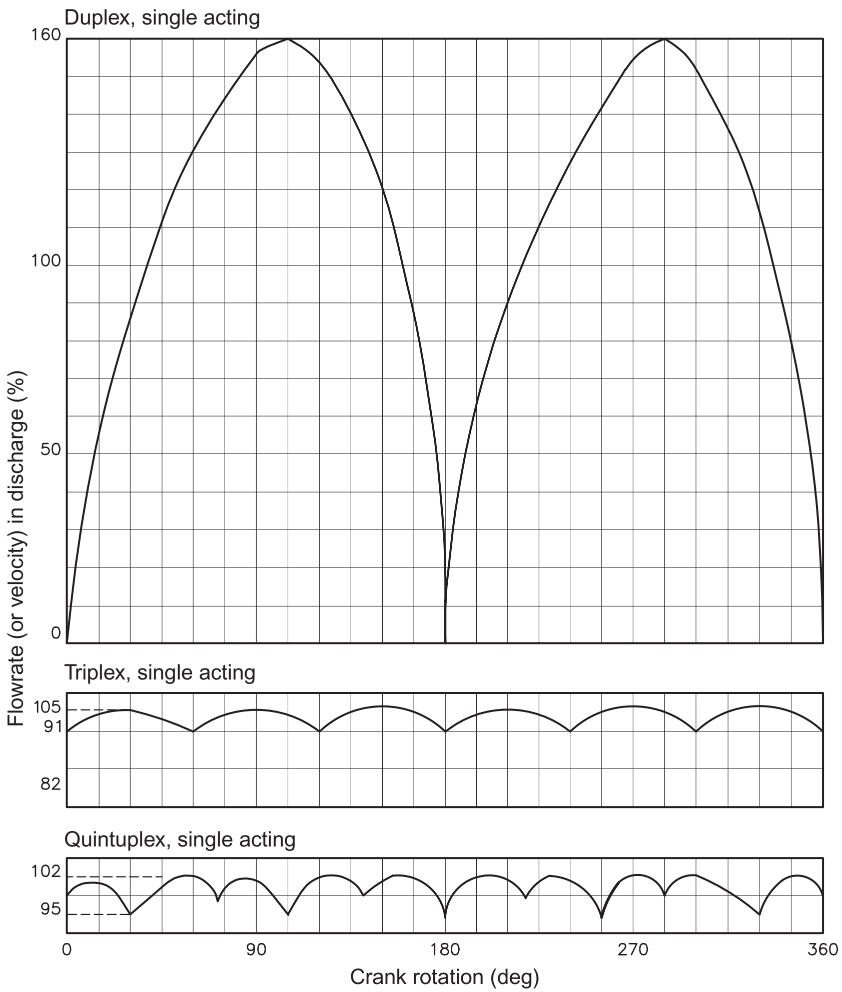


Fig. 4.22 Flow rate per stage.

as a function of crank angle for duplex, triplex, and quintuplex single-acting pumps. These changes become less severe as the number of stages increases.

4.3.2 Reciprocating pump selection charts

Pump selection charts is the equivalent to the centrifugal pump performance curve (Fig. 4.23A). Different manufacturers provide different forms. All charts consist of straight, capacity-speed lines because displacement is a direct function of speed. Speed is plotted against displacement for different piston/plunger diameters available in the pump. The maximum discharge pressures that each size piston/plunger could operate in order to fully stress the piston/plunger rod are listed across the top of the chart.

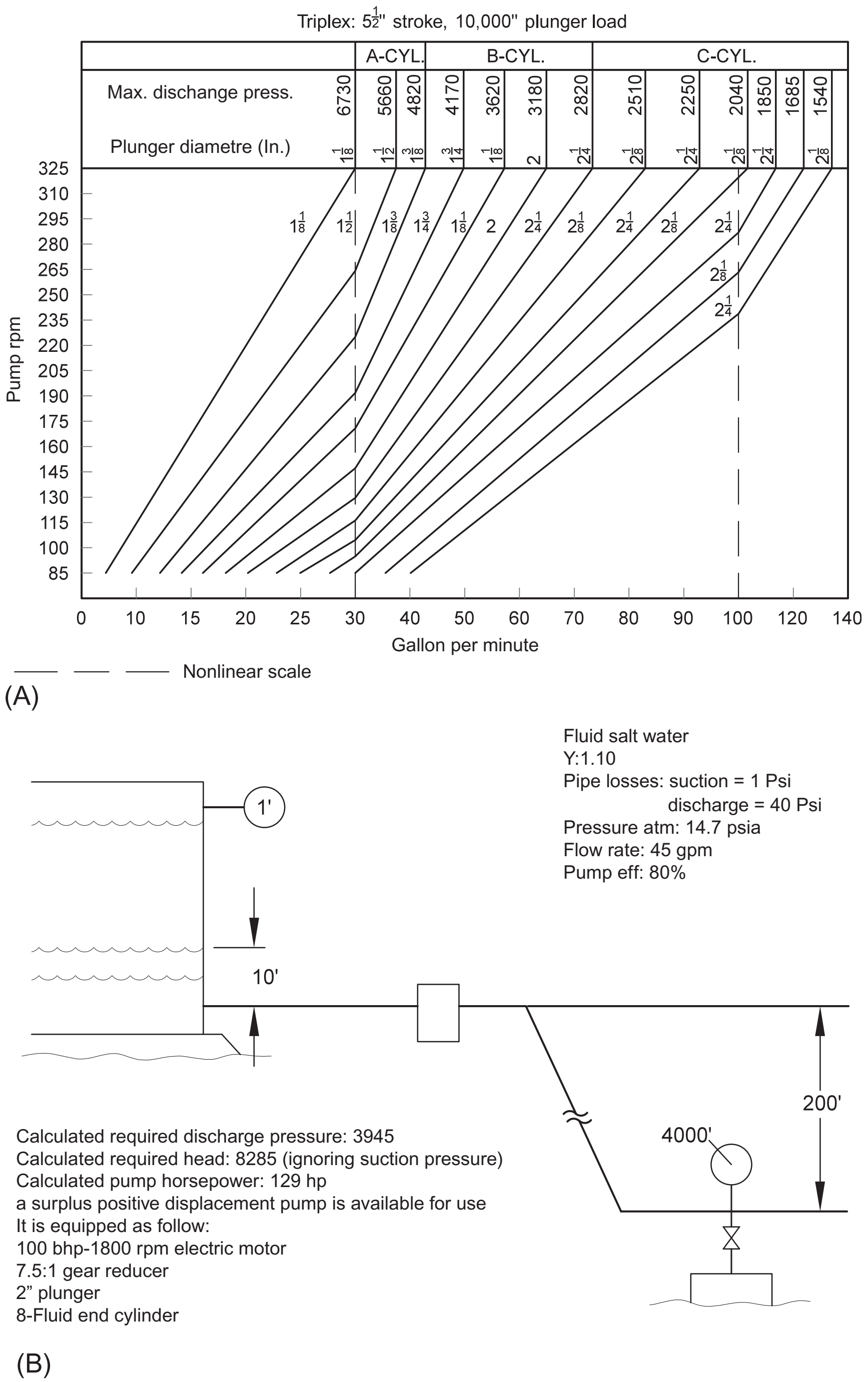


Fig. 4.23 (A) Typical reciprocating pump selection chart and (B) Example 4.2 problem schematic.

The plunger/piston load is the force transmitted to the power end by one piston/plunger for a single-acting pump and it can be calculated as

$$L = AP \tag{4.16}$$

where

L = plunger/piston load, Ibs (N)
A = area of face of plunger/piston, in^2 (cm^2)
P = discharge pressure, psig (kPa)

Example 4.2. Positive displacement pump selection (field units)

Given:

A positive displacement pump is required to waterflood a reservoir to increase oil production.

Determine:

Using Fig. 4.23A and B determine:

(a) The capacity and maximum discharge pressure of the pump.
(b) Does the pump, as currently configured, meet the system requirements? If not, what can be done?

Solution:

(1) Determine the surplus pump's operating speed.

$$\frac{1800}{7.5} = 240\,\text{rpm}$$

(2) Using Fig. 4.23A, determine the plunger's maximum capacity.
At 240 rpm, read 49 gpm. This exceeds the 45 gpm requirement.

(3) Determine if the pump meets the required discharge pressure of 3945 psi.
From Fig. 4.23A, the maximum plunger discharge pressure = 3180 psi. Thus the pump as configured *will not* provide the required injection pressure.

(4) Determine what can be done to meet the injection pressure requirement.
a. Change the plunger size to obtain an injection pressure ≥3945 psi. Using Fig. 4.23A, a 1-3/4-in. diameter plunger has a 4170 psi MAWP.
b. From Fig. 4.23A however, the 1-3/4-in. diameter plunger only produces 37 gpm at 240 rpm, and the flow rate requirement is 45 gpm. To obtain 45 gpm, the surplus pump's speed must be increased to 283 rpm.

(5) Determine the required horsepower for the new conditions in (4b) above.

$$BHP = \frac{(45\,\text{gpm})(4170\,\text{psi})}{(1714)(0.80)} = 137$$

Since 137 *BHP* motors are not standard, the next larger, standard motor is specified. Thus a new 150± *BHP* motor is needed. This is now a matter of project timing, costs, and economics whether to use the revamped surplus pump or to purchase a new packed unit.

4.4 Pump construction details

The mechanical components of a reciprocating pump vary according to design. Pumps are built in both horizontal (Fig. 4.24) and vertical (Fig. 4.25) construction. Horizontal pumps are available with capacities to 600 gpm (136 m^3/h) and discharge pressures up to 25,000 psi (172,000 kPa). Vertical pumps are available with capacities up to 5000 gpm (1140 m^3/h) and discharge pressures up to 10,000 psi (69,000 kPa).

The motor-driven or power input end of the pump is normally called the power end, and the pumping end is called the liquid end or fluid end. As shown in Figs. 4.26 and 4.27, the power end contains the following:

- frame
- crankshaft
- connecting rod
- crosshead
- pony rod
- bearings

As shown in Fig. 4.28, the liquid end contains the following:

- plunger
- piston
- stuffing box
- manifold
- cylinder

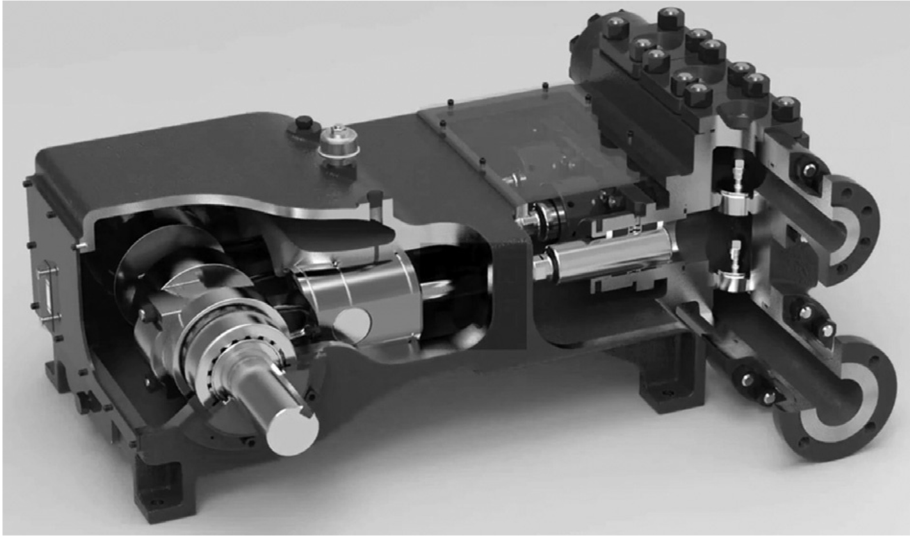


Fig. 4.24 Cutaway view of a typical horizontal pump.
Courtesy of GASO Pumps.

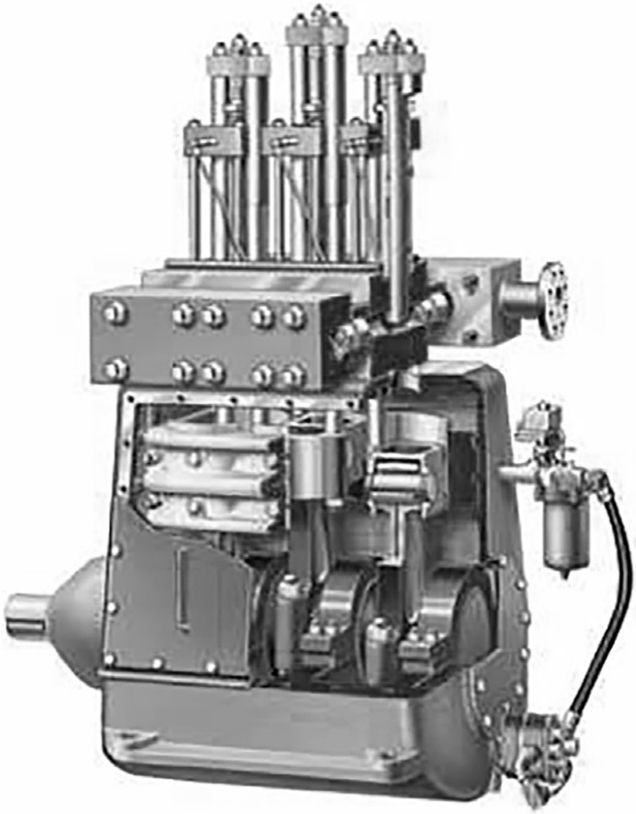


Fig. 4.25 Cutaway view of a typical vertical pump. Courtesy of Ingersoll-Rand Company.

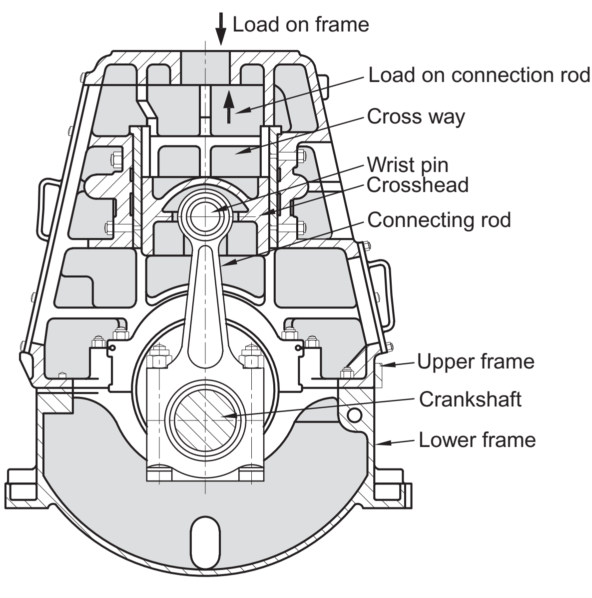


Fig. 4.26 Schematic diagram of the power end of a vertical pump. Courtesy of Ingersoll-Rand Company.

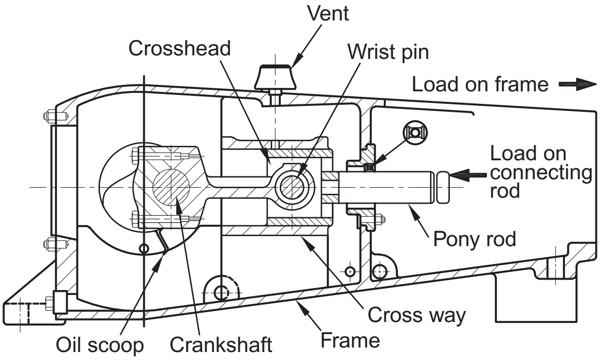


Fig. 4.27 Schematic diagram of the power end of a horizontal pump.
Courtesy of Ingersoll Rand Company.

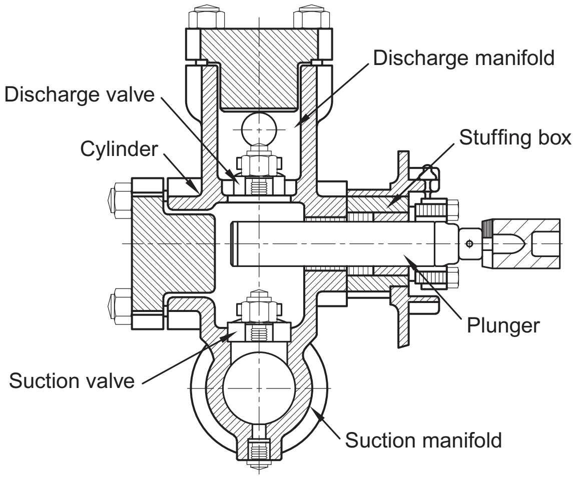


Fig. 4.28 Schematic diagram of the liquid end of a horizontal pump.
Courtesy Ingersoll Rand Company.

4.4.1 Power end components

4.4.1.1 Frame

The frame absorbs the plunger load and torque. On vertical pumps with an outboard stuffing box (Fig. 4.29), the frame is in compression. Lubrication is achieved by using an integral rotary gear pump in a wet sump much the same as in an engine. To avoid contamination of the lube oil with dirt, water, or pumped fluid, the crankcase is completely isolated by providing wiper packing for the rods and usually a chamber to accumulate leakage.

With horizontal single-acting pumps, the frame is in tension (Fig. 4.27). Frames usually are close-grain cast iron. Slurry pump frames, designed for mobile service at various sites, usually are fabricated steel. The frame is vented to the atmosphere. When the working atmosphere is detrimental to the working parts in the frame (e.g., ammonia attack on bronze bearings), the frame may be purged continuously with nitrogen.

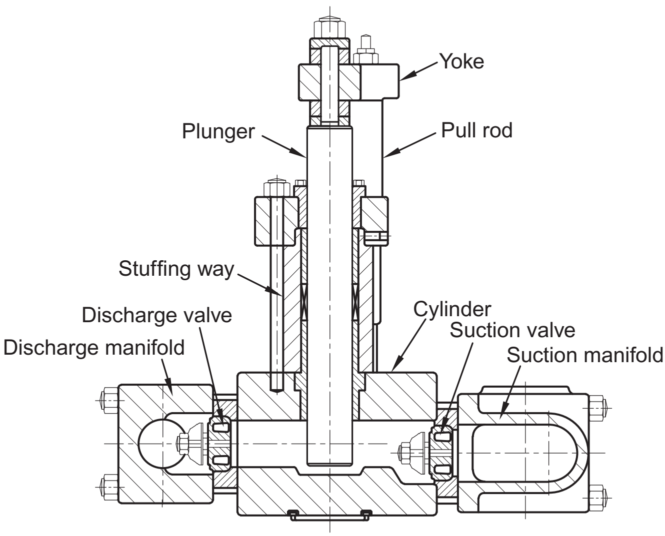


Fig. 4.29 Schematic diagram of the liquid end of a vertical pump.
Courtesy of Ingersoll Rand Company.

4.4.1.2 Crankshaft

The crankshaft (Figs. 4.30–4.32) varies in construction depending on design and power output. In horizontal pumps, the crankshafts are usually constructed of nodular iron or cast steel. Vertical pumps are constructed of forged steel or machined billet for high pressure. Since crankshafts operate at relatively low speeds and mass, counterweights are not used. Except in the case of duplex pumps, crankshafts are usually made with an odd number of throws to obtain the best pulsation characteristics. The firing order in a revolution depends upon the number of throws on the crankshaft, as presented in Table 4.8. Many designs incorporate rifle drilling between throws, thus furnishing lubrication to the crankpin bearings.

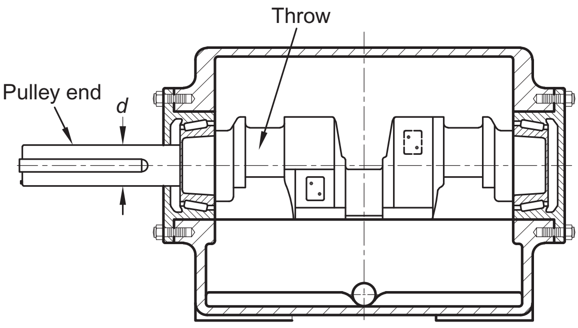


Fig. 4.30 Schematic diagram of a cast crankshaft.
Courtesy of Ingersoll Rand Company.

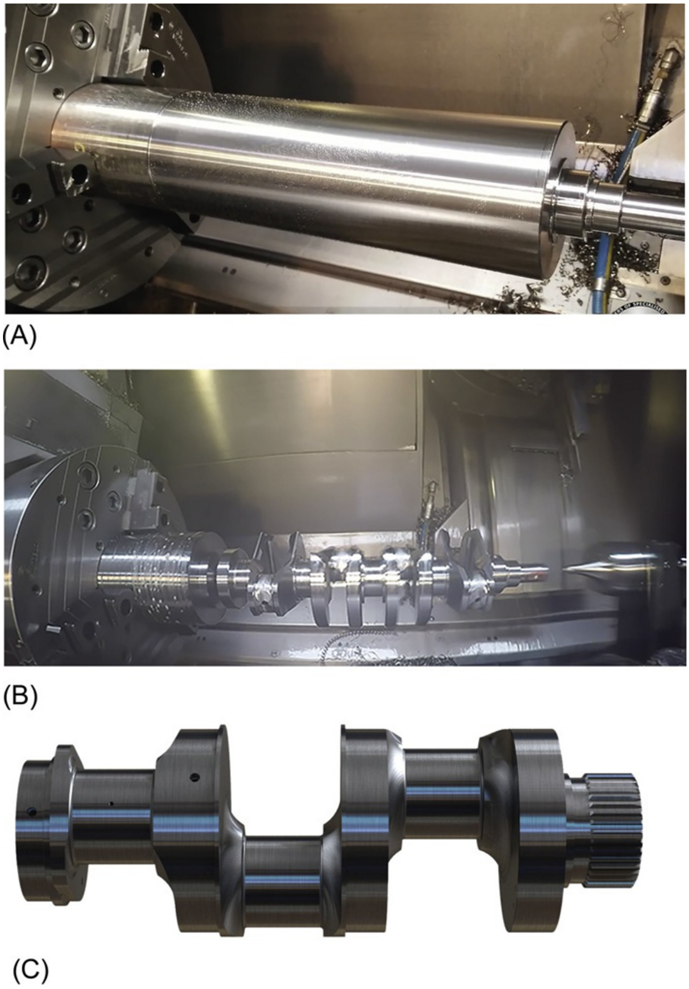


Fig. 4.31 Example of a crankshaft machined from a billet. Top: prior to machining; Center: billet during the machining process; and Bottom: final product.
Courtesy of Ingersoll Rand Company.

Fig. 4.32 Example of a cast crankshaft with an integral gear.
Courtesy of Continental EMSCO.

Table 4.8 Order of pressure buildup, or firing order

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

4.4.1.3 Connecting rods and eccentric straps

The connecting rods (Fig. 4.33) transfer the rotating force of the crank pin to an oscillating force on the wrist pin. Connecting rods are split perpendicular to their centerlines at the crank pin end for assembly of the rod onto the crankshaft. The cap and rod are aligned with a close-tolerance bushing or body-bound bolts. The rods may be rifle drilled or they may have cast passages for transferring oil from the wrist pin to the crank pin. A connecting rod with a tension load is made of forged steel, cast steel, or fabricated steel. Rods with a compression loading are cast nodular steel or aluminum alloy.

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

The eccentric strap (Fig. 4.34) has the same function as a connecting rod except that the former usually is not split. The eccentric strap is furnished with antifriction (roller or ball) bearings, whereas connecting rods are furnished with sleeve bearings. Eccentric straps are applied to mud and slurry pumps, which are started up against full load without requiring a bypass line.

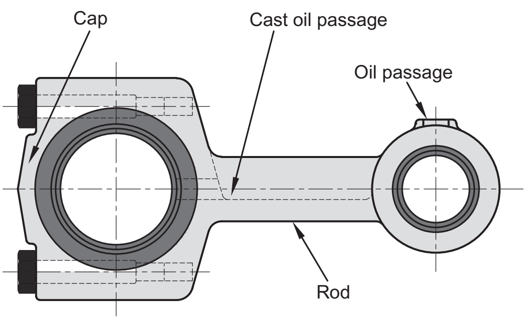


Fig. 4.33 Schematic diagram of a connecting rod.
Courtesy of Ingersoll Rand Company.

Fig. 4.34 Example of an eccentric strap.
Courtesy of Gardner Denver.

4.4.1.4 Wrist pin

The wrist pin is located in the crosshead and it transforms the oscillating motion of the connecting rod to reciprocating pumps. The maximum stress in the wrist pin from deflection is normally limited to 10,000 psi (69,000 kPa). The stress can be calculated from:

Field units

$$S = \frac{PL \times l}{8 \times (0.098)(d_w)^3} \tag{4.17a}$$

SI units

$$S = \frac{1275(PL \times l)}{(d_w)^3} \tag{4.17b}$$

where

S = stress, psi (kPa)
PL = plunger load, lb (kg)
l = length under load, in. (mm)
d_w = diameter of wrist pin, in. (mm)

Large pins are made hollow to reduce the oscillating mass and assembly weight. Depending on the design, pins can have a tight straight fit, a taper fit, or a loose fit in the crosshead. When needle or roller bearings are used for the wrist pin bearing, the wrist pin is used as the inner race.

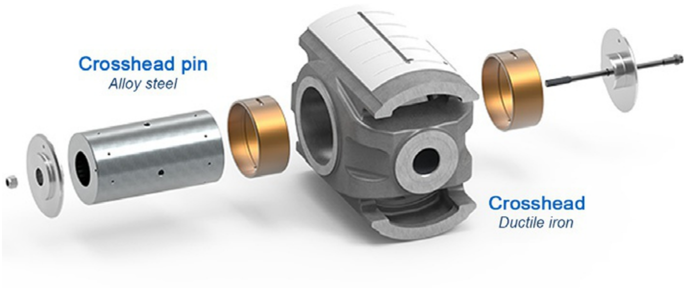


Fig. 4.35 Example of a crosshead.
Courtesy of Gardner Denver.

4.4.1.5 Crosshead

The crosshead (Fig. 4.35) moves in a reciprocating motion and transfers the plunger load to the wrist pin. It is designed to absorb the side, or radial, load from the plunger as the crosshead moves linearly on the crossway. The side load is ~25% of the plunger load. For cast iron crossheads the allowable bearing load is normally 80–125 psi (551–862 kPa). Crossheads are grooved for oil lubrication with a normal bearing surface of 63 rms. Crossheads are piston type (full round) or partial contact type. The piston type should be open end or vented to prevent air compression at the end of the stroke.

On vertical pumps the pull rods go through the crosshead so that the crosshead is under compression when the load is applied (Fig. 4.26).

4.4.1.6 Crossways

The crossway (Fig. 4.36) is the surface on which the crosshead reciprocates. On horizontal pumps it is cast integral with the frame (Fig. 4.27). On large frames it is usually replaceable and is shimmed to effect proper running clearance (Figs. 4.26 and 4.36). The crossway finish is normally 63 rms.

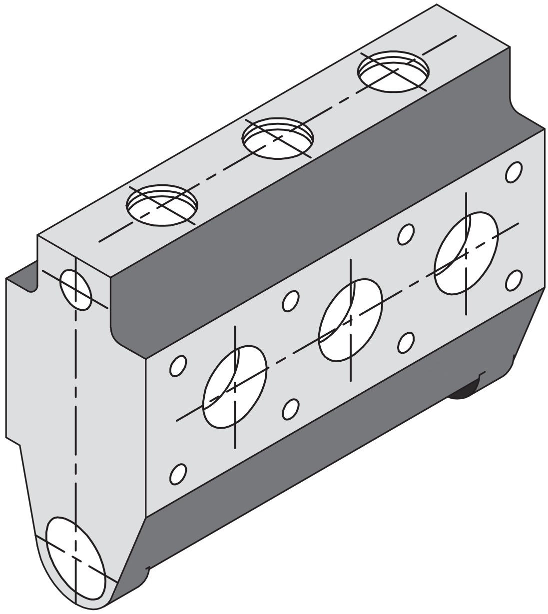
4.4.1.7 Pony rod (intermediate rod)

The pony rod is an extension of the crosshead on horizontal pumps (Fig. 4.27). It is screwed or bolted to the crosshead, and it extends through the "A" seal on the frame and against the pony rod preventing oil from leaking out of the frame. A baffle is fixed onto the rod to keep leakage from coming in contact with the frame (Fig. 4.27).

4.4.1.8 Pull rod (tie rod)

On vertical pumps two pull rods go through the crosshead. The rods are secured by a shoulder and nut so that the cast iron crosshead is in compression when the load is applied (Fig. 4.26). The rods extend out of the top of the frame and fasten to a yoke. The plunger is attached to the middle of the yoke with an aligning feature for the plunger (Fig. 4.29).

Fig. 4.36 Schematic diagram of a single cylinder.
Courtesy of Hydraulic Institute.



4.4.2 Fluid end components

The liquid end consists of the cylinder, plunger or piston, valves, stuffing box, manifolds, and cylinder head. Fig. 4.28 shows the liquid end of a horizontal pump, and Fig. 4.29 shows the same for a vertical pump.

4.4.2.1 Cylinder (working barrel)

The cylinder is the body in which the pressure is developed. It is continuously under stress. Cylinders on many horizontal pumps have the suction and discharge manifolds made integral with the cylinder. Vertical pumps usually have separate manifolds. A cylinder containing the passages for more than one plunger is referred to as a single cylinder (Fig. 4.36).

When the cylinder is used for only one plunger, it is called as individual cylinder (Fig. 4.37). Individual cylinders are used when developed stresses are high.

Cylinder allowable stresses range from 10,000 to 25,000 psi (69,000 to 172,000 kPa) depending on the material of the cylinder and the liquid being pumped. The allowable stress is a function of the fatigue stress of the material for the liquid pumped and the lifecycles required.

4.4.2.2 Plunger

The plunger transmits the force that develops the pressure. It is normally solid up to 5-in (127 mm) diameter. At large sizes, it may be made hollow to reduce weight. Small-diameter plungers used for 6000 psi (41,400 kPa) and higher should be

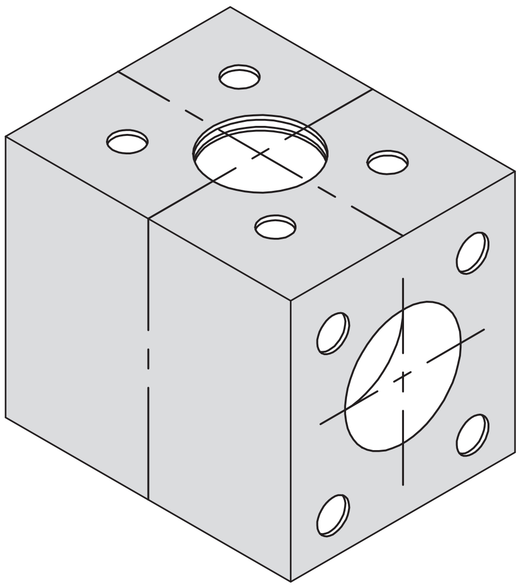


Fig. 4.37 Schematic diagram of an individual cylinder.
Courtesy of the Hydraulic Institute.

reviewed for possible buckling. Plunger speed ranges from 150 to 350 ft/min (46 to 107 m/s). The finish is normally 16 rms with a 30 to 58 Rockwell C hardness. Materials of construction are Colmonoy No. 6 on 1020, chrome plate on 1020, 440°C, 316, ceramic on 1020 with 93°C (200°F) limit, and solid ceramic. For <3000-psi (20,700 kPa) discharge pressure, ceramic is used for soft water, crude oil, mild acids, and mild alkalis. The problem with coatings on plungers is that at high pressures the liquid gets through the pores and underneath the coating. Pressure underneath the coating may cause it to flake off the plunger.

4.4.2.3 Pistons

Pistons are used in double-acting pumps and in some single-acting pumps for water pressures below 2000 psi (13,800 kPa); for higher pressures a plunger is usually used in single-acting pumps. Pistons are cast iron, bronze, or steel with reinforced elastomer faces (Fig. 4.38).

4.4.2.4 Cylinder liner

The cylinder wear liner (Figs. 4.38 and 4.39) is usually made of Ni-resist material. Its length is slightly longer than the stroke of the pump, allowing for an assembly entrance taper of the piston into the liner. On double-acting pumps, the liner has packing to prevent leakage from the high-pressure side to the low side of the cylinder. Because of the brittleness of the liner, the construction should be such that the liner is not compressed. The finish of the liner is normally 16 rms.

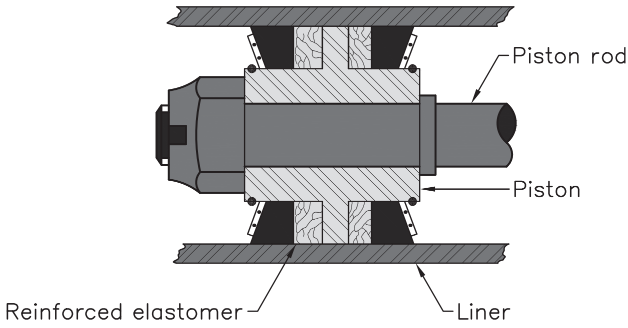


Fig. 4.38 Schematic diagram of a piston with an elastomer face.
Courtesy of FWI.

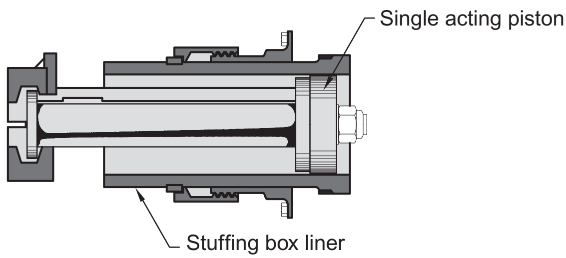


Fig. 4.39 Schematic diagram of a single-acting piston stuffing box.
Courtesy of Ingersoll Rand Company.

4.4.2.5 Manifolds

These are the chambers in which liquid is dispersed or collected for distribution before or after passing through the cylinder. On horizontal pumps, the suction and discharge manifold is usually made integral with the cylinder (Fig. 4.40).

Some horizontal pumps and some vertical pumps have only the discharge manifold integral with the cylinder (Fig. 4.41). On most vertical pumps, the suction and discharge manifolds are separate from the cylinder.

Suction manifolds are designed to eliminate air pockets from the flange to the valve entrance. Separate suction manifolds are cast iron or fabricated steel. Discharge manifolds are steel forgings or fabricated steel. The manifolds have a minimum deflection to prevent gasket shift when subjected to the plunger load.

4.4.2.6 Stuffing box

The function of the stuffing box is to seal the point where the plunger or piston enters the cylinder. The stuffing box of a pump consists of the box, lower and upper bushing, packing, and gland (Fig. 4.42). It should be removable for maintenance. The stuffing box bore is normally machined to a 63-rms finish to ensure packing sealing and life. A hoop stress calculation is used to determine the stuffing thickness, with an allowance of 10,000 to 20,000 psi (69,000 to 138,000 kPa).

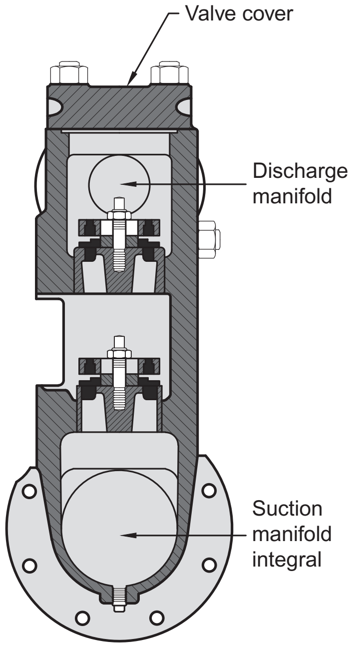


Fig. 4.40 Schematic diagram of an integral suction and discharge manifold.
Courtesy of Gardner Denver.

The bushings normally have a 63-rms finish with a clearance of ~0.001 to 0.002 in per in (0.001 to 0.002 mm per mm) of plunger diameter. The lower bushing is sometimes secured in an axial position to prevent the working of the packing. Bushings are made of bearing bronze, Ni-resist, or 316 stainless steel.

Stuffing box packing is either square cut (Figs. 4.43 and 4.44) or chevron (V-ring) (Fig. 4.45).

Some types use metal backup adapters. Unit packing consists of top and bottom adapters with a seal ring. A stuffing box can use three to five rings of packing or units, depending on the pressure and on the fluid being pumped. Packing or seal rings are usually made of reinforced asbestos, teflon, or neoprene.

Packing can be made self-adjusting by installing a spring between the bottom of the packing and the lower bushing or bottom of the cylinder. This arrangement eliminates overtightening and allows for uniform break-in of the packing. The packing is lubricated by injecting grease through a fitting, by gravity oil feed, or by an auxiliary lubricator driven through a takeoff on the crankshaft.

For chemical or slurry service a lower injection ring is used for flushing. This procedure prevents concentrated pumped fluid from impinging directly on the packing. The injection can be a continuous flush or can be synchronized to inject only on the suction stroke. Flush glands are employed where toxic vapor or flashing appears after the packing.

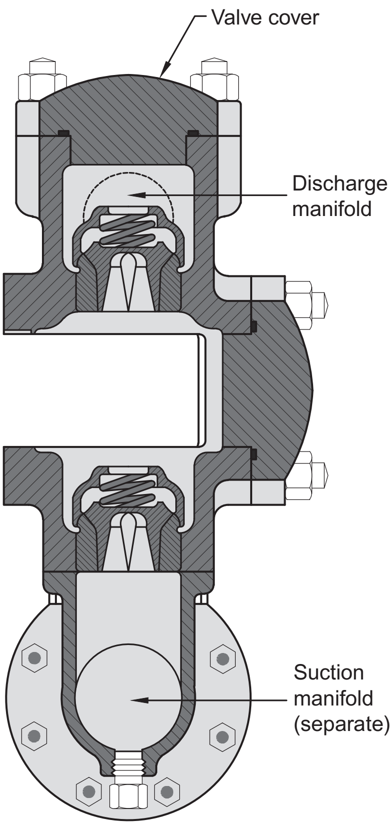


Fig. 4.41 Schematic diagram of a separate suction with integral discharge manifold.

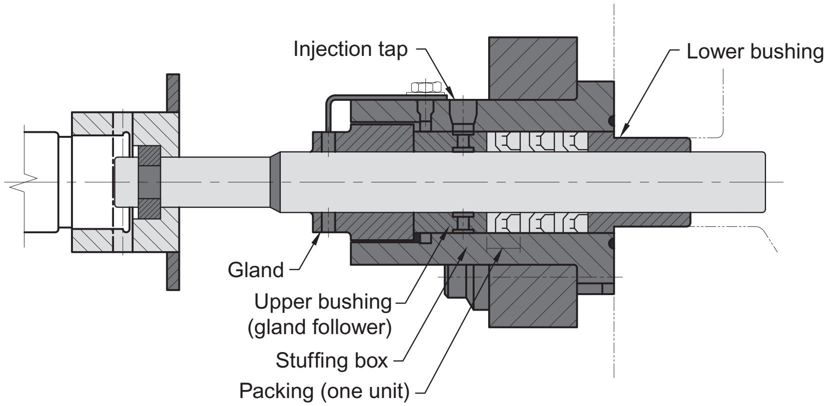


Fig. 4.42 Schematic diagram of a stuffing box.
Courtesy of Ingersoll Rand Company.

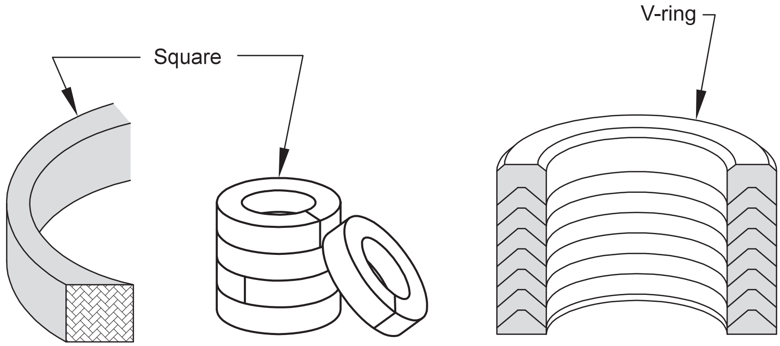


Fig. 4.43 Schematic diagram of square cut and chevron (V-ring) packing. Courtesy of Hydraulic Institute.

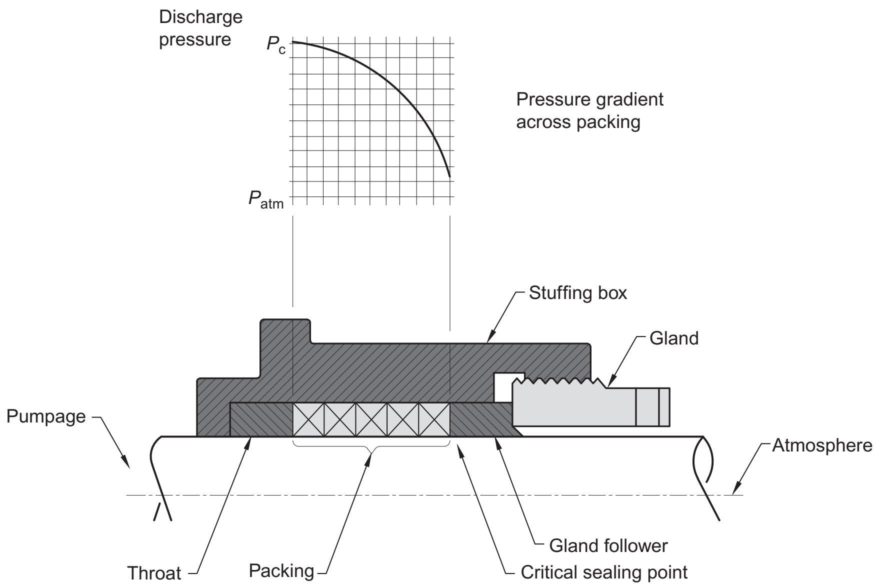


Fig. 4.44 Schematic diagram of the square cut packing arrangement. Courtesy of Hydraulic Institute.

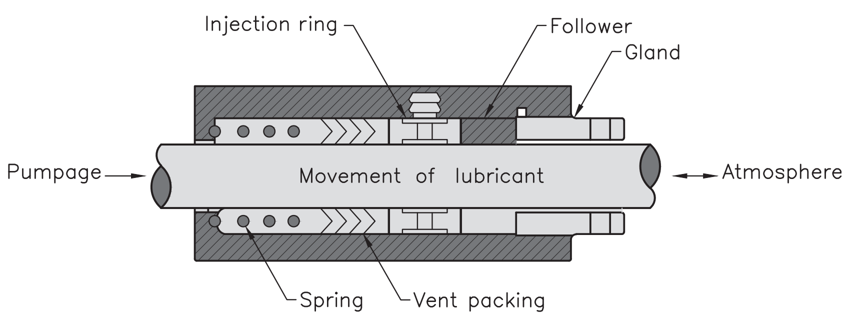


Fig. 4.45 Schematic diagram of the Chevron packing arrangement.

The stuffing box of a double-acting piston does not require an upper and lower bushing because the piston is guided by the cylinder liner. Single-acting pistons do not employ a stuffing box. Leakage of the piston goes into the frame extension to mix with the stuffing box's continuous circulating lubricant.

The advantages and limitations of square cut packing and chevron (V-ring) packing is as follows:

4.4.2.6.1 Square cut packing

The advantages of square cut packing are as follows:

- low cost
- smaller overall space required
- easy to repair

Limitations include

- requires continuous adjustment gland
- not good when solids are present in the fluid pumped unless an injection ring is installed

4.4.2.6.2 Chevron packing

Advantages of Chevron packing are as follows:

- Gland does not have to be adjusted as packing wears
- Spring provides a cavity for injection of clean seal fluids
- Less leakage than square cut

Limitations include

- Cost
- Spring cavity can reduce volumetric efficiency for high compressibility fluids as it is in communication with cylinder
- Vapors can accumulate in spring cavity

4.4.2.7 Valves

The valve assembly can be classified according to the manner in which it is installed in the pump (Fig. 4.46). In a caged assembly, the entire valve assembly fits into a straight bore machined into the pump. The seal between the fluid end and the valve assembly is maintained by a gasket. This is the most common design.

A tapered seat valve assembly fits into a taper machined into the fluid end. The fit between the fluid end and assembly is forced. This design is the original design offered by pump manufacturers and is best used for large flow rates. The various types of valves are shown in Fig. 4.47.

Plate, wing, and plug valves are normally limited to clean fluids because of the flow patterns through the open valve. Ball valves are normally applied in high pressure or abrasive service since the line contact at seating surfaces is better suited for abrasive service. The ball is free to rotate, eliminating side load. Plate and plug valves have

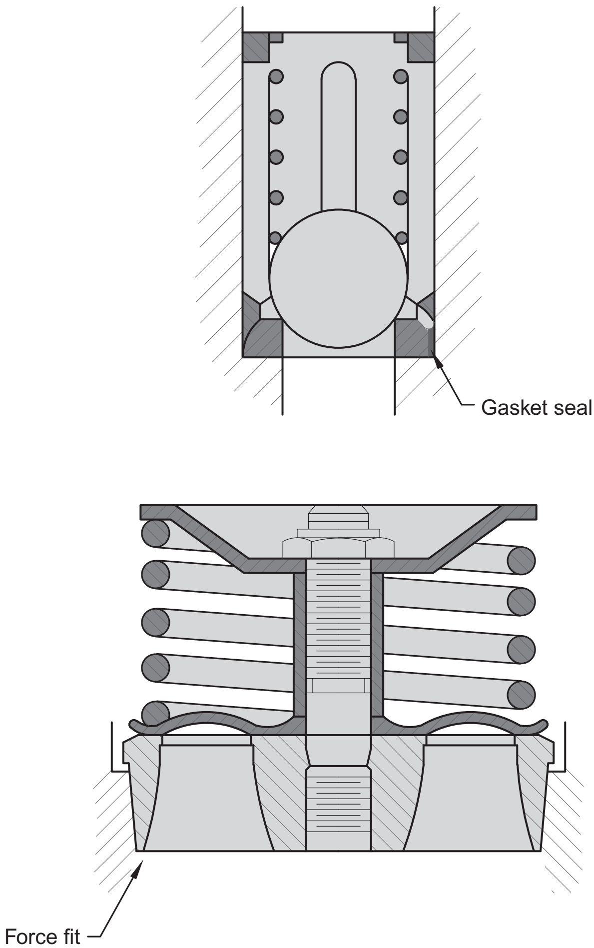


Fig. 4.46 Schematic diagram of the valve assembly classification.

smaller pressure losses than balls, but their flat valve surfaces are not suitable for abrasive surfaces. Wing valves are generally considered to be the best currently available pump valve. Pressure drop through the valves is low. Guided motion prevents flutter. Slurry valves are used in low-pressure service.

Some pumps use the same size suction and discharge valves for interchangeability. Some use larger suction valves than discharge valves for $NPSH_R$ reasons. Because of space considerations, valves are sometimes used in clusters on each side of the plunger to obtain the required total valve area. Table 4.9 presents seat and plate hardness for some valve materials.

Seats and plates made of 316 stainless steel are chrome plated or Colmonoy No. 6 plated to give them surface hardness. Seats and plates normally have a 32-rms finish.

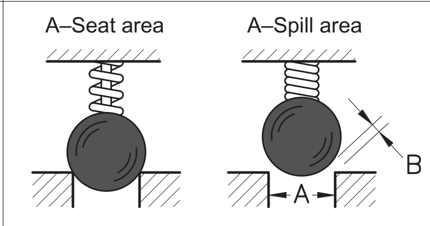
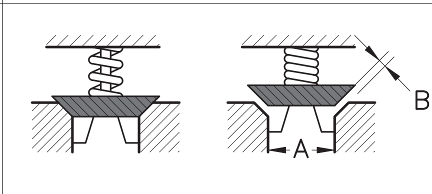
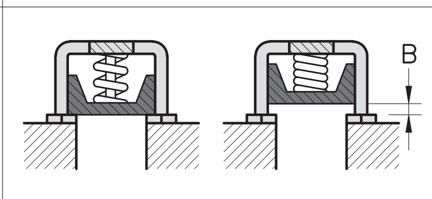
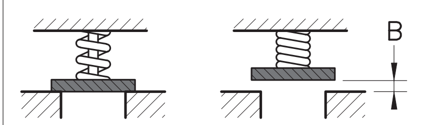
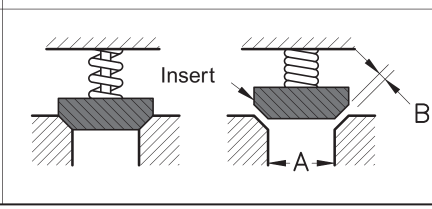
Type	Sketch	Pressure psi (bar)
Ball		30,000 (2069)
Wing		10,000 (690)
Plug		6000 (414)
Plate		5000 (345)
Slurry		2500 (172)

Fig. 4.47 Various types of valves and their pressure applications. Courtesy of Ingersoll Rand Company.

The number and size of valves are chosen so that the velocity through any valve, sometimes referred to as the "spill velocity," is controlled. Some representative velocities are given in Table 4.10. The spill velocity is given by

Field units

$$V = \frac{(\text{gpm through valve})(0.642)}{\text{Spill area of the valve, in}^2} \tag{4.18a}$$

Table 4.9 Recommended material hardness for valve plate and seat

Material	Plate	Seat
Rockwell C hardness		
329	30 to 35	38 to 43
440	44 to 48	52 to 56
17-4 PH	35 to 40	40 to 45
15-5 PH	35 to 40	40 to 45
Brinell hardness		
316	150 to 180	150 to 180

Table 4.10 Typical medium valve spill velocities

	Spill velocity	
Valve	**m/s**	**(ft/s)**
Clean liquid suction valve	0.9–2.4	(3–8)
Clean liquid discharge valve	1.8–6	(6–20)
Slurry suction and discharge valve	1.8–3.7	(6–12)

SI units

$$V = \frac{(\mathrm{m}^3/\mathrm{h\ through\ valve})(556.0)}{\mathrm{spill\ area\ of\ the\ valve,\ mm}^2} \tag{4.18b}$$

where:

V = velocity, ft/s (m/s)

4.4.2.8 Bearings

Bearings are classified as either hydrodynamically lubricated (oil film sleeve type) or rolling contact (antifriction). With the hydrodynamically lubricated bearing the primary motion is a sliding of one surface over another. The surfaces of this bearing are separated by a relatively thin layer of lubricant, which prevents any metallic contact, except when starting or stopping under load. With the roller type, the primary motion is a rolling of one surface over another. Balls or rollers are used to support the applied load by actual metal-to-metal contact over a relatively small area.

The advantages of hydrodynamic lubricated bearings are as follows:

- The bearing life is not a function of speed or load. Very little wear occurs.
- Oil forces tend to hold the shaft in the center of the bearing, providing some degree of dampening.
- Journal bearings and flat plate thrust bearings have very low relative cost.

The limitations of hydrodynamic lubricated bearings are as follows:

- They require a continuous supply of cool, uncontaminated oil, at a constant fixed pressure.
- Journal bearings can tolerate no axial load, while flat plate and Kingsbury bearings can tolerate no radial load.
- They have a relatively high friction drag as opposed to rolling contact bearings and thus have higher horsepower losses.

The advantages of rolling contact bearings are as follows:

- A continuous lubrication system is generally not required. Occasional greasing of the bearing is all that is required.
- They offer very little resistance to movement; therefore power losses are smaller than with hydrodynamic lubricated bearings.
- They can be designed for both thrust and radial loads simultaneously.

The limitations of rolling contact bearings are as follows:

- They have a specific life for the design load.
- They have limited load/speed capability.
- Both hydrodynamic and antifriction rolling contact bearings are used in pumps. Some frames use all hydrodynamic, other use all rolling contact, and others use a combination of both.

4.4.2.8.1 Hydrodynamic bearings

When properly installed and lubricated, hydrodynamic bearings are considered to have infinite life. They are designed to operate within a certain speed range, and too high or too low a speed will upset the film lubrication. They cannot be operated satisfactorily below 40 rpm with the plunger fully loaded, using standard lubrication. Below this speed, film lubrication is inadequate. The finish on the hydrodynamic bearing is normally 16 rms. Clearances are ~0.001 in per in (0.001 mm per mm) of diameter of the bearing.

Wrist pin bearings have only oscillating motion. On single-acting pumps with low suction pressure, there is adequate reverse loading on the bearing to permit replenishment of the oil film. High suction pressure on horizontal pumps increases the reverse loading. On vertical pumps, high suction pressure can produce a condition of no reversal loading, and in this case higher oil pressure is required. Allowable projected area loading is 1200 to 1500 psi (8300 to 10,300 kPa) with bronze bearings.

The crank pin bearing is a rotating split bearing that has a better oil film than the wrist pin bearing. It is clamped between the connecting rod and cap. The bearing is bronze-backed Babbitt metal bearing. Allowable projected area loading is 1200 to 1600 psi (8300 to 11,000 kPa).

The main bearings absorb the plunger load and gear load. The total plunger varies during the revolution of the crankshaft. The triplex main bearings receive the greatest variations in loading because the crankshaft has the greatest relative span between bearings. Sleeve bearings are flanged to lock them in an axial position, and the flange absorbs residual axial thrust. On large-stroke vertical pumps, there is a main bearing between every connecting rod. Split bearings are bronze-backed Babbitt metal with an allowable projected area load of 750 psi (5200 kPa).

4.4.2.8.2 Rolling contact bearings

A pump with rolling contact bearings can be started under full plunger load without a bypass line. Rolling contact bearings allow the pump to operate continuously at low speeds with full plunge load. They are selected for a designated bearing life.

Wrist pin bearings are needle or roller bearings. The outer race is a tight fit into the strap, and the wrist pin is used as the inner race. This reduces the size of the bearing and strap. The wrist pin is held in the crosshead with a taper or keeper plate (Fig. 4.48).

Crank pin bearings are roller bearings. The outer race is mounted separately from the inner race; the inner race is mounted on the crankshaft and secured axially by a shoulder on the shaft and by a keeper plate, or by keeper plates on both sides of the race. The outer race is assembled in the strap in the same way as the inner race mounting. The strap is then slipped over the shaft and assembled to the inner race.

The main bearings used in conjunction with the hydrodynamic wrist bearing and the hydrodynamic crank pin bearings are usually tapered roller bearings, as in the case of a horizontal triplex plunger pump. The main bearings are ordinarily mounted directly into the frame (Fig. 4.49).

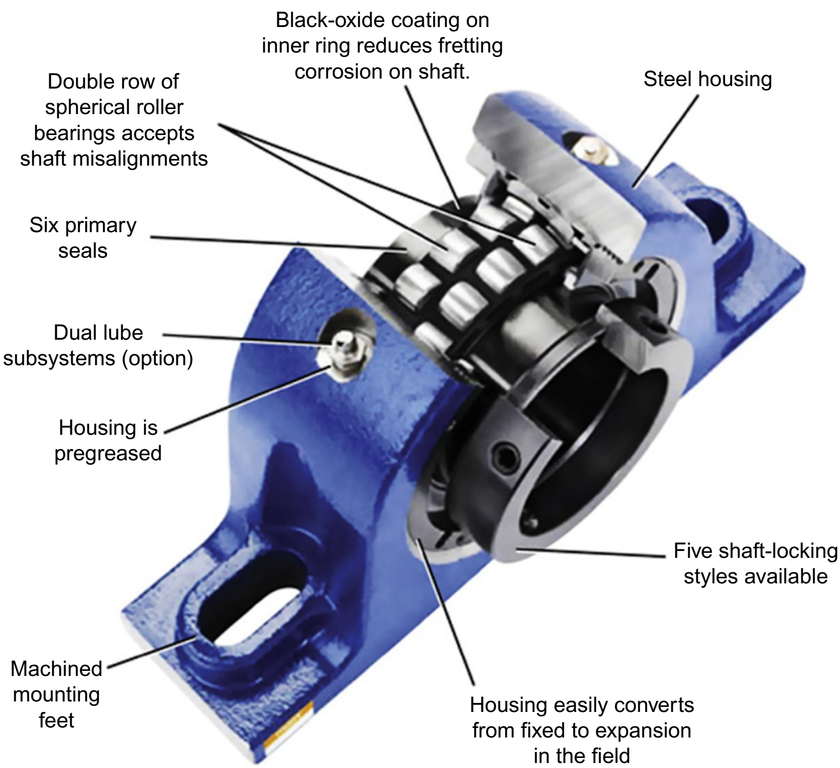


Fig. 4.48 Schematic diagram of a main bearing mounted in a bearing holder.

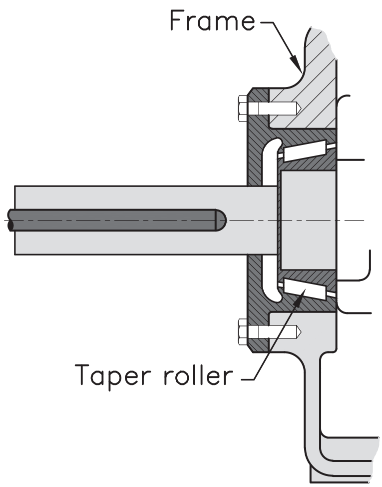


Fig. 4.49 Schematic diagram showing the main bearing mounted directly in the frame. Courtesy of Ingersoll Rand Company.

The main bearings used with full rolling contact bearing design are self-aligning spherical roller bearings. These bearings compensate for axial and radial movement of the crankshaft and are usually mounted in bearing holders, which in turn are mounted on the frame (Fig. 4.48). This type of design is used on mud and slurry pumps.

4.4.2.9 Lubrication system

The power end of large slow-speed positive displacement pumps operating at speeds below 160 to 165 rpm is normally furnished with pressurized (force-feed) lubrication systems, while smaller positive displacement pumps operating at speeds above 165 rpm are normally furnished with splash-type lubrication systems. Splash-type lubrication systems use checks on the crankshaft or oil scoops to throw oil by centrifugal force against the frame wall. This oil is then distributed by gravity to the crosshead, wrist pin, and crank pin. Pressurized or force-feed lubrication systems may use either a built-in direct drive lube oil pump or an external direct drive or motor-driven lube pump to lubricate power end components. Force-feed lubrication systems give better lubrication but are more expensive to purchase.

4.5 Reciprocating pump standards

4.5.1 Overview

Most reciprocating pumps used in oil field service are built to manufacturer's standards, although they have many features required by *API Standard 674 (Positive Displacement Pumps Reciprocating)*. The basic requirements of *API*

Standard 674 are summarized as follows to provide some guidance in making selections of various options:

- *Pressure ratings*: All pressure and temperature ratings encountered in oil field production operations.
- *Pump speed*: Speeds for single-acting plunger pumps are limited to the values presented in Table 4.11.
- *Cylinder design*: Requires ASME code Section VIII Division 1 for cylinders. Forged cylinders required for applications above 3000 psi (20,700 kPa).
- *Cylinder liner*: Required.
- *Plunger or rods*: Requires that all piston rods or plungers in contact with packing be hardened or coated to Rockwell C35.
- *Stuffing boxes*: Required to accept at least three packing rings.
- *Bearings*: Ball bearings must be capable of three years continuous operation at rated pumping conditions.
- *Materials*: Cast iron or ductile iron is not allowed for pressure-retaining parts handling flammable or toxic fluids.
- *Testing*: Pressure-retaining parts (including auxiliaries) are tested hydrostatically with liquid at a minimum of 1.1 times the maximum allowable working pressure.

4.5.2 Water injection pumps—NACE RP-04-75

The National Association of Corrosion Engineers has published their "Standard Recommended Practice for the selection of Metallic Materials to be used in All Phases of Water Handling for Injection into Oil Bearing Formations." This standard provides guidelines for selecting materials used in water handling equipment of all types, with one section devoted to pumps. This is often a useful reference when specifying pumps or reviewing manufacturer proposals for water handling and injection services.

4.5.3 General oilfield standards

Table 4.12 presents a comparison of some of the major requirements of API STD 674 compared to manufacturer's standards generally used in oil field service. It can be seen that the API standard is more stringent in design requirements and quality control and

Table 4.11 API Standard 674—Maximum allowable speed

Stroke length		RPM	Max allowable speed	
mm	**(in.)**	**rpm**	**m/s**	**(ft/min)**
50.8	(2)	450	0.76	(150)
76.2	(3)	400	1.02	(200)
101.6	(4)	350	1.18	(233)
127.0	(5)	310	1.31	(258)
152.4	(6)	270	1.37	(270)
177.8	(7)	240	1.42	(280)
203.2	(8)	210	1.42	(280)

Table 4.12 Comparison of API Standard 674 and normal oilfield specifications

	API Standard 674	Normal oilfield specifications
Pump casings rating	110% of the design discharge pressure	120% of the design discharge pressure
Pump speed	Speeds for single-acting plunger pumps are limited to the values presented in Table 4.11	85% of manufacturer's maximum continuous rpm rating
Cylinder design	Requires ASME Code Section VIII, Division 1 for cylinders. Forged cylinders required for applications above 3000 psi (20,700 kPa)	No requirements
Plunger or rods	Requires that all piston rods or plungers in contact with packing be hardened or coated to Rockwell C35	Hardened in the packing area
Stuffing boxes	Required to accept at least three packing rings	No requirements
Bearings	Ball bearings must be capable of three years continuous operation at rated pumping conditions	No requirements
Materials	Cast iron or ductile iron is not allowed for pressure-retaining parts handling flammable or toxic fluids	Forged steel fluid end
Testing	Pressure-retaining parts (including auxiliaries) are tested hydrostatically at a minimum of 11/2 times the maximum allowable working pressure	No requirements

is thus normally used only for critical services where reliability is important. Pumps for normal oil field applications are less expensive and much more readily available. Whenever service conditions allow considerable time, cost savings are possible by specifying manufacturer's standards.

4.5.4 Data sheets

API has a standard 4-page data sheet for pumps. An example copy may be found in API Standard 674.

4.6 Materials of construction

The choice of materials used for pump parts varies with operating conditions and liquids handled. Tables 4.13–4.15 from *API Standard 674* list combinations of metals most commonly used in pumps. Table 4.16 from *API Standard 610* may be used as a guide in selecting material classes for pumps. Table 4.17 and Table 4.18 from *NACE RP-0475* present pump materials for water service.

Table 4.13 Materials and material specifications for major component parts

Materials for reciprocating pump liquid end parts[a] **Material class and material class abbreviations**[b]				
API material class code Part	I-1 CI/CI	I-2 CI/BRZ	S-1 STL/CI	S-2 STL/HI ALLOY
Pressure-retaining parts				
Cylinder	Cast iron	Cast iron	Carbon steel	Carbon steel
Cylinder head, valve cover	Cast iron or carbon steel	Cast iron or carbon steel	Carbon steel	Carbon steel
Stuffing box	Cast iron or carbon steel	Cast iron or carbon steel	Carbon steel	Carbon steel
Gland	Cast iron or carbon steel	Cast iron or carbon steel	Carbon steel	Carbon steel
Bolting	Carbon steel	Carbon steel	AISI 4140 steel	AISI 4140 steel
Nonpressure-retaining parts				
Lantern ring, throat bushing, packing follower	Cast iron, 12% chrome, or any stainless steel	Cast iron or bronze	Cast iron, 12% chrome, or any stainless steel	12% chrome, or any stainless steel
Valve and seat	Stainless steel	Bronze	Carbon steel or any stainless steel	12% chrome, or any stainless steel
Valve spring	Stainless steel	Bronze or stainless steel	Stainless steel	Inconel or stainless steel
Piston[c]	Cast iron	Cast iron	Cast iron	Ni-resist
Piston rod[c]	12% chrome	12% chrome	12% chrome	12% chrome
Piston ring[c]	Cast iron or nonmetallic	Nonmetallic	Cast iron or nonmetallic	Special cast iron or Ni-resist
Piston ring expander[c]	Cast iron or stainless steel	Stainless steel	Cast iron or stainless steel	Stainless steel
Plunger (power pumps)[c]	12% chrome, hard surfaced	12% chrome, hard surfaced	12% chrome, hard surfaced	12% chrome, hard surfaced
Plunger (direct-acting pumps)[c]	12% chrome, hardened	12% chrome, hardened	12% chrome, hardened	12% chrome, hardened
Cylinder liner	Cast iron	Bronze	Cast iron	Ni-resist or 12% chrome

[a]This table is to be used as a guide.

[b]The abbreviation above the diagonal line indicated cylinder material: the abbreviation below the diagonal line indicates fitting material. Abbreviations are as follows: *CI*, cast iron; *BRZ*, bronze; *STL*, steel; *HI ALLOY*, high-temperature alloy.

[c]Applies where applicable.

Table 4.14 Materials and material specifications for major component parts

Materials for direct-acting reciprocating pump gas end parts[a] **Material class and material class abbreviations**[b]				
API material class code Part	I-1 CI/CI	I-2 CI/BRZ	S-1 STL/CI	S-2 STL/HI ALLOY
Pressure-retaining parts				
Cylinder	Cast iron	High-temperature cast iron	Carbon steel	Carbon steel
Head, chest, cover	Cast iron or steel	High-temperature cast iron or steel	Carbon steel	Carbon steel
Stuffing box	Cast iron or steel	High-temperature cast iron or steel	Carbon steel	Carbon steel
Gland	Cast iron or steel	High-temperature cast iron or steel	Carbon steel	Carbon steel
Head, chest, box and gland bolting	AISI 4140 steel	AISI 4140 steel	AISI 4140 steel	AISI 4140 steel
Nonpressure-retaining parts				
Cylinder liner	None	None	Cast iron	High-temperature cast iron
Box bushing[c]	Cast iron	Cast iron	Cast iron	Cast iron
Piston	Cast iron	Cast iron	Cast iron	High-temperature cast iron
Piston rod	12% chrome	12% chrome, hardened	12% chrome	12% chrome, hardened
Piston ring	Cast iron	Cast iron	Cast iron	Special cast iron
Main valve	Cast iron	Cast iron	Cast iron	High-temperature cast iron
Valve rings[c]	Cast iron	Cast iron	Cast iron	Special cast iron
Auxiliary valve[c]	Cast iron	Cast iron	Cast iron	High-temperature cast iron
Valve rod	12% chrome	12% chrome, hardened	12% chrome	12% chrome, hardened

[a]This table is to be used as a guide.
[b]The abbreviation above the diagonal line indicates cylinder material: the abbreviation below the diagonal line indicates fitting material. Abbreviations are as follows: *CI*, cast iron; *BRZ*, bronze; *STL*, steel; *HI ALLOY*, high-temperature alloy.
[c]Applies where applicable.

Table 4.15 Materials for reciprocating pump parts[a]

Material	Castings	Forgings	Bar stock	Bolts and studs
Cast iron	ASTM A 48 or A 278	–	–	–
Nodular iron	ASTM A 395 or A 536	–	–	–
High-temperature cast iron	ASTM A 278, Class 40 or 60, stress relief, annealed	–	–	–
Carbon steel	ASTM A 216, Grade WCA or WCB, or ASTM A 352	ASTM A 105 or ASTM A 576	ASTM A 108 or ASTM A 575	–
5% chrome steel	ASTM A 217, Grade C5	ASTM A 182, Grade F5	–	–
12% chrome steel	ASTM A 296, Grade CA6NM, or CA15	ASTM A 182, Grade F6	ASTM A 276, Type 410, or ASTM A 582, Type 416	ASTM A 193, Grade B6
18-8 stainless steel	ASTM A 296, Grade CF20	ASTM A 182, Grade F304	ASTM A 276, Type 304	ASTM A 193, Grade B8
316 stainless steel	ASTM A 296, Grade CF8M	ASTM A 182, Grade F316	ASTM A 276, Type 316	ASTM A 193, Grade B8M
AISI 4140 steel	–	–	ASTM A 322, Type 4140	ASTM A 193, Grade B7
Bronze	ASTM B 584	–	ASTM B 139	ASTM B 5124
Material	Typical description			
Ni-resist	Type 1, 2, or 3 as recommended by International Nickel Co. for service conditions			
Hardenable stainless steel	ARMCO 17.7 PH, ARMCO 17.4 PH, US Steel Stainless W, Allegheny Ludlum AM 350 and AM 355, nitronic 50 or 60			
Colmonoy	Sprayed or fused deposit of 5.08 mm (0.020 in) or gas weld deposit of 7.94 mm (1/32 in) minimum thickness of Wall-Colmonoy AWS, Class R, Ni-Cr-C Metso 16c			

[a]This table is to be used as a guide.

Table 4.16 Material classes for centrifugal pumps

Caution: This table is intended as a general guide. It should not be used without a knowledgeable review of the specific services involved.

Service	On-plot process plant	Off-plot transfer and loading	Temperature range (F)	Pressure range (psig)	Material class	Note reference number
Fresh water, condensate, cooling tower water	X	X	<212	All	1-1 or 1-2	
Boiling water and process water	X	X	<250	All	1-1 or 1-2	7
	X	X	250–350	All	S-5	7
	X	X	>350	All	D-6	7
Boiler feed water						
Cast horizontally split	X	X	>200	All	C-6	
Forged barrel	X	X	>200	All	S-6	
Boiler circulator	X	X	>200	All	C-6	
Foul water, reflux drum water, water draw, and hydrocarbons containing these waters, including reflux streams	X	X	<350	All	S-3	2
Propane, butane, liquefied petroleum gas, and ammonia (NH3); diesel oil; gasoline; naphtha; kerosene, gas oils; light, medium, and heavy lube oils; fuel oil residuum; crude oil; asphalt; synthetic crude buttons	X	X	<450	All	S-1	
	X	X	<450	All	S-1	
	X	X	450–700	All	S-6	2.5
	X	X	>700	All	C-6	2
Noncorrosive hydrocarbons, for example, catalytic reformate, desulfurized oils	X	X	450–700	All	S-4	5
Xylene, toluene, acetone, benzene, furfural, MEK	X	X	<450	All	S-1	

Sodium carbonate, doctor solution	X	X	<350	All	1-Jan	
Caustic (sodium hydroxide) concentration of 20% or less	X	X	<140	All	S-1	8
	X		140–200	All	S-3	8
	X		>200	All		6
MEA, DEA, TEA-stock solutions	X	X	<250	All	S-1	
DEA, TEA-lean solutions	X	X	<250	All	S-1	
MEA-lean solution (CO_2 only)	X	X	175–300	All	S-9	
MEA-lean solution (CO_2 and H_2S)	X	X	175–300	All		9
MEA, DEA, TEA-rich solutions	X	X	<175	All	S-1	
Sulfuric acid concentration						
85%	X	X	<100	All	S-1	2
85%–15%	X	X	<100	All	A-8	2
15%–1%	X	X	<100	All	A-8	2
1%	X	X	<450	All	A-8	2
Hydrofluoric acid concentration of over 96%	X	X	<100	All	S-9	2

Notes:
The materials for pump parts for each material class are given in API 618.
The corrosiveness of foul waters, hydrocarbons over 450°F, acids, and acid sludges may vary widely. A materials recommendation should be obtained for each service. The material class indicated before will be satisfactory for many of these services but must be verified.
Cast iron cases, where recommended for chemical services, are for nonhazardous locations only. Use steel cases (S-1 or 1-1) for pumps in services located near process plants or in any location where released vapor from a failure could create a hazardous situation or where pumps could be subjected to hydraulic shock, for example, in loading service.
Obtain separate materials recommendations for services not clearly identified by the service descriptions listed in this table.
If product corrosivity is low, Class S-4 materials may be used for services at 451 to 700°F. Obtain a separate materials recommendation in each instance.
Use Alloy 20 or Monel pump material and double mechanical seals with a pressurized seal oil system.
Consider oxygen content and buffering of water in the selection of material.
All welds should be stress relieved.
Use Class A-7 materials except for a carbon steel case.
Mechanical seal materials for streams containing chlorides, all springs, and other metal parts should be Alloy 20 or better. Buna-N and neoprene should not be used in any service containing aromatics. Viton should not be used in services containing aromatics above 200°F.

Table 4.17 Injection pumps (plunger)

	Environment			
	Aerated		Nonaerated	
	Without H2S	**With H2S**	**Without H2S**	**With H2S**
Fluid end	Ti 400M, K500, AlBr6ca,w, A205, In6255	Ti NiAl6w, 3167,8, 316L7,8, A205, In6255, K5005	Ti 400M, K500, AlBr6ca,w, A205, In6255	Ti AlBr6, NiAlw, 400M5, K5005, A205, In6255
Plunger	NM6 HardNi1,6	NM6 HardNi1,6	NM6 HardNi1,6	NM6 HardNi1,6
Valve	NM, Ti, HardCO_2, WC_2 NiAlca, 316, 17-4, 15-7MO	NM, Ti, HardCO_2, WC_2 NiAlca, 316, 17-4, 15-7MO	NM, Ti, HardCO_2, WC_2 NiAlca, 316, 17-4, 15-7MO	NM, Ti, HardCO_2, WC_2 NiAlca, 316, 17-48, 15-7MO
Valve seat	400M, 17-4, NiAlca, 3163,7	400M, 17-4, NiAlca, 3163,7	400M, 17-4, NiAlca, 3163,7	400M, 17-48, NiAlca, 3163,7
Valve spring	Ti, In600, In750, 400 M	Ti, In600, In750, 400 M	Ti, In600, In750, 400 M	Ti, In600, In750, 400 M
Stuffing box	AlBrca, 3164	AlBrca, 3164	AlBrca, 3164	AlBrca, 3164

Notes

1. For many services can be on carbon steel base. For extreme corrosion, should be austenitic stainless steel.
2. Ball valve.
3. For ball valves with hard insert.
4. Other materials can be furnished to match fluid ends.
5. Limited experience—unrated.
6. Widely used materials.
7. Subject to chloride cracking and subject to differential aeration.
8. Subject to stress cracking under some conditions.

Table 4.18 Material code for use with Table 4.17

Code	Description of material (ASTM designations unless otherwise noted)	Code	Description of material (ASTM designations unless otherwise noted)
Fe1	Carbon or low-alloy steel	AlBrw	Aluminum bronze, B150, alloy 614 (9C sic) (wrought)
Fec1	Carbon or low-alloy steel, coated or lined internally	NiAlca	Nickel aluminum bronze, B148, alloy 955 (9D) (cast)
Fe2c1	Carbon or low-alloy steel, coated in or out		
Cl1	Gray cast iron, A48, Class 30	NiAlw	Nickel aluminum bronze, B150, alloy 630 (9D sic) (wrought)
Clc1	Cast iron, coated internally	NiVBr	Ni-Vee3 bronze, Type D

Table 4.18 Continued

Code	Description of material (ASTM designations unless otherwise noted)	Code	Description of material (ASTM designations unless otherwise noted)
Cl2c1	Cast iron, coated in and out	A20	Alloy/20 (wrought) or A296, Grade CN-7M (cast)
D11	Ductile (nodular) iron, A536, various grades		
NiRei	Ni-resist iron, A436, Type 1	400M	Monel3 400 (wrought) or Monel3 410 (cast)
NiRei 2A	Ni-resist iron, A436, Type 2A	K500	Monel3 K-500 (wrought) or Monel3 505 (cast)
NiRe D2	Ductile Ni-resist3 iron, A439, Type D-2	Inall	Inconel3 metal alloy (all listed grades)
410	AISI 410 stainless steel (wrought) or A296, Grade CA-15 stainless steel (cast)	In600	Inconel3 metal alloy 600
303	AISI 303 stainless steel (wrought)	In625	Inconel3 metal alloy 625
304	AISI 304 stainless steel (wrought) annealed or A296, Grade CF-8 (cast) annealed	In718	Inconel3 metal alloy 718
		In750	Inconel3 metal alloy X-750
316	AISI 316 stainless steel (wrought) annealed or A296, Grade CF-8M (cast) annealed	C1	Carbon or graphite
		Ti1	Titanium or titanium alloys
316L	AISI 316L stainless steel (wrought) annealed or A296, Grade CF-3M (cast) annealed	WC1	Tungsten carbide
		HaC	Hastelloy4 "C," B334 and B336 (wrought) or A494 (cast)
17-4	17-4 PH stainless steel (wrought) or ACI Grade CB-7Cu (cast)	Hard Ni1	Ni-Cr-B-Si alloys (such as Colmonoy) 2,5
15-7Mo	15-7 Mo PH stainless steel	Hard Co1	Cobalt-base alloys (such as stellite) 2,4
BrgBr	High-leaded tin bronze, B144, alloy 943 (70-5-25)	Cr/1	Chromium plating
63 Br	Zinc-less bronze, SAE 63 (Copper Alloy 927)	Ni/1	Nickel plating
AlBrca	Aluminum bronze, B148, alloy 954 (9c) salt water anneal (cast)	NM1	Nonmetallic

Notes

1. Material group having various analysis—not listed in Table 4.17.
2. Various forms—sprayed and fused coatings, weld overlays, cast, wrought.
3. Reg. T.M. The International Nickel Company.
4. Reg. T.M. Cabot Corporation.
5. Reg. T.M. Wall-Colmonoy Corporation.

4.7 Installation considerations

4.7.1 Overview

If positive displacement pumps are properly installed and operated, satisfactory performance can be realized for a long time. These pumps are manufactured in a variety of designs for many different services. Each manufacturer's instructions should be carefully followed for specific machine or application requirement. The following discussion relates to general installation guidelines for positive displacement reciprocating pumps.

4.7.2 Foundations and alignment

4.7.2.1 Background

Most pump foundations are constructed of reinforced concrete. Pump and driver are bolted to a cast iron or steel baseplate that is secured to the concrete foundation via anchor bolts. Small pumps need a foundation large enough to accommodate the baseplate assembly. On the other hand, large pumps require a foundation that is 3 to 4 times the weight of the pump and driver.

4.7.2.2 Anchor bolt-sleeve installation

Each bolt is fitted with a washer and passed through a pipe sleeve that has a diameter 3 to 4 times greater than the bolt. The bolt-sleeve unit is set into the concrete at the predetermined base hole positions. The flexibility in the sleeve-washer unit allows minor adjustments to be made in the bolt position prior to final tightening, even after the concrete foundation has set.

4.7.2.3 Metal shim adjustments

Metal shims are used to position the pump on the foundation. Adjustments are made until the pump shaft and port flanges are completely leveled. Alignment between the pump and driver is then adjusted before connecting the pump to the suction and discharge lines. The latter should have previously been aligned during the initial positioning of the baseplate.

4.7.2.4 Grouting

Once the piping has been securely bolted, the entire pump assembly should be rechecked for flexure due to pipe strain. If the drive train alignment has not been changed by bolting the piping, the space between the baseplate and concrete foundations is filled with grouting. Grouting should be sufficiently fluid so as to fill all the available space under the baseplate.

4.7.2.5 Operating temperature considerations

It is essential the alignment between the piping, pump, and driver do not change. Ideally, alignments should be made at the operating temperature of the pumping system thus eliminating any alignment changes due to thermal expansion.

4.7.3 Process piping considerations

4.7.3.1 General piping considerations

Next to the selection of operating speeds, proper piping design is the most important consideration in pump installation design. Poor piping is often the result of inattention to details which can lead to

- more than average down time
- higher maintenance costs
- loss of operating personnel confidence

Fig. 4.50 illustrates some of the following *recommended installation* guidelines for reciprocating pumps.

4.7.3.2 Suction piping

It is critical that the suction piping be sized to assure the $NPSH_A$ exceeds the $NPSH_R$ by the pump. Often, this can be arranged by elevating the suction tank or by providing a low-head centrifugal charge pump to feed the reciprocating pump. If the $NPSH_A$ is too low, valve breakage and pump maintenance costs will be excessive.

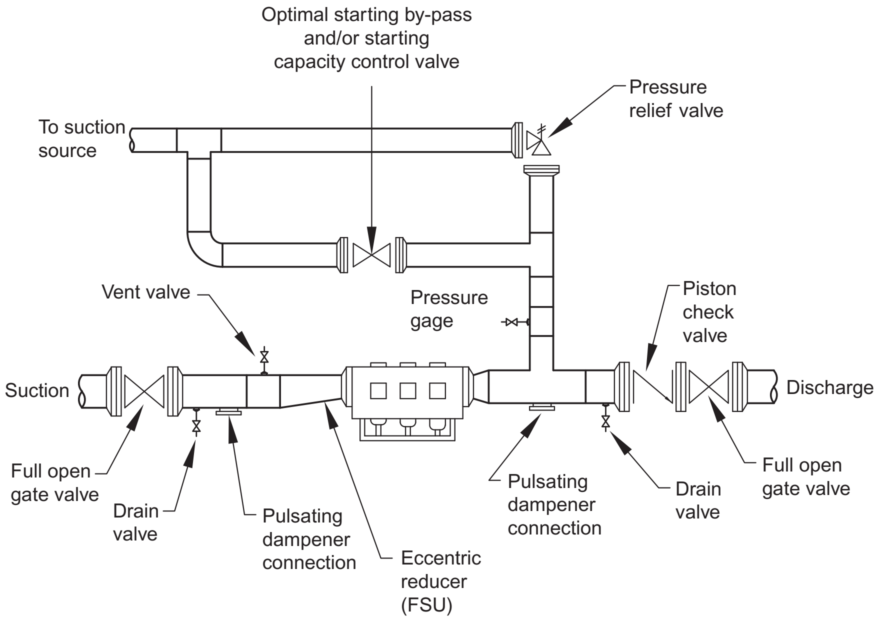


Fig. 4.50 Typical reciprocating pump installation.

The pump inlet size has no bearing on the required suction piping size. The available suction head and the suction requirement and losses must be considered and piping sized on this basis, regardless of the pump size.

Both the suction and discharge piping should be

- short
- direct
- free of bends, if possible
- minimum number of elbows and fittings
- avoid piping high points where vapors can become trapped
- elevation and plan changes should be laid out using 45° ells rather than 90° ells (if 90° ells are used, they should be long radius type)
- suction pipe diameter changes should be made with eccentric reducers, with flat side up (FSU) to eliminate gas pockets
- at least one nominal pipe size larger than the pump suction
- to allow pump isolation, a full opening block valve should be installed in the suction piping

The suction piping should include a strainer and a pulsation dampener, if required. The suction strainer should not be installed unless regular maintenance can be assured. A fluid starved condition, resulting from a plugged strainer, can cause more damage to the pump than solids ingestion.

The suction supply vessel outlet should be slightly higher than the pump inlet so that gases accumulating in the system may flow back to the vessel rather than through the pump. The supply vessel should have sufficient retention time for the evolution of "free" gas. The suction and bypass lines should enter the supply vessel below the minimum fluid level. A vortex breaker should be installed on the outlet of the supply vessel.

Table 4.19 lists some suggested maximum flow velocities for sizing suction and discharge piping for reciprocating pumps. The piping should be large enough so that the velocity limits are not exceeded. A low flow velocity for the suction piping is particularly important. Some companies use a maximum velocity of 1 ft/s (0.3 m/s) regardless of pump speed.

When two or more pumps are installed in parallel, each pump's suction line between the tank and the pump should be piped separately (rather than in common) to preclude mutually reinforced pulsations. In most cases, however, this procedure is not practical, and the suction lines are often manifold together. If manifold together, the lines should be sized so that the velocity in the common feed line is approximately equal to the velocities in the lateral lines feeding the individual pumps. This avoids abrupt velocity changes and minimizes acceleration head effects. (The acceleration head requirement for multiple pumps on a common suction line is not the sum of

Table 4.19 Maximum suction and discharge pipe velocities for reciprocating pumps

Pump speed, rpm	Suction velocity, ft/s (m/s)	Discharge velocity, ft/s (m/s)
<250	2 (0.6)	6(1.8)
250–330	1.5 (0.46)	4.5 (1.37)
>330	1 (0.3)	3 (0.9)

single pump requirements; it increases approximately by the square of the number of pumps. For example, 3 pumps operating on a common suction line require approximately nine times the acceleration head (H_A) of a single pump.)

4.7.3.3 Discharge piping

Just as fluid flows to a reciprocating pump in a pulsating flow pattern, it is discharged in the same manner. It has been shown that these pressure surges travel through the fluid in a straight line and are reflected back toward the source by restrictions or bends in the system. Also, when two or more pumps are discharging into a common header, these pressure surges may be amplified, causing damage to the pumps and piping. When designing a discharge piping system for reciprocating pumps:

- Avoid sharp bends, reducers, valves with less than full opening, and so on, near the pump; these may reflect pressure surges back toward the pump.
- Manifolds in which two or more pumps are tied into a common system should be located as far from the pumps as practical to allow dampening of surges. For most applications, 100 to 150 ft. (30 to 45 m) is adequate.
- Discharge piping should be securely anchored as near the pump as practical to prevent system vibrations from acting directly on the pump.
- Concentric reducers may be used, but they should be placed as near to the pump as practical.
- Pressure safety valves (PSVs) should be installed in the discharge piping near the pump and certainly upstream of the first block valve. Discharge/outlet piping should have a high enough pressure rating for potential future needs. Consideration must also be given to discharge PSVs flange ratings. PSVs must be set high enough to avoid inadvertent discharges.
- Directional piping changes should be made with 90° long radius ells.
- Pipe diameters should be based on the maximum velocities recommended in Table 4.19. Common practice is to size discharge pipe one nominal pipe size larger than the pump discharge connection.
- To facilitate priming and starting the following should be installed: Recycle (bypass) piped back to suction vessel, check valve, and block valve.
- If a pulsation dampener is not included in the initial installation, a flanged connection should be provided should pulsation attenuation be required in the future.

4.7.3.4 Piping hook-up considerations

Fig. 4.51 shows an example hook-up for two reciprocating pumps operating in parallel. Since the pump can be accidently started when the discharge block valve is closed, a PSV is installed in the discharge line to keep the pump from overpressuring the pipe and flanges. The PSV should be installed downstream of the pulsation dampener. It is also possible to leave the suction valve closed while the discharge valve is opened. Discharge fluid could leak through the discharge FSV and pump valves, pressuring up the suction piping, which is rated for ASME 150 Class. Thus, in this installation, a PSV was installed in the pump suction piping. Some companies believe a suction PSV is unnecessary because of the low probability of multiple failures.

An appendage dampener and cone strainer are installed in the suction line. An inline bladder/desurger and FSV are installed in the discharge line. The FSV protects against leakage from discharge when the pump is not running. It is preferable that this

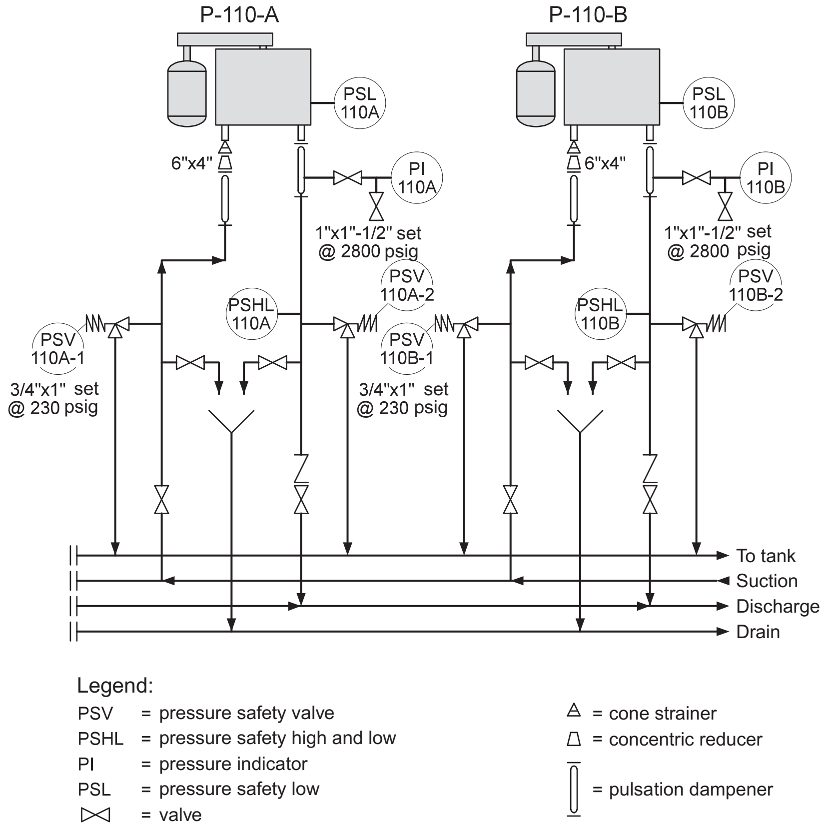


Fig. 4.51 Mechanical flow diagram of two reciprocating pumps in parallel.

be a piston FSV to keep it from chattering due to pressure pulsations. Nevertheless, swing FSVs are used successfully in some installations that follow the design practices for minimizing pulsations. Drain valves are provided so that the pump can be maintained easily, and an oil PSL is provided to shut-in the motor.

API RP 14C requires a PSH be installed on the discharge so that the pump will shutdown before the discharge PSV opens. It also requires that a PSL sensor on the discharge be installed to shutdown the pump in case of a large leak in the discharge piping. These two functions are carried out by one device.

4.7.4 Pulsation considerations

4.7.4.1 General considerations

Flow from a reciprocating pump is not uniform but pulsating. The oscillating motion of the plungers creates disturbances (pulsations) that travel at the speed of sound from pump cylinder to the piping system. Pulsations are a function of the pump piston/plunger velocity and the pump's internal valves, which generate liquid pulsations at integral multiples of the pump operating speed. The pressure pulsation can also excite both mechanical resonances and piping acoustic liquid resonances. Figs. 4.52 and 4.53 show the suction and discharge pressure variations versus time for a water injection station with 5 quintuplex pumps operating in parallel.

4.7.4.2 Adverse effects

The possible adverse effects of reciprocating pump pressure pulsations are as follows:

- Suction pulsations can cause the pressure level to drop instantaneously below fluid vapor pressures, resulting in cavitation. Cavitation can contribute to failure of pump parts such as valves, crossheads, rods, and so on. Severe cavitation can cause high piping vibrations and failures of vents, drains, and gauge lines.
- Pulsation forces can be so high that normal pipe clamps and supports may be ineffective in controlling the vibrations.
- Pulsations generated by the flow modulation from the pump plungers can be amplified by the acoustical resonances of the piping system, which can result in pump fluid end failures and piping failures due to the shaking caused by pressure pulsations.

4.7.4.3 Pulsation dampeners

In many installations when the piping system is sized in accordance with the recommendations in the process piping discussed in Section 4.7.3, there may be no need to

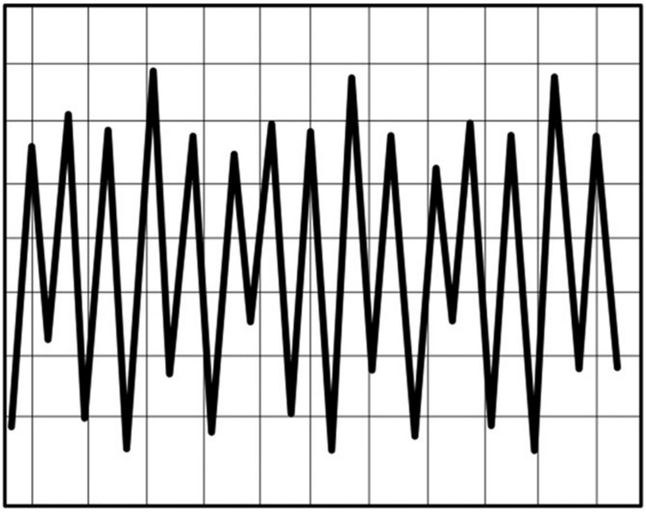


Fig. 4.52 Suction pressure variation versus time for a water injection station with 5 quintuplex pumps operating in parallel (mean suction pressure 516 psig).

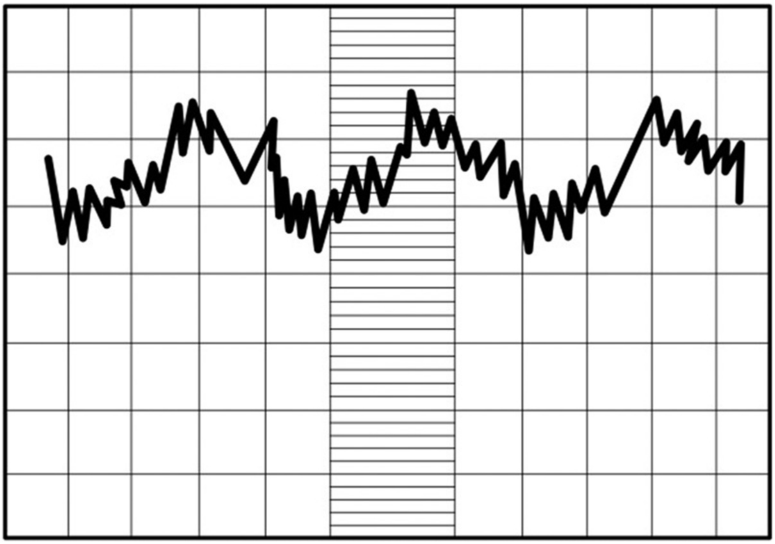


Fig. 4.53 Discharge pressure variation versus time for a water injection station with 5 quintuplex pumps operating in parallel (mean discharge pressure = 1450 psig).

install dampeners to reduce liquid pulsations. However, in most instances, it is possible to reduce pulsations further and thus reduce pump maintenance costs and piping vibration through the use of pulsation dampeners. Dampeners are recommended for all major multipump installations, unless computer analog studies indicate that they are not needed. Often it is cheaper to install the dampeners than to perform the detailed engineering studies to prove that they are not needed. For simple piping layouts and low to moderate pump speeds, pulsation dampeners are used to attenuate the effects of pulsating flows. Fig. 4.54 shows the discharge pressure variations "before" and "after" the installation of a pulsation dampener. Fig. 4.55 shows the suction and discharge pressure variations "before" and "after" the installation of a pulsation dampener.

Pulsation dampeners are normally installed on both the suction and discharge of the pump. Dampeners can be

- Liquid filled (Fig. 4.56)
- Gas cushioned (Fig. 4.57)
- Tuned-acoustical filter (Figs. 4.58 and 4.59)

4.7.4.3.1 Liquid-filled dampeners

A liquid-filled dampener is a large surge vessel located close to the pump. It uses the compressibility of the liquid to absorb pressure pulsations, and thus it works better on gaseous liquids (hydrocarbons, rich glycol) than on relatively gas-free liquids (water). The volume of the vessel is normally recommended to be ten times the pump

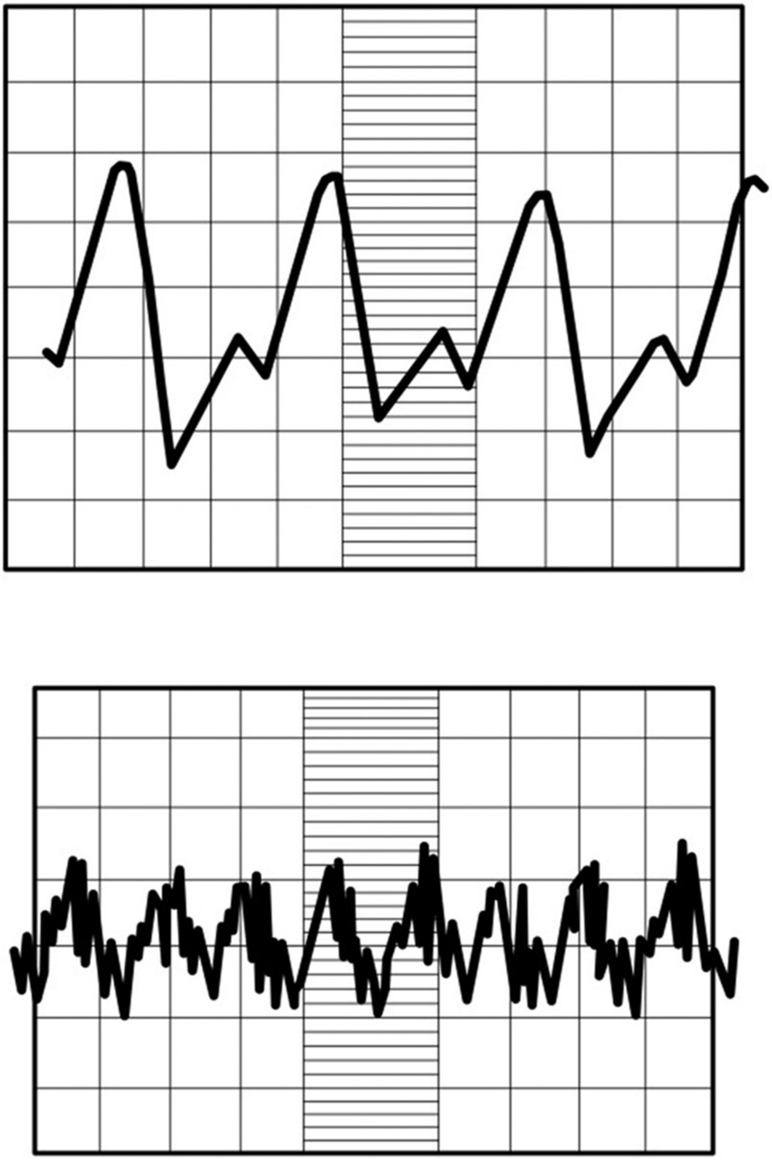


Fig. 4.54 Discharge pressure variations. Top: before installation of a pulsation dampener (pressure fluctuation = 48 psi) and Bottom: after the installation of a pulsation dampener (pressure fluctuation = 12 psi).

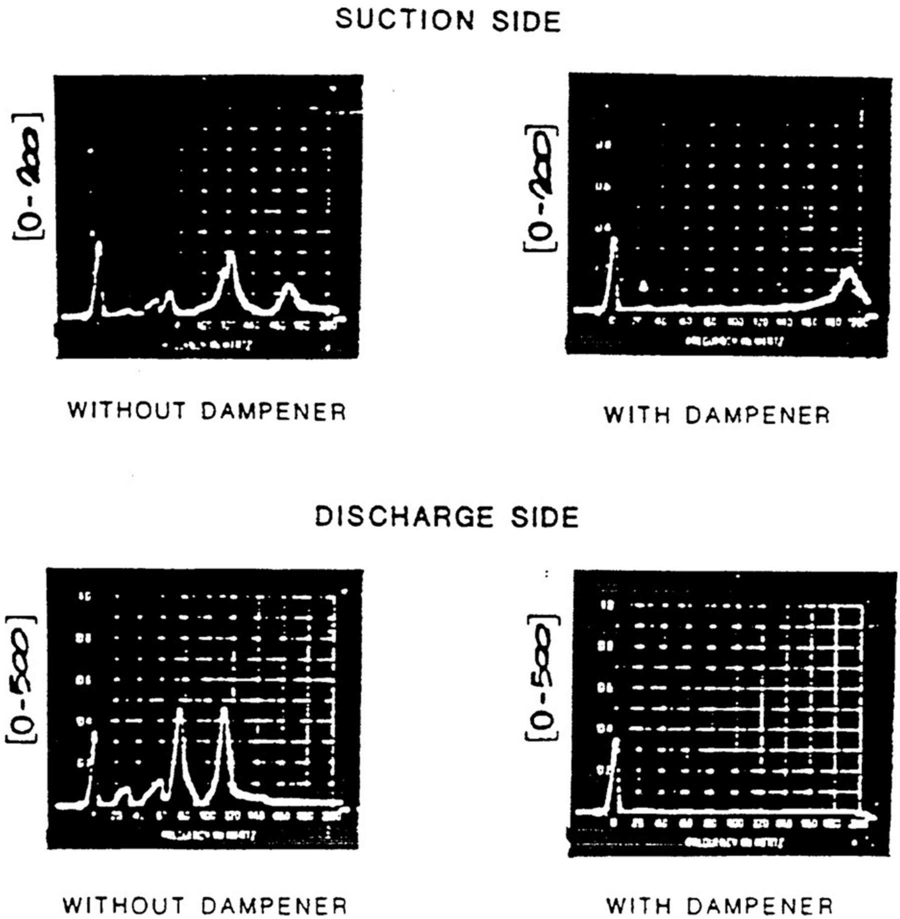


Fig. 4.55 Analog study showing the suction and discharge pressure variations "before" and "after" the installation of a pulsation dampener.

displacement (ft^3/min). Thus, for single-acting pumps, volume can be calculated from the following:

Field units

$$\mathrm{Vol} = 1.337q \tag{4.19a}$$

SI units

$$\mathrm{Vol} = 0.167q \tag{4.19b}$$

where

Vol = volume of surge tank, ft^2 (m^3)

Q = flow rate, gpm (m^3/h)

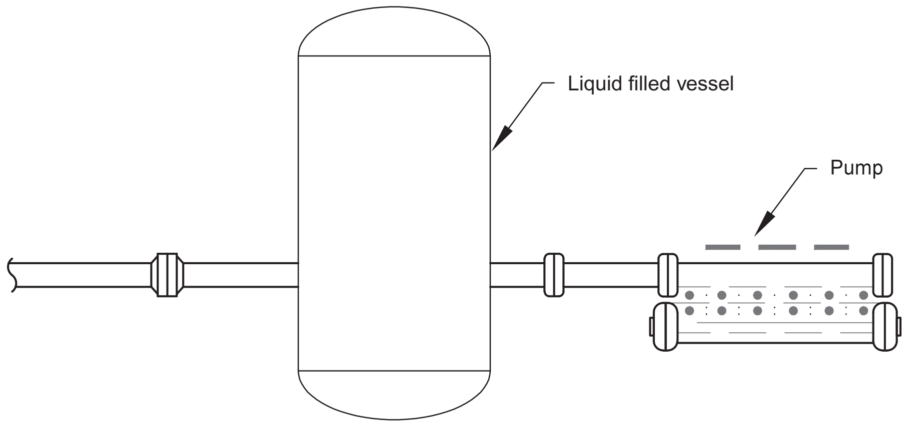


Fig. 4.56 Schematic diagram of liquid-filled dampener.

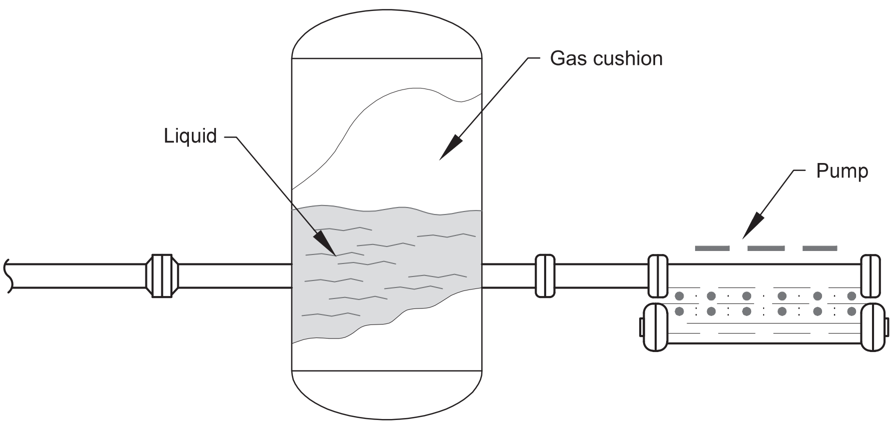


Fig. 4.57 Schematic diagram of gas-cushioned dampener.

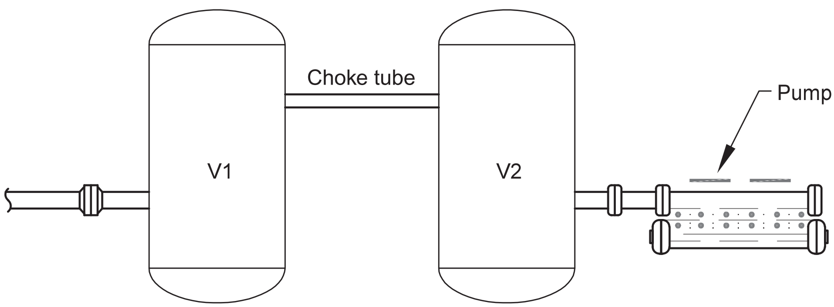
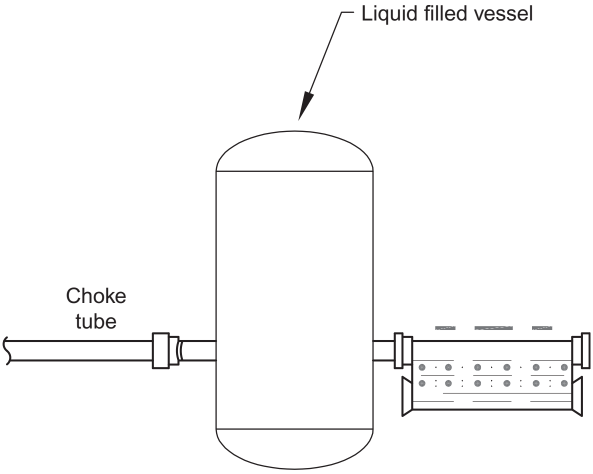


Fig. 4.58 Schematic diagram of "Type 1" tuned-acoustical filter.

Fig. 4.59 Schematic diagram of "Type 2" tuned-acoustical filter.



Advantages

- Maintenance free
- Insignificant pressure drop

Disadvantages

- Takes up a large amount of space
- Heavy when filled with water
- Expensive because they are rated to the MAWP of the piping system

Fig. 4.60 shows an installation for a quintuplex pump with liquid surge section and discharge dampeners.

4.7.4.3.2 Gas-cushioned dampeners

Fig. 4.61 shows typical gas-cushioned dampeners. The simplest type is a vertical surge bottle partially filled with pumped fluid and partially filled with gas. The highly compressible gas over time absorbs the pressure pulses. The gas in the vapor space is normally natural gas, which over a period of time can dissolve in the liquid. Thus a level gauge is installed so that the interface position can be observed and either more gas added or excess gas vented to maintain the required gas volume. Air should be avoided because it will increase the likelihood of corrosion, scale, and bacteria in water and increases the potential for an explosion in hydrocarbon service.

The required gas volume is given by

Field units

$$(\mathrm{Vol})_{\mathrm{g}} = \frac{KSd^2P}{748(\Delta P)} \tag{4.20a}$$

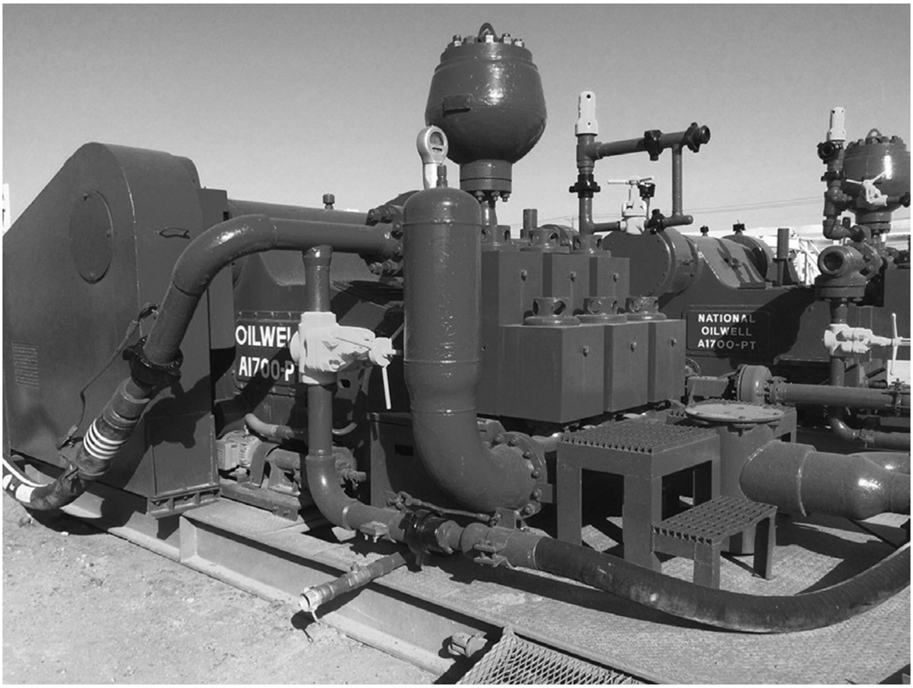


Fig. 4.60 Example of a quintuplex pump installation showing a liquid surge suction and discharge dampener.

SI units

$$(\mathrm{Vol})_g = \frac{KSd^2P}{4.328 \times 10^8(\Delta P)} \tag{4.20b}$$

where

$(\mathrm{Vol})_g$ = required gas volume, ft^3 (m^3)
K = pump constant (from Table 4.20)
S = pump stroke, in. (mm)
d = pump piston or plunger diameter, in. (mm)
P = average fluid pressure, psi (kPa)
ΔP = allowable pressure fluctuation, psi (kPa)
= Fig. 4.62

The allowable pressure pulsation amplitude is somewhat arbitrary. Fig. 4.62 can be used as an estimate if other information is not readily available.

Gas-cushioned dampeners are much smaller than liquid-filled dampeners but require monitoring of the interface. They are impractical in locations where the discharge pressure varies widely and the gas volume expands and contracts in response.

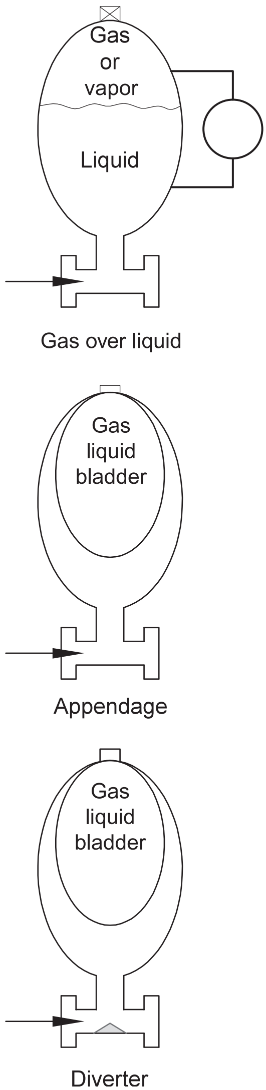


Fig. 4.61 Schematic diagram of typical gas-cushioned pulsation dampeners.

They require a vertical vessel otherwise gas will escape chamber. Thus they require a system for recharging.

Gas-cushioned dampeners can employ a pressured bladder to keep the gas from being absorbed in the liquid. They are small, inexpensive, and can be mounted in a horizontal vessel. They can have a configuration such as that shown in Fig. 4.61, or the bladder can be in the shape of the cylinder, as in the in-line bladder of

Table 4.20 Pump constant "*K*" for gas cushion design

Pump type	Action	*K*
Simplex	Single	0.67
Simplex	Double	0.55
Duplex	Single	0.55
Duplex	Double	0.196
Triplex	Single	0.098
Triplex	Double	0.196
Quintuplex	Single	0.360
Quintuplex	Double	0.060
Setuplex	Single	0.017
Setuplex	Double	0.034

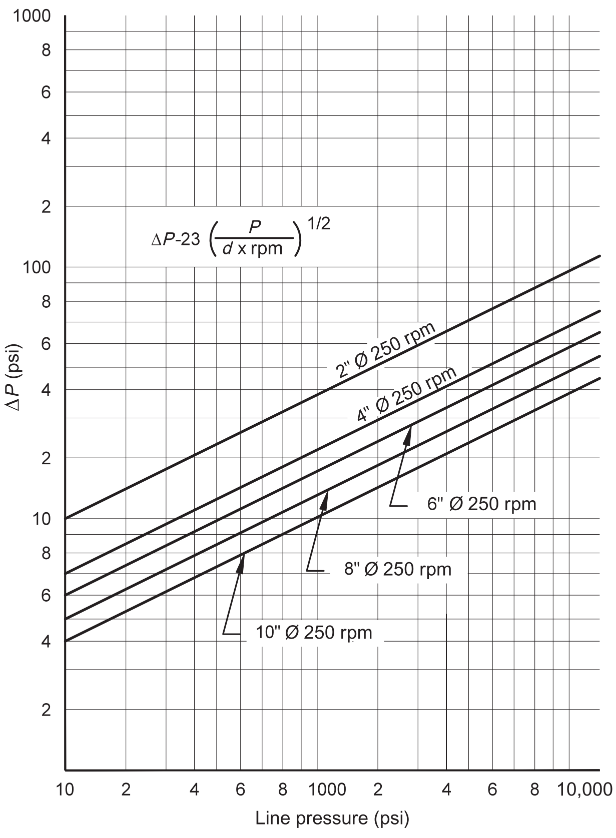


Fig. 4.62 Allowable pressure fluctuation versus frequency.

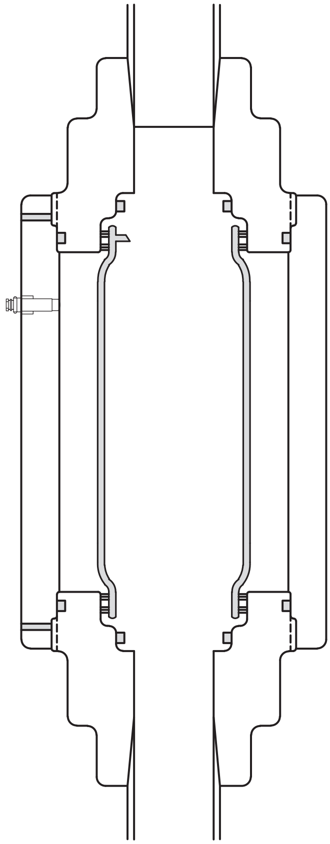


Fig. 4.63 Schematic diagram of an in-line "desurger" or dampener.

Fig. 4.63. The use of diverters in appendage-type dampeners or in-line configurations aids in attenuating high-frequency pulsations. The size of the gas volume depends upon the bladder properties and configuration of the design. For approximating purposes, Eqs. (4.18a, 4.18b) can be rewritten as follows:

Field units

$$(\mathrm{Vol})_{\mathrm{g}} = \frac{KAd^2P^2}{748(\Delta P)(P_c)} \tag{4.21a}$$

SI units

$$(\mathrm{Vol})_{\mathrm{g}} = \frac{KSd^2P^2}{4328 \times 10^8(\Delta P)(\Delta P_c)} \tag{4.21b}$$

where:

P_c = bladder precharge pressure, psi (kPa)

The bladder precharge pressure is normally set at 60% to 70% of average fluid pressure. Bladder-type dampeners are normally an economical solution and are in common use.

Bladder-type dampeners have the following limitations:

- Bladder is limited to applications below 300°F (149°C).
- Impractical at locations where pressures vary widely.
- Elastomer eventually will wear out and will need to be replaced.

4.7.4.3.3 Tuned-acoustical dampeners

Tuned-acoustical dampeners are formed when two liquid-filled vessels are connected by a short section of small diameter pipe called a choke tube. This system can be designed to have a specific resonant frequency. Pressure pulsations at frequencies above this level are attenuated considerably.

Type 1 is formed when two liquid-filled dampeners are connected by a short section of small diameter pipe called a "choke tube." System is designed, using a straightforward equation, to have a specific resonant frequency. Pressure pulsations at frequencies above this level are attenuated considerably.

Type 2 is formed by connecting the pump to a liquid-filled dampener. The choke tube is then installed between the dampener and the piping system. Acoustical filters are best done on computer analogs.

Acoustical dampeners have the following advantages:

- Can be used in high-temperature situations.
- Are essentially maintenance free.
- Temperature is not a limiting factor.
- Low maintenance costs.
- Filter can be tuned to suppress all frequencies low and high, that the pump will generate.

Acoustical dampeners have the following disadvantages:

- High relative cost.
- Requires a lot of space.
- Choke tube requires a high-pressure drop, which is a limitation in low NPSH applications (not used on suction lines in which NPSH may be a problem).

Fig. 4.64 shows a pump installation with an appendage bladder dampener on the suction and a tuned-acoustical filter on the discharge. The pressure vessel contains two sections, one above the other, with a choke tube connecting them internally. An extremely efficient dampener can be made by connecting two gas-cushioned dampeners in series with a short run of pipe that acts as a choke tube. The gas-cushioned dampeners attenuate low-frequency pulsations, and the choke tube arrangement alleviates high-frequency pulsations. Most installations do not require this complexity. The design of acoustical filters is best done on computer analogs and is beyond the scope of this text.

4.7.4.3.4 Performance curve comparison

Fig. 4.65 shows the gain factors for a tuned Helmholtz filter and a liquid-filled surge vessel under the same conditions. The filter gain factor is determined from Eq. (4.22):

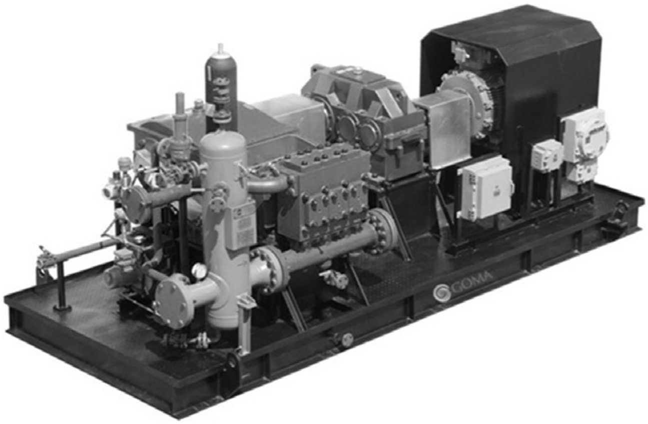


Fig. 4.64 Photograph of a pump with an appendage bladder dampener on the suction and a tuned-acoustical filter on the discharge.

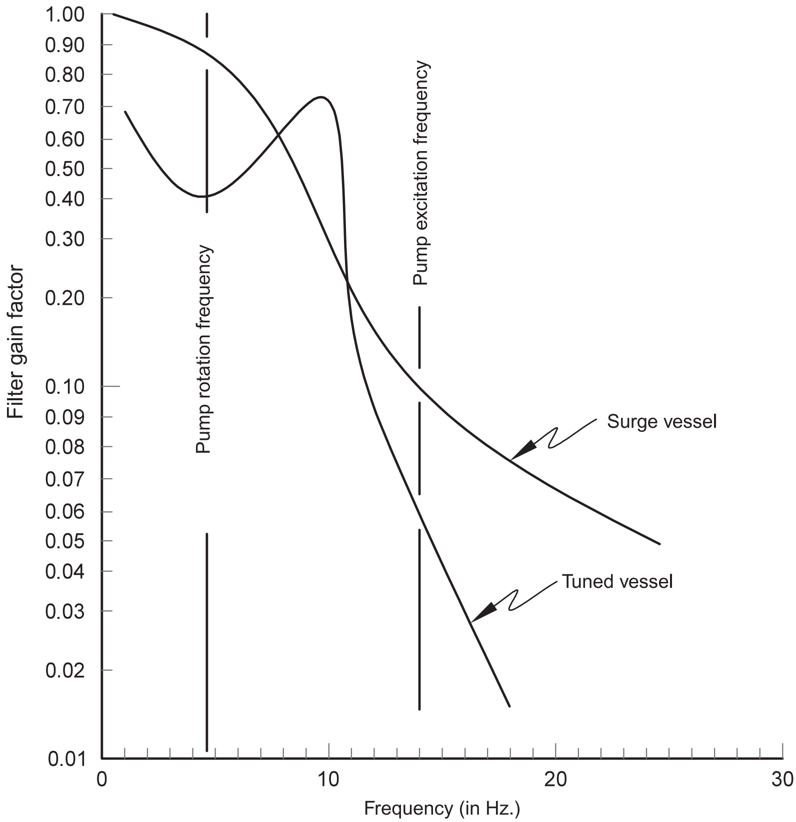


Fig. 4.65 Pump excitation frequency versus pulsation level.

$$(\text{Filter gain factor}) = \frac{(\text{Pressure pulse e filter inlet})}{(\text{Pressure pulse e filter outlet})} \tag{4.22}$$

Fig. 4.65 shows that at a pump excitation frequency (x-axis) of 14 Hz, attenuation through the tuned filter reduces the pulsation level at the filter discharge to 5% of the original discharge level. The surge vessel reduces the pulsation level to 10% of the original discharge level.

4.7.5 Pipe vibrations

To minimize pipe vibrations it is necessary to design pipe runs so that the "acoustic length" of the pipe run does not create a standing wave that adds to the pressure pulsations in the system. The acoustic length is the total overall length from end point to end point, including all elbows, bends, and straight pipe runs. Typical pipe runs with respect to acoustic length are considered to be

- Pipe length from suction tanks to the pump suction.
- Long pipe sections between pump and pulsation dampener.
- Pipe section between pump and manifold.

Piping should be designed so that piping runs with

- "*similar*" ends should not equal 0.5λ, λ, 1.5$\lambda\lambda$, 2λ… where λ is the acoustic wavelength
- "*dissimilar*" ends should not equal 0.25λ, 0.75λ, 1.25λ, 1.75λ… .

The ends of pipe runs are considered similar if both are either open or closed from an acoustic standpoint. They are dissimilar if one is open and one is closed. Examples of end classifications are as follows:

- If pipe size is dramatically reduced, it tends to act as a closed end.
- Orifice plates act as closed ends.
- Short-length flow nozzles act as closed ends.
- Abrupt pipe diameter enlargements act as open ends.
- Pipe tees presenting an increase in flow area (such as a tee with three equal legs) act as open ends.
- Pipe size changes that could occur smoothly over a pipe length corresponding to several pipe diameters do not act as a termination.

The acoustic wavelength (λ) is determined from the following:
Field units

$$\lambda = \frac{720\left(\frac{gB_L}{P}\right)^{\frac{1}{2}}}{(N)(n)}$$

$$\lambda = \frac{720\left(\frac{gB_L}{\rho}\right)^{\frac{1}{2}}}{(N)(m)} \tag{4.23a}$$

SI units

$$\lambda = \frac{10\left(\frac{gB_L}{\rho}\right)^{\frac{1}{2}}}{N(m)} \tag{4.23b}$$

where

B_L = bulk modules of the fluid, psi (kPa)
 = Fig. 4.66
g = gravitational constant, 32.2 ft/s^2 (9.81 m/s^2)
ρ = fluid density, Ib/ft^2 (kg/m^3)
N = pump speed, rpm (rps)
m = number of plungers

It is also desirable to assume that the natural frequency of all pipe spans is higher than the calculated pump pulsation frequency to minimize mechanical pipe vibrations. The pump pulsation frequency is given by the following:

Field units

$$f_P = \frac{Nm}{60} \tag{4.24a}$$

SI units

$$f_P = Nm \tag{4.24b}$$

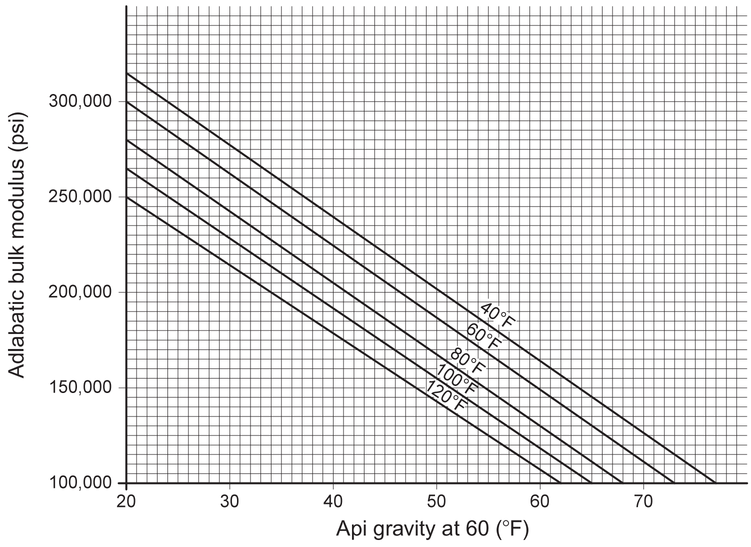


Fig. 4.66 Adiabatic bulk modules of crude oil versus API gravity. Courtesy of API.

where

f_p = pump pulsation frequency, cycles/s
N = pump speed, rpm (rps)

The natural frequency of pipe spans can be estimated from Fig. 4.67. Normally, mechanical pipe vibrations will not be a problem if:

- Pipe support spacing is kept short.
- Pipe is securely tied down.
- Support spans are not uniform in length.
- Fluid pulsations have been adequately dampened.

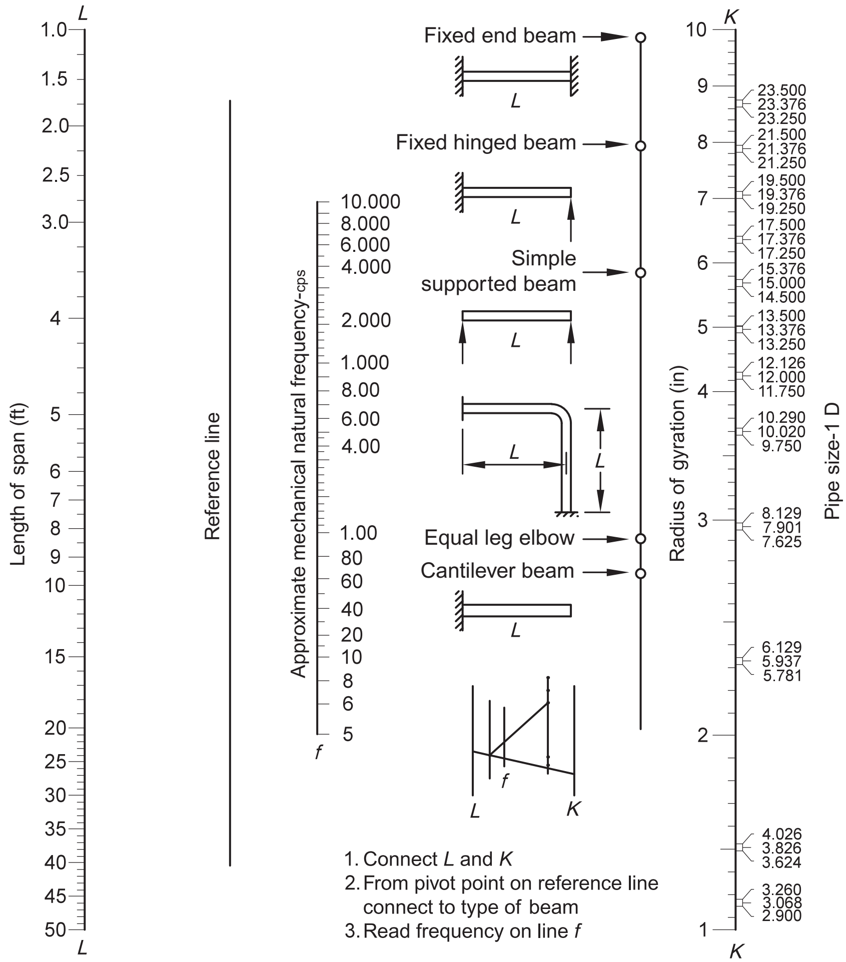


Fig. 4.67 Approximate mechanical natural frequency of pipe spans.

In summary, each pumping system must be individually evaluated to determine the specific dampening requirements. The slower a pump operates and the more cylinders it has, the chances of operation without dampeners or only with small dampeners are good. Double-acting piston pumps will generally result in smaller dampeners than single piston and plunger pumps.

4.7.6 Discharge PSV considerations

The purpose of the PSV on the pump discharge is to

- Protect the pump from excessive pressures.
- Protect the piping from excessive pressures.
- Protect the driver from excessive torque and horsepower.

The causes of excess pressure are clogged injection wellbore or accidental closure/failure of the pump discharge block valve. The PSV must be located between the pump and the first discharge block valve. The PSV should be as close to the pump as possible. The discharge of the PSV should be piped back to the source tank which avoids overheating the pumped fluid due to continued recirculation. The PSV and piping may have to be heat traced in high viscosity and in low-temperature services.

The PSV should be designed for intermittent safety use only and not be used to control pressure or flow. Most companies recommend that they be located in the piping even though some pumps have built-in relief valves. Recirculation flow with a built-in valve goes directly back to the suction side of the pump.

4.8 Reciprocating pump selection criteria

When selecting a reciprocating pump for oilfield applications it is necessary to determine if the proposed service calls for features required by API Standard 674 or if pumps built to manufacturer standards and modified for proposed oilfield usage are satisfactory.

There is normally little problem in choosing between the two basic types of pumps, direct-acting gas-driven pumps and crank-driven power pumps. Gas-driven pumps, once the workhorse of the industry, are generally limited to utility functions by the availability of compressed gas such as steam, air, or field gas. Power pumps, which are motor, turbine, or engine driven, are available in a wide spectrum of capacities and heads.

4.8.1 General design features

Items to consider when selecting a reciprocating pump are as follows:

1. The pump should be sized so that it is capable of delivering the required flow at the required discharge pressure at 85% of its maximum continuous rpm.
2. The pump fluid end should be a steel forging. The maximum allowable working pressure of the fluid end should exceed the design discharge pressure by 20%.
3. Pump valves should be removable and replaceable without disturbing the suction and discharge piping. Valve assemblies should be caged.

4. Pumps with a pressure lubrication system should be provided with a low oil pressure shutdown switch.
5. For pumped fluids that do not have lubrication qualities (e.g., seawater, fresh water), plungers should be lubricated. All lubrication tubing should be 316 SS.
6. All rolling contact bearings within the pump should be designed for a 3-year life.
7. Lube oil day tank should be provided for pump lube oil and plunger lube oil. Tanks should be sized for 30 days continuous operation. An automatic level instrument should be provided to maintain constant oil level. Day tanks should be equipped with level gauges.
8. Pump, driver, and all other appurtenances should be skid mounted. Piping connections should terminate at skid edge.
9. Skid material should be ASTM A36. Skid should be provided with a minimum of 4 leveling screws.
10. Unless specified otherwise, drain pans will not be provided by manufacturers. Where appropriate they should be specified.
11. Distance piece cover should be provided.
12. Piping connections of 1-1/4, 2-1/2, 3-1/2, 5, and 7 in. (31.75, 63.6, 88.9, 127, 177.8 mm) should be used.
13. Surfaces of rods or plungers in contact with packing shall be hardened or coated with hardened material.
14. All electrical connections should terminate at skid edge in a weatherproof, explosion proof, junction box.

4.8.1.1 Preparation for shipment

All openings should be plugged. Flanged openings should be covered with plywood or plastic protective covers. Instruments that can be damaged during shipment should be removed, boxed, and shipped separately.

4.8.1.2 Painting

Painting should be done in accordance with customer standards.

4.8.1.3 Documentation

Manufacturer should provide one (1) set of reproducible, good quality approval drawings that show skid outline and location of all customer connections. Drawing should be provided no later than two (2) weeks after receipt of order. Manufacturer should revise drawing to incorporate any comments. Operating and maintenance manuals should be provided.

4.8.1.4 Driver options

A designer should have some idea of the type of driver he desires and the type of power transmission device to be used to transfer power to the pump.

4.8.1.4.1 Choices of driver

- *Electric motor*: Usually <200 hp (149 kW), except for special situations.
- *Engine*: Normal for most oilfield applications over 200 hp (149 kW), but sometimes used down to 50 hp (37 kW).

- *Air/gas/hydraulic motor*: These usually require some transmissions, torque converters, or flexible couplings are available.

4.8.1.4.2 Choice of transmission

- Direct coupled: A number of different types of gear transmissions, torque converters, or flexible couplings are available.
- Belt driven: One of the most common types of power transmission in the oilfield, allows increased machinery life under load fluctuations.

4.8.2 Selecting a reciprocating pump

The following steps may be used to select a reciprocating pump.

1. Determine process duty.
2. Calculate liquid properties, if necessary.
3. Determine pipe pressure losses.
4. Calculate the suction head (same as for centrifugal pump).
5. Calculate the discharge head (same as for centrifugal pump)
6. Calculate the total head (same as for centrifugal pump)
7. Convert total head to pressure rise.
8. Calculate the $NPSH_A$. The procedure for calculating the $NPSH_A$ for a reciprocating pump is similar to that for a centrifugal pump except that acceleration head is included. Acceleration head is the force required to accelerate the fluid in the suction line.
9. Calculate brake horsepower.
10. Select particular pump. Using the pump manufacturers' literature and catalogs, select the pump for the conditions obtained in the calculation. If possible, avoid selecting the largest piston or plunger size for the pump case. Also avoid pumps which would have to operate continuously at maximum allowable speed.
11. Consult pump vendor. Discuss pump selection with the vendor for further recommendations and as a check of the selection procedure.
12. Prepare pump data sheet and specification.

Table 4.21 summarizes the advantages and disadvantages or positive displacement reciprocating pumps.

4.9 Reciprocating pump types

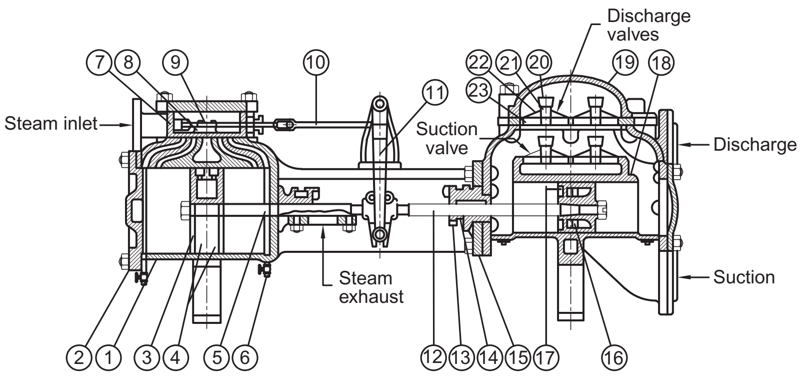
4.9.1 Reciprocating, piston, duplex, direct-acting gas-driven (process gas, air, steam) (Fig. 4.68)

Typical service	Relief drum pump-out Low-pressure boiler feed Water Sludge Sump pump Transfer

Pressure range	0 to 700 psig (0 to 4826 kPa)
Capacity range	0 to 500 gpm
Speed range	30 to 60 rpm (with piston speeds usually between 50 and 100 rpm)
Max allowable temperature	350°F (177°C)
Standard materials	Normal duplex, double-acting, simplex available. Normally cast iron and liquid ends with steel or bronze rods and trim
Specification	API 674
Typical control method	Speed control by throttling drive gas (steam, air, process gas), usually manual
Advantages	Self-priming Will operate at very low speeds High efficiency Minimizes liquid emulsification Handles viscous fluids No electrical power is required Suitable for unattended remote installation
Limitations	Pump speed is affected by system pressure Subject to vapor lock with low $NPSH_A$ Will stall with too-high system backpressure Pulsating flow can affect sensitive instrumentation downstream

Table 4.21 Summary of advantages and disadvantages of reciprocating pumps

	Advantages	**Disadvantages**	**Use**
Piston	Accurate flow rate	Jerky flow rate	Pure, slightly corrosive nonhazardous liquids
	Adjustable flow rate	High price	P: 100 bars
	High efficiency	Sensitive to particles	V: 20 m^3/h
		Limited chemical resistance	
Diaphragm	Accurate flow rate	Jerky flow rate	Slurry, corrosive, hazardous liquids
	Adjustable flow rate	High price	P: 100 bars
	High efficiency	Limited operating temperature	V: 20 m^3/h
	Less sensitive to particles		
	Very good chemical resistance		



Part no.	Name of part	Part no.	Name of part	Part no.	Name of part
1	Steam cylinder	10	Valve rod	19	Valve cover
2	Steam cylinder head	11	Valve rod lever	20	Valve stem
3	Steam piston	12	Liquid end piston rod	21	Valve spring
4	Steam piston ring	13	Gland	22	Valve
5	Steam end piston rod	14	Gland bushing	23	Valve seat
6	Condensate drain	15	Stuffing box		
7	Steam chest	16	Liquid piston		
8	Slide valve	17	Liquid piston rings		
9	Valve rod adjustment nut	18	Cylinder liner		

Fig. 4.68 Reciprocating pump-piston, duplex, direct-acting gas driven (process gas, air, steam).

4.9.2 Reciprocating pump, plunger power pump (Fig. 4.69)

Typical service	High pressure/low flow Gathering systems/pipelines Waterflood Well workover Mud pumps
Pressure range	500 to 6000 psi(3447 to 41,368 kPa)
Capacity range	10 to 600 gpm
Speed range	0 to 450 rpm
Max allowable temperature	400°F (204°C)
Standard materials	Available in duplex through nonuplex SS, although triplex most common Vertical configurations available up to 200 hp (149 kW) Crank driven with motor, turbine with gearbox, or engine drivers Steel liquid end Cast iron and steel power end Self-contained lubrication system
Specification	API 674 Variable speed or flow bypass

Typical control method	
Advantages	Higher pressures available than with piston pumps (up to 30,000 psi) Self-priming Constant delivery at high efficiency over wide pressure range Minimum fluid emulsification Handles viscous stocks Can run dry for a limited time
Limitations	Pulsing flow Low capacity High first cost and maintenance cost Low tolerance for abrasives Subject to vapor lock at low suction pressure with high vapor pressure fluids

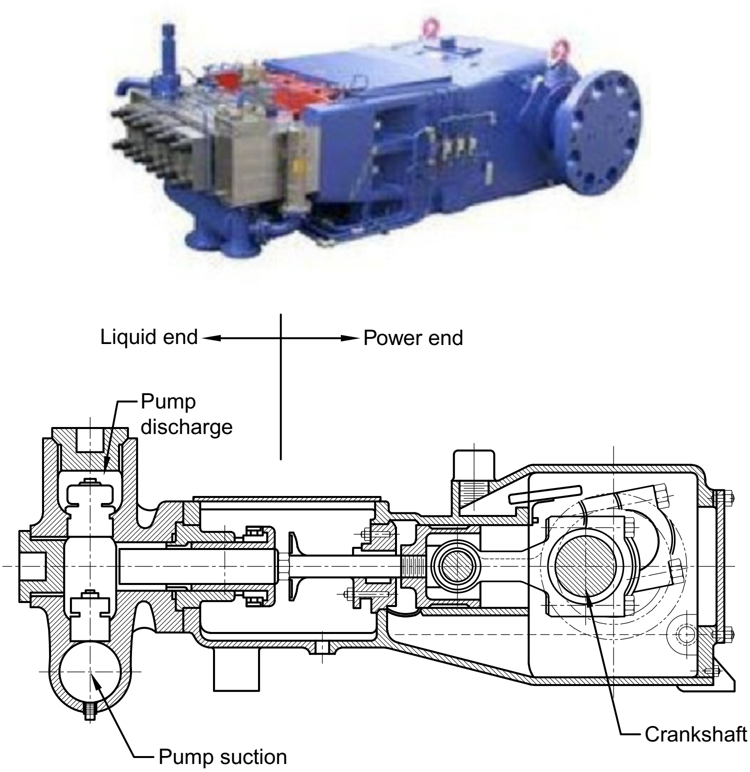


Fig. 4.69 Reciprocating plunger power pump. Top: assembled and Bottom: sectional. Courtesy of Worthington a Division of Ingersoll Dress Company.

4.10 Start-up considerations

4.10.1 Pump priming considerations

All pumps and associated piping must be purged of all vapors that may have accumulated since hydrostatic testing or while the unit was down. Purging a system is accomplished by priming to replace the vapors with liquids.

Procedure for a *positive static head* is as follows:

(1) Open the discharge vent valve.
(2) Open the discharge bypass valve.
(3) Partially open the suction block valve to allow fluid from the suction to fill the pump.
(4) Close the bypass valve and discharge vent valve when the fluid exits the vent valve.

Procedure for priming a pump with *static lift:*

(1) Open the discharge and suction block valves provided there is fluid in the discharge line.
(2) Pump fluid into the suction line and pump casing from an alternate source using an auxiliary pump.

4.10.2 Start-up guidelines

4.10.2.1 Initial start-up planning

Initial start-up planning should always be thoroughly planned and reviewed especially for critical services such as pumping hydrocarbons. All spare parts recommended by the manufacturer and deemed necessary by operating personnel should be on hand in case of a malfunction.

The operating manual should be reviewed with operating personnel. To help solve start-up problems, the manufacturer's representative should be present. All piping should be thoroughly cleaned before start-up and filled with fluid. For viscous or cold services, it may be necessary to provide heat tracing on the suction line.

4.10.2.2 Starting reciprocating pumps

The general start-up procedure is as follows:

(1) Suction and discharge block valves must be fully open. Failure to do this may result in severe damage to the pump.
(2) Discharge bypass valve can be turned back to a point where the pump is delivering the desired flow rate. For engine-driven pumps, the flow rate can be adjusted by changing the engine speed.
(3) The pump should run for several minutes under close supervision ensuring the desired fluid is being pumped, pumping system is not vibrating, and excessive heat buildup does not occur.
(4) After reaching operating temperatures, it is generally necessary to adjust the rod/plunger packing. Some leakage is necessary for lubrication and cooling. Trying to stop all leakage will cause heat buildup and premature pump failure.

4.11 Operations and maintenance considerations

4.11.1 Overview

Positive displacement pumps are very dependable machines. Extended pump life depends upon proper operation and good maintenance practices. Pump surveillance should be a daily routine for all people involved including engineers, operators, and maintenance personnel. This will ensure minimum down time, guard against the removal of a pump that is operating properly, and reduce expenses.

4.11.2 Troubleshooting

If a pump has no output or is producing less flow than expected, it is best to shut it down until the cause of the problem can be located. General troubleshooting procedure involves:

(1) Verify that liquid is reaching the pump.
- Check for an adequate supply of liquid at the suction source.
- Make certain that block valves on the suction line are fully open.
- Open bleed valves to bleed off trapped vapor.

(2) Check that downstream flowline is open.
- Check that all valves are open.

(3) If conditions seem normal to this point, briefly activate the pump and check for normal shaft rotation.
- If the shaft is not moving, or does not move at the correct speed, disconnect the pump from its driver and operate the driver by itself.
- If the driver operates properly, then the problem is in the pump.
- If not, ask a mechanic or electrician to inspect the driver.

(4) With the driver disconnected, rotate the pump shaft and listen for any abnormal noise (a stethoscope can be helpful).
- Remove the crankcase cover, rotate the crankshaft, and look for damage or misalignment.
- Check the crankshaft for bent or broken parts.

(5) Check packing for signs of scorching or poor lubrication.

(6) Check valves for damage.

4.11.3 Crankcase inspections and maintenance

The following inspections can be routinely performed without taking the pump out of service. Even so, the pump must be blocked in and depressurized and the pump driver must be locked out before the crankcase is opened.

(1) Crankcase oil should be sampled and inspected for brass or steel particles; these indicate abnormal bearing wear.
- Also check for water in the oil.

(2) Check cranks and connecting rods for discoloration caused by heat or misalignment.

(3) Turn the crankshaft manually (it may be necessary to first disconnect the driver).
- Listen for binding or abnormal bearing noise.

(4) Measure crankpin bearing clearances.
  - Four to six thousandths of an inch is a normal clearance. A higher reading could indicate bearing or crankpin wear.
  - Clearances can be adjusted by adding or removing some of the shims that separate the two halves of the bearing.

(5) Check crankshaft end play (i.e., the extent the shaft can move along its own length). This is done with a dial indicator and a pry bar.
  - Check the reading against manufacturer's specifications.
  - If endplay is excessive, shims must be removed from the crankshaft end bearing housing.
  - Shims of the same thickness must be removed from both housing until end play is within specifications.

(6) Check the clearance between the top of each crosshead and the crosshead guide with a feeler gauge.

If the above inspections indicate that bearings need to be changed, excessive clearance exists or a problem has been detected that is not immediately apparent, then the bearings, crankshaft, and crossheads will have to be removed for additional examination. In most cases, this requires a trip to the shop.

4.11.4 Liquid end inspection and maintenance

Inspecting the liquid end of the pump involves inspecting the valves, packing, and plunger (or piston). These inspections cannot be performed unless the pump has been

- isolated
- depressurized
- driver has been locked out

Valve chambers should be opened carefully since pressure still may be trapped inside even when the pump has been isolated and depressurized.

Inspection procedures include

(1) All valves should be removed and disassembled.
  - Worn or broken parts must be replaced.
  - Seating surfaces must be smooth.
  - Valve cavities must be cleaned before valves are reinstalled.

(2) The plunger must be removed and inspected for wear, especially at the point where it comes in contact with the packing.
  - The diameter of the plunger should be checked at several points with an outside micrometer.
  - If the plunger is badly worn or damaged, it must be replaced.

(3) New packing rings should be installed whenever the liquid end of a pump is overhauled.
  - If the pump is equipped with a force-feed lubricator, a lantern ring must be installed between two sets of packing rings immediately below the lubrication point on the packing box. Otherwise, lubricant will not be distributed to the packing and the packing will quickly wear out.

(4) The packing box bore should be inspected before new rings are installed. If the bore is worn, consider installing a stainless steel sleeve.

Example 4.3A. Reciprocating pump selection (field units)

Given:

An onshore water injection station has a pumping requirement of 6500 BWPD at a discharge pressure of 2150 psig. The discharge pressure requirement will remain relatively constant throughout the life of the field. As the reservoir fills up the injection flow rate will decrease to ~3000 BPD. The water is obtained from a source water well with a specific gravity of 1.0 and a temperature of 80°F. Fig. 4.70 illustrates the system that is to be considered for this problem. Table 4.22 is a sample Reciprocating Pump Data Sheet presenting the information required to select a pump.

In discussions with field operating personnel the following design criteria have been agreed:

(1) Pumps will be driven by a constant speed electric motor.
(2) Reciprocating pumps will be used.
(3) Bypassing of excess flow is acceptable as long as bypass rate can be minimized.
(4) Pump fluid end is to be a forging.
(5) Standby capacity is not required.

A Quintuplex pump is the First Choice with a triplex as an alternate.

Determine:

(1) Flow rate
(2) Discharge head
(3) Number of pumps
(4) Following mechanical features:
  (a) Valve assembly type
  (b) Type of wrist pin bearing
  (c) Type of packing
  (d) The pump material class is S-2, specify material for the following item: (1) fluid end and (2) plungers.

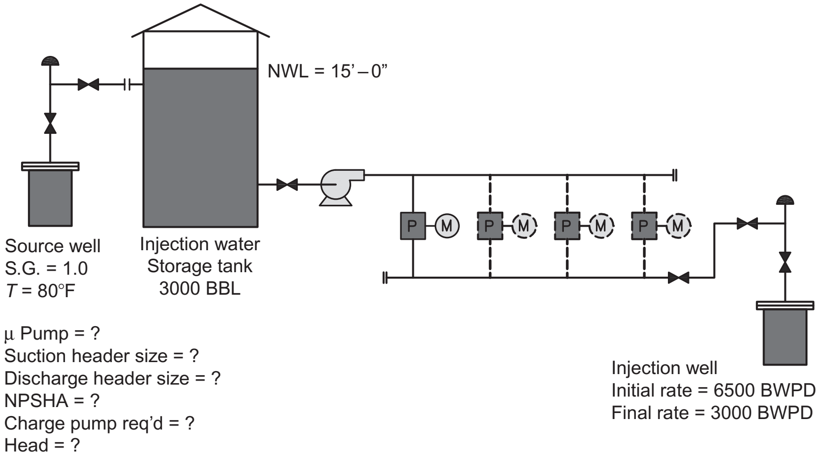


Fig. 4.70 Schematic flow diagram of a pump system.

Table 4.22 Example data sheet for Example 4.3A

A. Information required by pump manufacturer for establishing basic pump size
Flow (total flow for facility and flow for each pump)
Head
NPSH available
Liquid pumped
Pumping temperature
Pumped fluid = corrosive/erosive
B. Specify construction grade
API 674_______
Other_______
C. Specify major construction features
1. Materials of construction
Fluid end_______
Plunger_______
Valve_______
Valve spring_______
Packing gland_______
Cylinder liner_______
2. Valve assembly type
Caged_______
Tapered seat_______
3. Bearing type
Manufacturer's standard_______
Rolling contact_______
*Unless specific pump application dictates otherwise, bearing type should be Manufacturer's standard
4. Packing type
Square cut_______
V-ring_______
5. Lubrication type
Splash_______
Pressure_______
Plunger lubrication_______
6. Pressure rating of fluid end 120% of design discharge pressure_______
7. Pump baseplate
Common baseplate, pump skid and driver_______
Drain pan_______
Skid material_______
Skid leveling screws_______
D. Specify testing
Performance (witness/nonwitness)_______
Hydrostatic (witness/nonwitness)_______
NPSH (witness/nonwitness_______
E. Driver type
Electric motor_______
Gas engine_______
Other_______

Solution:

(1) The flow rate (final and initial)

$$\text{Initial flow rate} = 6500\,\text{BWPD} \times \frac{.0291\,\text{gpm}}{\text{BWPD}}$$
$$= 189\,\text{gpm}$$

$$\text{Final flow rate} = 3000\,\text{BWPD} \times \frac{0291\,\text{gpm}}{\text{BWPD}}$$
$$= 88\,\text{gpm}$$

(2) Discharge head

$\text{Head} = \frac{2.31}{SpGr} \times \Delta P = \frac{2.31(2150)}{1.0} = 4966\,\text{ft}$

Calculate the $NPSH_A$ from the following:

ΔP = [(SG)(TDH)]/(2.31)
= [(1)(15)]/(2.31)
= 6 psi

The preliminary data indicates that since there is very little $NPSH_A$, a small charge pump will likely be required.

After pump manufacturers offerings have been reviewed by the engineer, some decisions will have to be made regarding:

- NPSH of charge pump
- $NPSH_A$ for pumps selected
- Size of suction and discharge headers
- Losses due to friction

An estimate for electric power load per pump can be made from the following (assume the charge pump has a discharge pressure of 244 psi):

HHP = [(189)(0.5)(2150 − 6 − 244)]/(1714)
= 104.8 hp

Assume a 90% mechanical efficiency:

BHP = (104.8)/(0.90)
= 116.4 hp

Discharge head

Head = [(231)ΔP]/(SG)
= [(2.31)(2150)]/(1.0)
= 4966 ft

(3) Number of pumps

Option I: Two pumps each with 100% capacity (189 gpm)

This pump system will definitely be the most reliable as only one pump will need to be operated and 100% standby is provided. However, the installed cost will be high and the amount of flow bypassed in the later years' operation will be ~50% resulting in wasted horsepower.

Option II: Two pumps each with 50% capacity (94.5 gpm)

This pump system does not provide standby capacity. However, it does have the following advantages:

(a) Lower installed cost
(b) Less bypassed water in later years' operation

Option III: Three pumps each with 50% capacity (94.5 gpm)

This will cost more than Option II, but has the advantage of providing standby capacity.

Option IV: Three pumps each with 33% capacity (63 gpm)

This selection will have a higher maintenance, installation, and operating costs than Option I and II. There is less loss of capacity when one pump is down, as compared to the two—50% pump system. In later years' operation more water will have to be bypassed.

Option V: Four pumps each with 33% capacity (63 gpm)

This is identical to Option IV except standby capacity is provided.

Selection

Option II is selected since standby capacity at the initial throughput rate is really not necessary.

(4) Mechanical features
(a) Valve assembly type

Caged valve should be specified as it is currently the most common available, and does not have to be forced into fluid end, which brings on the possibility of damaging.

(b) Type of wrist pin bearing

Due to the low suction pressure journal bearings are acceptable for this application.

(c) Type of packing

Either type is suitable for this application. Since fluid will contain some sand V-ring packing should be provided or square cut packing with a lantern ring.

(d) Pump materials

The material selection of the pump parts should be in accordance with NACE RP 0475. For this problem, assume that the water is aerated and that no H_2S is present. Table 4.23 presents a completed data sheet for this problem. It is worth noting that some thought should be given to the pressure rating of the piping and pump fluid end. The 2150 psi discharge pressure requirement and subsequent 900 ASME pressure rating is close to the maximum design working pressure (DWP) of valves, flanges, and fittings, leaving no margin for PSH and PSV setting. The reservoir engineer should be interviewed closely to determine the accuracy and need for this surface injection pressure. If still required, a change in fluid end to a higher pressure rating would probably be recommended.

This example illustrates that all pumping situations present engineer with several choices, alternatives, and unanswered questions. In the real world decisions must be made based on individual experience, on the engineering principles presented here, and on sound engineering judgment.

Table 4.23 Completed reciprocating pump data sheet for Example 4.3A

A. Information required by pump manufacturer for establishing basic pump size	
Flow	
Total flow for facility	189 gpm
Flow for each pump	94.5 gpm (2 required)
Head	15 ft
NPSH available ($NPSH_A$)	6 psi
Liquid pumped	Salt water—$SG = 1.0$
Pumping temperature	80°F
Pumped fluid = corrosive/erosive	Mildy corrosive
Pressure rating of fluid end	900 ANSI
B. Specify construction grade	
API 674	
Other	NACE NR04-75/fluid parts
C. Specify major construction features	
1. Material of construction	
Fluid end	Cast aluminum bronze
Plunger	Ceramic
Valve	Mfg std—note in response
Valve spring	Mfg std—note in response
Valve cage	Mfg std—note in response
Packing gland	Mfg std—note in response
Cylinder liner	Mfg std—note in response
2. Valve assembly type	
Manufacturer's standard	
Caged	Yes
Tapered seat	
3. Bearing type	
Manufacturer's standard	Yes
Rolling contact	
Hydrodynamic	
4. Packing type	
Manufacturer's standard	
Square cut	Acceptable w/lantern ring
V-ring	Preferred
5. Lubrication type	
Manufacturer's Standard	Yes
Splash	
Pressure	Preferred
6. Pump baseplate	
Common baseplate, pump skid, and driver	Skid mounted
Drain pan	None required

Continued

Table 4.23 Continued

Skid material	A-36 CS
D. Required testing	
Performance (witness/nonwitness) Hydrostatic (witness/nonwitness) $NPSH_R$ (witness/nonwitness)	Witness Witness Nonwitness
E. Driver type	
Electric motor Gas engine Other	TEFC electric motor, 440 VAC 65% reduced voltage motor starter required
F. Power transmission	
Direct coupled Belt driver	 Yes

Example 4.3B. Reciprocating pump selection (SI units)

Given:

An onshore water injection station has a pumping requirement of 43 m^3/h at a discharge pressure of 14,800 kPa. The discharge pressure requirement will remain relatively constant throughout the life of the field. As the reservoir fills up the injection flow rate will decrease to ~20 m^3/h. The water is obtained from a source water well with a specific gravity of 1.0 and a temperature of 27°C. Fig. 4.70 illustrates the system that is to be considered for this problem. Table 4.22 is a sample Reciprocating Pump Data Sheet presenting the information required to select a pump.

In discussions with field operating personnel the following design criteria have been agreed:

1. Pumps will be driven by a constant speed electric motor.
2. Reciprocating pumps will be used.
3. Bypassing of excess flow is acceptable as long as bypass rate can be minimized.
4. Pump fluid end is to be a forging.
5. Standby capacity is not required.

A Quintuplex pump is the First Choice with a triplex as an alternate.

Determine:

1. Flow rate
2. Discharge head
3. Number of pumps
4. Following mechanical features:
   - **(a)** Valve assembly type
   - **(b)** Type of wrist pin bearing

(c) Type of packing
(d) The pump material class is S-2, specify material for the following item: (1) fluid end and (2) plungers.

Solution:

1. The flow rate (final and initial)
Initial flow rate = $43\,m^3/h$
Final flow rate = $20\,m^3/h$

Calculate the $NPSH_A$ from the following:

$$\Delta P = \frac{(SG)(\text{Head in in})}{0.102}$$
$$\Delta P = \frac{(1)(4.6)}{0.102}$$
$$\Delta P = 45\,\text{kPa}$$

The preliminary data indicates that since there is very little $NPSH_A$, a small charge pump will likely be required.

After pump manufacturers offerings have been reviewed by the engineer, some decisions will have to be made regarding:

- NPSH of charge pump
- $NPSH_A$ for pumps selected
- Size of suction and discharge headers
- Losses due to friction

An estimate for electric power load per pump can be made from the following (assume the charge pump has a discharge pressure of 244 psi):

$$\begin{aligned} HHP &= [(43)(0.5)(14{,}800 - 45 - 1680)]/(3600) \\ &= 78\ \text{kW} \end{aligned}$$

Assume a 90% mechanical efficiency:

$$\begin{aligned} BHP &= (78)/(0.90) \\ &= 86.8\ \text{kW} \end{aligned}$$

2. Discharge head

$$\text{Head} = \frac{0.102}{SG} \times \Delta P = \frac{0.102(14,800)}{1.0} = 1510\,\text{m}$$

3. Number of pumps

Option I: Two pumps each with 100% capacity (189 gpm)

This pump system will definitely be the most reliable as only one pump will need to be operated and 100% standby is provided. However, the installed cost will be high and the amount of flow bypassed in the later years' operation will be ~50% resulting in wasted horsepower.

Option II: Two pumps each with 50% capacity (94.5 gpm)

This pump system does not provide standby capacity. However, it does have the following advantages:

(1) Lower installed cost.
(2) Less bypassed water in later years' operation.

Option III: Three pumps each with 50% capacity (94.5 gpm)

This will cost more than Option II, but has the advantage of providing standby capacity.

Option IV: Three pumps each with 33% capacity (63 gpm)

This selection will have a higher maintenance, installation, and operating costs than Option I and II. There is less loss of capacity when one pump is down, as compared to the two—50% pump system. In later years' operation more water will have to be bypassed.

Option V: Four pumps each with 33% capacity (63 gpm)

This is identical to Option IV except standby capacity is provided.

Selection

Option II is selected since standby capacity at the initial throughput rate is really not necessary.

4. Mechanical features
(a) Valve assembly type

Caged valve should be specified as it is currently the most common available, and does not have to be forced into fluid end, which brings on the possibility of damaging.

(b) Type of wrist pin bearing

Due to the low suction pressure journal bearings are acceptable for this application.

(c) Type of packing

Either type is suitable for this application. Since fluid will contain some sand V-ring packing should be provided or square cut packing with a lantern ring.

(d) Pump materials

The material selection of the pump parts should be in accordance with NACE RP 0475. For this problem, assume that the water is aerated and that no H_2S is present. Table 4.24 presents a completed data sheet for this problem. It is worth noting that some thought should be given to the pressure rating of the piping and pump fluid end. The 14,800 kPa discharge pressure requirement and subsequent 900 ASME pressure rating is close to the maximum Design Working Pressure (DWP) of valves, flanges, and fittings, leaving no margin for PSH and PSV setting. The reservoir engineer should be interviewed closely to determine the accuracy and need for this surface injection pressure. If still required, a change in fluid end to a higher pressure rating would probably be recommended.

This example illustrates that all pumping situations present engineer with several choices, alternatives, and unanswered questions. In the real world decisions must be made based on individual experience, on the engineering principles presented here, and on sound engineering judgment.

Table 4.24 Example data sheet for Example 4.3B

A. Information required by pump manufacturer for establishing basic pump size	
Flow	
Total flow for facility	
Flow for each pump	
Head	
NPSH available ($NPSH_A$)	
Liquid pumped	
Pumping temperature	
Pumped fluid = corrosive/erosive	
Pressure rating of fluid end	
B. Specify construction grade	
API 674	
Other	
C. Specify major construction features	
1. Material of Construction	
Fluid end	
Plunger	
Valve	
Valve spring	
Valve cage	
Packing gland	
Cylinder liner	
2. Valve assembly type	
Manufacturer's standard	
Caged	
Tapered seat	
3. Bearing type	
Manufacturer's standard	
Rolling contact	
Hydrodynamic	
4. Packing type	
Manufacturer's standard	
Square cut	
V-ring	
5. Lubrication type	
Manufacturer's standard	
Splash	
Pressure	
6. Pump baseplate	
Common baseplate, pump skid, and driver	
Drain pan	
Skid material	

Continued

Table 4.24 Continued

D. Required testing	
Performance (witness/nonwitness) Hydrostatic (witness/nonwitness) $NPSH_R$ (witness/nonwitness)	
E. Driver type	
Electric motor Gas engine Other	
F. Power transmission	
Direct coupled Belt driver	

4.12 Exercises

1. The most commonly used positive displacement pumps in production operations are____and ____

2. Using Fig. 4.70, identify the components of a reciprocating plunger pump.

a.	________________	h.	________________
b.	________________	i.	________________
c.	________________	j.	________________
d.	________________	k.	________________
e.	________________	l.	________________
f.	________________	m.	________________
g.	________________	n.	________________

3. The primary disadvantage of a reciprocating pump is
- **a.** Lower efficiencies
- **b.** High $NPSH_R$
- **c.** Pulsating flow
- **d.** Cannot be used for water service
- **e.** Low pressure output

4. In reciprocating pumps, the fluid is displaced by a ________________ or a ________________________________.

5. The displacement of fluid by a piston or plunger pump is ____ ________________________

Questions 6–9

A positive displacement pump is required to transfer 6000 barrels per day of crude oil from a storage facility to a refinery. The pump is to be operated 80% of the time. The discharge pressure required is 975 psig. Using the figure on the following page.

6. What type pump would be selected?
- **a.** Type A
- **b.** Type B
- **c.** Type C
- **d.** Type D
- **e.** Type E

7. What is the maximum discharge pressure?
- **a.** 800 psig
- **b.** 1025 psig
- **c.** 2700 psig
- **d.** 670 psig
- **e.** 3200 psig

8. What size plungers would be used?
- **a.** 3 x 4-3/8
- **b.** 2x5-1/2
- **c.** 1-1/2x2-1/2
- **d.** 3-1/4x5-1/2
- **e.** 2-1/4x4-3/8

9. What is the maximum rpm for this type pump?
- **a.** 550
- **b.** 450
- **c.** 380
- **d.** Any of the above
- **e.** None of the above

		Displacement			
Pump type	**Size**	**Maximum rpm**	**Barrel, per day**	**Gallon. per minute**	**Maximum discharge pressure psig**
A	¾ x 2 ½	550	270	7 9	5660
	3/8 x 2 ½	550	367	10 7	4160
	1 x 2 ½	550	480	14	3190
	1 1/8 x 2 ½	550	607	17.7	2520
	1 ¼ x 2 ½	550	750	21 9	2040
	1 3/8 x 2 ½	550	910	26.5	1680
	1 ½ x 2 ½	550	1080	31 6	1420
B	1 x 4 3/8	450	690	20 1	6050
	1 1/8 x 4 3/8	450	870	25 4	4780
	1 ¼ x 4 3/8	450	1076	31 4	3870
	1 3/8 x 4 3/8	450	1300	38 0	3200
	1 ½ x 4 3/8	450	1550	45 2	2700
C	1 ¼ x 5 ½	380	1140	33 3	6930
	1 1/8 x 5 ½	380	1380	40 3	5720

Continued

Continued

Pump type	Size	Displacement			Maximum discharge pressure psig
		Maximum rpm	Barrel, per day	Gallon. per minute	
	1 ½ x 5 ½	380	1645	48 0	4810
	1 3/8 x 5 ½	380	1930	56 3	4100
	1 ¾ x 5 ½	380	2240	65 3	3530
	1 2/8 x 5 ½	380	2570	74.9	3080
D	1 1/8 x 4 3/8	450	1300	38	3200
	1 ½ x 4 3/8	450	1550	45.2	2700
	1 3/8 x 4 3/8	450	1820	53	2300
	1 ¾ x 4 3/8	450	2110	61.5	1970
	2 x 4 3/8	450	2750	80.3	1510
	2 ¼ x 4 3/8	450	3450	101.5	1200
	2 ½ x 4 3/8	450	4300	125.5	970
	2 ¾ x 4 3/8	450	5200	151.8	800
	3 x 4 3/8	450	6190	180.7	670
E	1 ¾ x 5 1/2	380	2240	65.3	3530
	2 x 5 ½	380	2920	85.3	2700
	2 ¼ x 5 ½	380	3700	108	2140
	2 ½ x 5 ½	380	4570	133	1730
	2 ¾ x 5 ½	380	5530	161	1430
	3 x 5 ½	380	6580	192	1200
	3 ¼ x 5 ½	380	7690	224	1025
	3 ½ x 5 ½	380	8950	261	885

10. To account for additional energy that must be provided on the suction side of a reciprocating pump the $NPSH_A$ is reduced by the ____

a. Pulsations per minute
b. C factor
c. Acceleration head
d. Plunger capacity × 80%
e. Packing efficiency

11. Using Fig. 4.71, identify the components of a reciprocating pump installation

a. ____________________ **f.** ____________________
b. ____________________ **g.** ____________________
c. ____________________ **h.** ____________________
d. ____________________ **i.** ____________________
e. ____________________ **j.** ____________________

12. Name two of the most commonly used valves in a positive displacement pump are _______________ and ____________________ valves.

13. Plunger pumps use __________________ packing, while piston pumps use ___________________________ packing.

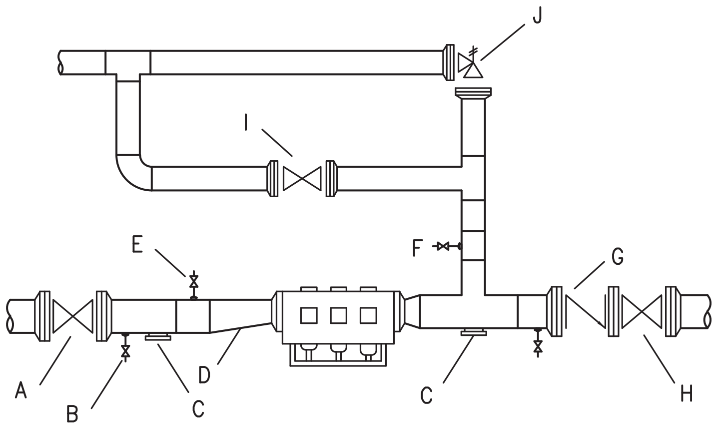


Fig. 4.71 Schematic for Problem 11.

14. To smooth the discharge flow from a reciprocating pump ______________________________ are most often used.
   a. Control valves
   b. Pulsation dampeners
   c. Acoustical filters
   d. All of the above
   e. B and C only

15. A discharge check valve for a reciprocating pump should be a ________________________________ type.
   a. Globe
   b. Piston
   c. Plunger
   d. Swing
   e. Water

16. What is the head required to accelerate 8000 barrels of crude oil per day from a storage tank to a pipeline for the following conditions:
   Pump: Reciprocating single-acting duplex, 300 rpm
   Suction piping: 6-in., Schedule 40, ID = 6.065 in., 5 feet long
   a. 3 ft
   b. 8 ft
   c. 12 ft
   d. 17 ft
   e. 29 ft

17. Liquid carbon dioxide at −10°F is to be pumped into a reservoir in a miscible flood EOR project. The injection pump is a quintuplex pump with 1.5-in. diameter plungers and a 5-in. stroke. Pump speed is 282 rpm and the pump capacity is 46 gpm. The suction line is 50-ft long and has a 4.026-in. diameter. Calculate the acceleration head.

18. It is desired to move 10,000 BPD of a 57° API condensate from atmospheric storage tanks to a pipeline operating at 500 psia. The distance from the storage area to the main pipeline is 10 miles. What size pipe would you recommend for the lateral line? What size pump would you specify (power and number of stages)? The pump must have a minimum efficiency of 60%. It is recommended that the pump speed be 330 rpm. The condensate viscosity is 0.9 cp.

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