
SECTION 3.2

POWER PUMP DESIGN AND CONSTRUCTION

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The power pump, as a piece of rotating equipment, would seem to have a great deal in common with a reciprocating gas compressor or diesel or gas engine. All have crankshafts, bearings, connecting rods, and lubricating oil systems that use the pump frame as an oil reservoir. All have a fluid or gas end that contains relatively high pressure and includes some type of valving that enables the equipment to perform its required task.

Due to the unique operating characteristics of the power pump, very little design criteria or construction details can be directly transferred from these other mechanisms. Because of the high compressibility of most gases, the resulting cyclic loadings are applied to the various rotating, reciprocating, and pressure-retaining components in a smoother manner. Therefore, gas-driven or gas-pumping mechanisms can operate at speeds three or four times higher than a reciprocating pump with good reliability and low vibration levels.

It has been said that a power pump has more in common with a jackhammer because of the constant high-cycle shock loading that it must be designed to withstand. Therefore, very conservative material selections, stress limits, and factors of safety have been established by the pump designers for all continuous duty applications. Intermittent duty applications can be met using less stringent design criteria.

Seldom is a power pump designed for one specific application having one specific set of operating conditions. Consequently, few pump designs are truly optimized. The best economical approach has been to design a power end capable of accepting a number of fluid ends, which in turn allows the pump to operate over a wide range of hydraulic conditions. By using different plunger diameters for a specific operating pressure, the designer tries to "load up" the power to its maximum continuous rod load rating while maintaining the pump speed at a level just sufficient to provide the required flow rate.

A power pump has its cylinders operating in parallel with interconnecting suction and discharge manifolds. Additional cylinders are sometimes added to a basic power-end design in order to obtain higher flow rates. The most common examples of this approach are the quintuplex pumps that have an 80-percent component interchangeability with

triplex pumps. Triplex (three cylinder), quintuplex (five cylinder), septuplex (seven cylinder), and nonoplex (nine cylinder) pumps have been designed that have over a 90-percent interchangeability of parts. Only the power frame and crankshaft are unique to a specific pump size.

Factors of safety are used in power pump design and application. A connecting rod assembly, for instance, normally has a safety factor in excess of a 3:1 ratio because if a connecting rod fails in operation, it normally destroys many other power-end parts. A nodular iron frame will have a high safety factor also, primarily due to the requirement for thick wall sections to mount interfacing parts and to reduce operating noise levels. Some fluid-end expandable parts in the stuffing box and valves have safety factors below a 1.5:1 ratio because these parts are normally replaced before they would fail due to fatigue.

There has been a trend in past years in most industries to purchase power pumps that operate at higher and higher speeds. This has been driven by the economics of buying smaller, higher speed equipment versus the older, large, slower speed pumps. This trend seems to be losing some momentum, especially in critical applications. It remains for the pump purchaser and pump supplier to jointly evaluate each application and try to arrive at a pump selection that will provide the most reliable performance for the lowest installed cost and operating cost.

LIQUID END

The liquid end consists of the cylinder, the plunger or piston, the valves, the stuffing box, the manifolds, and the access covers. Figure 1 shows the liquid end of a horizontal pump, and Figure 2 shows the same for a vertical pump.

Cylinder (Working Barrel) The cylinder is the body where the pump pressure to overcome discharge pressure is developed. It is continuously under fatigue. Cylinders on many horizontal pumps have suction and discharge manifolds integral with the cylinder. Vertical pumps have separate manifolds.

A cylinder containing the passages for more than one plunger is referred to as a *single cylinder*. When the cylinder is used for one plunger, it is called an *individual cylinder*. Individual cylinders are used where developed stresses or replacement costs are high.

Cast cylinders are usually limited to the developed pressures listed in Table 1. Forged cylinders are made from 1020 and 4140 carbon steel; 304L, 316L, 17-4 PH, and 15-5 PH stainless steels; and nickel aluminum bronze. In recent years, duplex stainless steels have seen an increase in usage when higher strength and corrosion resistance is required.

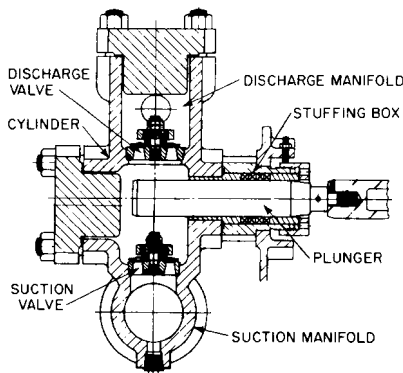


FIGURE 1 A liquid-end horizontal pump (Flowsolve Corporation)

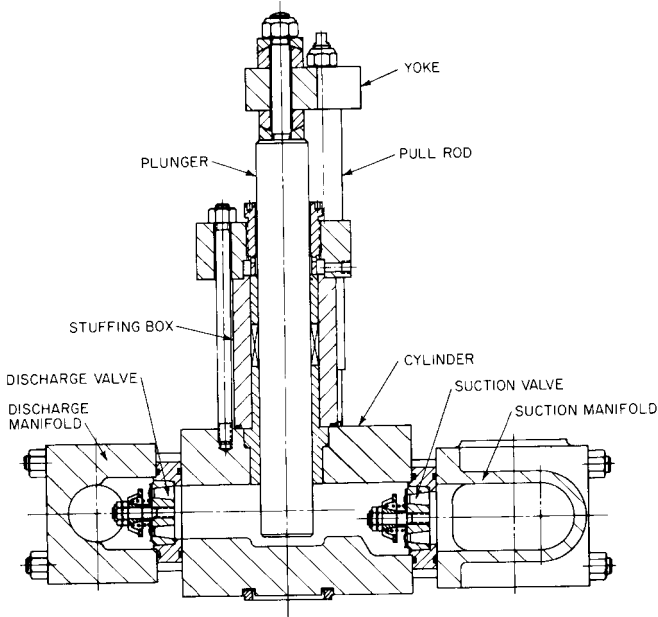


FIGURE 2 A liquid-end vertical pump (Flowserve Corporation)

TABLE 1 Maximum pressure in cast cylinders

Material	lb/in ² (bar)
Cast iron	2000 (137)
Aluminum bronze	2500 to 3000 (172 to 207)
Steel	3000 (207)
Ductile iron	3000 (207)

Forged cylinders have a 4:1 to 6:1 forging reduction on all sides of the cylinder to obtain a homogeneous internal structure. This type of cylinder requires heat treatment after forging to eliminate the stresses that occur during the forming operation. The pump designer must minimize the section thickness of the cylinder or residual stresses may remain after machining the internal bores.

The highest stresses occur at the intersection of the horizontal and vertical internal bores. Unlike a simple cylinder where the internal stresses are analyzed as a single hoop stress, the stress at the intersecting bores of a power pump cylinder are based on a double hoop stress. In Figures 3 and 4, the distances b and d are the internal diameters, a and c are the diameters to the nearest obstruction of the solid material of the cylinder, P is the developed pressure, and S is the resulting stress. Stress concentration factors at the intersecting bores can be omitted if the radius is not less than 0.25 in (1.5 mm). Above 3000 lb/in² (207 bar), the internal bores should have a surface finish of 63 rms minimum.

During each plunger cycle, the developed pressure goes from suction pressure to discharge pressure and back to suction. For a pump operating at 360 strokes per minute, over six million fatigue cycles will occur in less than 12 days. For this reason, very conservative allowable stress limits are used. Normally, these limits range from 10,000 to 25,000 lb/in² (69 to 172 Mpa), depending on the material of the cylinder, duty cycle, and liquid being

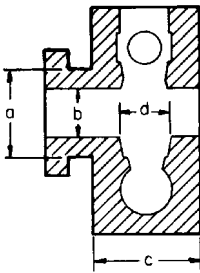
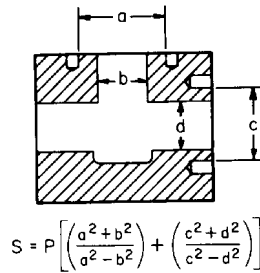


FIGURE 3 Stress dimensions for a horizontal liquid end (Flowsolve Corporation)



$$S = P \left[\left(\frac{a^2 + b^2}{a^2 - b^2} \right) + \left(\frac{c^2 + d^2}{c^2 - d^2} \right) \right]$$

FIGURE 4 Stress dimensions for a vertical liquid end (Flowsolve Corporation)

pumped. The allowable stress is a function of the fatigue stress of the material for the liquid being pumped and the life cycles required.

Due to improper suction system design, misapplications, or process upsets, instantaneous pressure in the cylinder may be much higher than design pressure. When the liquid contains entrained gas that can be released because of inadequate suction pressure, the resulting cavitation can cause instantaneous pressure four to five times the design pressure. This results in reducing the life of the cylinder and other liquid-end components, as well as damaging pressure pulsations in the suction and discharge piping systems.

For continuous duty pumps operating above 10,000 lb/in² (690 bar) of discharge pressure, special liquid-end designs have been developed. In Figure 5, the intersecting bores have been eliminated by arranging the suction and discharge valves on the same axis as the plunger. This means the stresses in the cylinder will be only half of those in a comparable T-block design.

Plungers The plunger transmits the force that develops the pressure. It is normally a solid construction of up to 5 in (127 mm) in diameter. Above that dimension, it may be made hollow to reduce its weight. Small-diameter plungers used for 6000 psi (414 bar) and above should be reviewed for possible buckling. Plunger speeds range from 150 to 350 fpm (46 to 107 m/s). The surface finish normally is between 14 and 20 rms. A finish below 8 rms should be avoided because excessive packing leakage may occur due to the inability of the packing to seal properly on the smooth surface.

Although some plungers are made of heat-treated or case-hardened steel, the most common are the hard-coated or solid ceramic. The hard coatings are normally flame sprayed powders of Colmonoy or tungsten carbide, or ceramic oxides. These can be applied over base materials of 1020 carbon steel or 316L stainless steel. Ceramic oxide is normally limited to 200°F (93°C) and is used for soft water, crude oil, mild acids, and mild alkalis.

The porosity and bond strength of coatings must be carefully evaluated for use with higher operating pressures. Under those conditions, the liquid may penetrate the pores of the coating and lift the coating off the base material. Ceramic plungers also have special requirements and limitations. This type of plunger is often constructed as a solid bar, closed-end tube of ceramic, bonded to a metal end cap or plug. A vent must be provided to allow the pressure inside the plunger to equal the atmospheric pressure or the ceramic-to-metal bond may fail or the plunger may explode. In addition to the normally fragile nature of any ceramic plunger, the solid ceramic plunger is susceptible to failure due to thermal shock.

Pistons Pistons are used for water pressures up to 2000 lb/in² (138 bar). For higher pressures, a plunger is usually used. Pistons are cast iron, bronze, or steel with reinforced elastomer sealing rings (see Figure 6). They are most frequently used in duplex, double-acting pumps. The latest trend is to use pistons in single-acting triplex pumps.

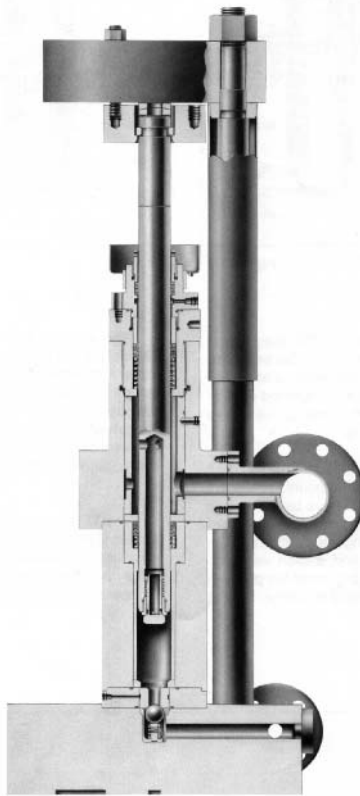


FIGURE 5 Special high-pressure liquid-end designs arrange the suction and discharge valves on the same axis as the plunger, thus eliminating intersecting bores (Flowserve Corporation).

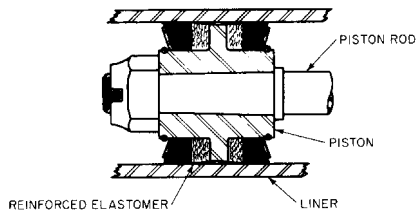


FIGURE 6 Elastomer face piston (FWI)

Stuffing Box The stuffing box assembly consists of a stuffing box, upper and lower bushings, packing, and a gland. For ease of maintenance, the stuffing box assembly is usually removable (see Figure 7).

The stuffing box bore is machined to a 63 rms finish to ensure packing sealing and life. A single hoop stress equation is used to determine the wall thickness with an allowable stress limit of 10,000 to 20,000 lb/in² (69 to 138 MPa).

The bushings that guide the plunger have a 63 rms finish with approximately 0.001 to 0.002 in (0.02 to 0.05 mm) of diametrical clearance per inch of the plunger diameter. The

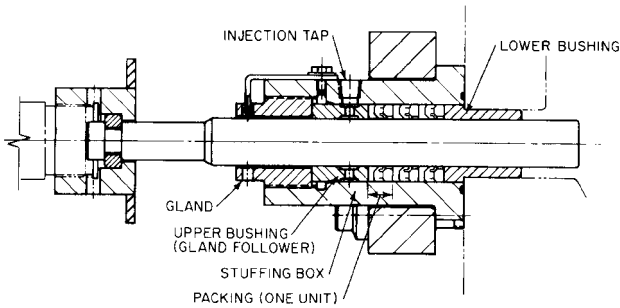


FIGURE 7 A stuffing box (Flowserve Corporation)

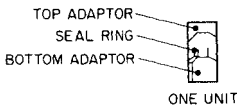


FIGURE 8 Chevron packing (Flowserve Corporation)

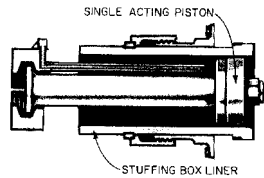


FIGURE 9 Single-acting piston stuffing box (Flowserve Corporation)

lower bushing is sometimes secured in an axial position to prevent movement of the packing. Bushings are made of bearing bronze, Ni-resist, or cast iron. Stainless steel can be used if its antigalling properties with the plunger coating can be verified.

Packing is either square cross-section, woven construction or molded V or chevron-shaped (see Figure 8). Some types use metal backup adapters. A packing set consists of top and bottom adapters and one or more packing sealing rings. A stuffing box can use two to five rings of packing, depending on the pressure and the fluid being pumped. Packing rings are usually made up of a number of composite materials selected for their strength, wear resistance, and lubricity, including neoprene, Teflon, cotton duct, and Kevlar.

Packing can be made self-adjusting by installing a spring between the bottom of the packing set and the lower bushing or bottom of the stuffing box. This arrangement eliminates overtightening the packing and enables a more uniform break-in. Packing can be lubricated through a grease fitting or by an auxiliary lubricator driven through a take-off on the crankshaft.

For chemical or slurry service, a lower injection ring is used for flushing. This prevents concentrated pumped fluid from impinging directly on the packing. This injection can be continuous or synchronized to inject only on the suction stroke. Flush glands at the outboard end of the stuffing box are employed when a toxic vapor is present or when the leakage may flash.

The stuffing box for the piston rod of a double-acting pump is similar in construction to that of a stuffing box for a plunger. The primary difference is that it must seal against pressure being developed while the rod moves back through the packing. Single-acting pistons do not employ a stuffing box. Leakage past the piston-sealing rings goes into the frame extension to mix with the continuously circulating lubricant.

Cylinder Liner The cylinder wear liner (refer to Figure 6 and see Figure 9) is usually of Ni-resist material. Its length is slightly longer than the stroke of the pump to enable an assembly entrance taper of the piston into the liner. In double-acting pumps, the liner has packing to prevent leakage from the high-pressure side to the low pressure 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 bore of the liner is approximately 16 rms.

Valves Many types of valve designs exist. Which type is used depends on the application. The main parts of a valve assembly are the seat and sealing member, usually a disc, ball, or plate. The plate movement is controlled by a spring or retainer. The seat usually uses a taper where it fits into the cylinder or manifold. The taper not only gives a positive fit but permits easy replacement of the seat.

Some pumps use the same size suction and discharge valve for interchangeability (refer to Figure 2). Some use larger suction valves than discharge valves for improved NPSH reasons. Others use larger discharge valves than suction valves because the cylinder configuration requires the suction valve to be removed through the location of the discharge valve (refer to Figure 1). Because of space considerations, valves are sometimes used in clusters on each side of the plunger to obtain the required total valve area that reduces valve velocity.

Table 2 shows seat and plate hardness for some valve materials. Seats and plates made of 316 stainless steel material are usually chrome-plated or flame-sprayed to give them the desired surface hardness.

The flow area of the valve must be large enough to prevent significant pressure drop or restriction to flow. Normally, suction valves are sized for 6 to 8 ft/s velocity, and the discharge valves are sized for 8 to 12 ft/s.

The valve springs must be made of corrosion-resistant material and designed to withstand high-cycle fatigue stresses. Under no situation should the valve operation enable the spring to go to a solid stack height or wire-to-wire condition. The ends of the spring must be ground flat and square with a maximum perpendicularity of 3 degrees.

Manifolds These are the chambers where the liquid is disbursed or collected for distribution before or after passing through the cylinder. In horizontal pumps and some vertical pumps, the manifolds are cast or machined integral with the fluid cylinder. Most vertical pumps have the suction and discharge manifolds separate from the cylinder (refer to Figure 2).

Suction manifolds are designed to eliminate air pockets from the flange to the valve entrance (see Figure 10). Separate suction manifolds are cast iron, cast bronze, or fabricated steel. Discharge manifolds are steel forgings or fabricated steel. The manifolds have a minimum deflection to prevent gasket shifts when subjected to maximum operating pressures.

The velocity through the manifolds of a clean liquid is 3 to 5 ft/s (0.9 to 1.5 m/s) at the suction and 6 to 16 ft/s (1.8 to 4.9 m/s) at the discharge. Suction and discharge manifold velocities in a slurry service are 6 to 10 ft/s (1.8 to 3 m/s). Slurry services have a minimum velocity of 6 ft/s (1.8 m/s) to prevent the heavier slurry particles from falling out of solution.

In USCS units, V (ft/s) = pump gpm \times 0.321/cross-sectional area of the manifold, in²

In SI units, V (m/s) = pump m³/h \times 277.8/cross-sectional area of the manifold, mm²

TABLE 2 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

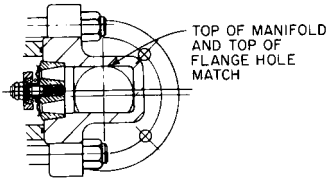


FIGURE 10 A suction manifold construction to eliminate air pockets (Flowserve Corporation)

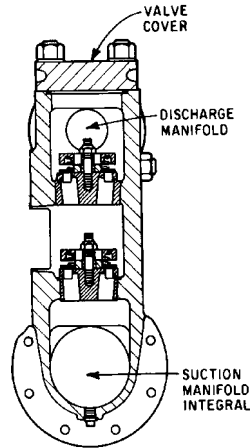


FIGURE 11 An integral suction and discharge manifold (Gardner-Denver)

A water hammer creates an additional pressure (which is added to the rated pump pressure), as does hydraulic shock loading. The discharge manifold rating is then made equal to or greater than the sum of these pressures.

Valve Covers Valve covers are used to provide accessibility to the valves without disturbing the cylinder, manifold, or related piping (see Figures 11 and 12).

Plunger Covers and/or Cylinder Heads These are used on horizontal pumps to provide accessibility to the plunger, piston, and cylinder liner (refer to Figures 11 and 12 and see Figure 13).

POWER END

The power end contains the crankshaft, connecting rods, crossheads, pony rod, bearings, and frame (see Figures 14 and 15). Basic designs are horizontal and vertical with sleeve or antifriction bearings. Some have integral drive gears.

Frame The frame absorbs the plunger load and torque. Vertical pumps with outboard packed stuffing boxes (refer to Figure 2) have the frame in compression (refer to Figure 14). With horizontal single-acting pumps, the frame is in tension (refer to Figure 15). Frames are usually close-grain cast iron with relatively thick cross-sections. This combination provides superior dampening for the normal low-frequency cyclic loading. Some intermittent duty pumps, specifically designed for mobile services, are constructed of lighter weight, fabricated steel plate.

The frame is usually vented to the atmosphere. However, when the atmosphere is detrimental to the working parts of the frame, such as an ammonia attack on bronze bearings, the frame can be purged continuously with nitrogen gas.

Crankshaft The crankshaft (see Figures 16, 17, and 18) varies in construction, depending on the design and power output of the pump. In horizontal pumps, the crankshafts are usually of nodular iron or cast steel. Vertical pumps use forged steel or machined billet crankshafts. Because the crankshafts have a low mass and operate at relatively low

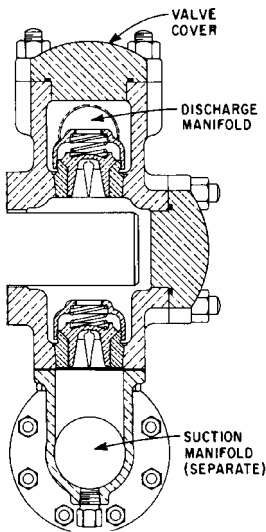


FIGURE 12 Separate suction with an integral discharge manifold (Gardner-Denver)

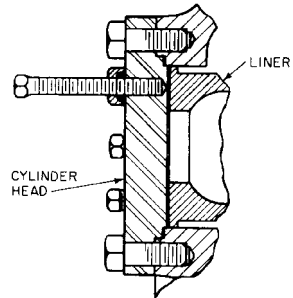


FIGURE 13 A plunger cover (FWI)

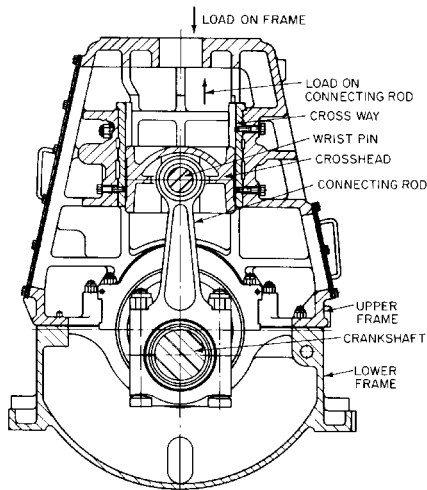


FIGURE 14 A vertical pump power end (Flowserve Corporation)

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

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

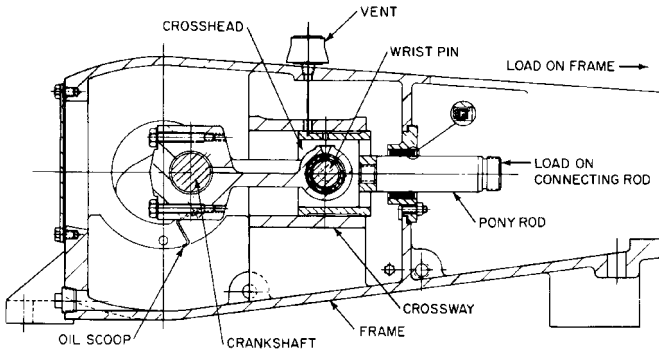


FIGURE 15 A horizontal pump power end (Flowserve Corporation)

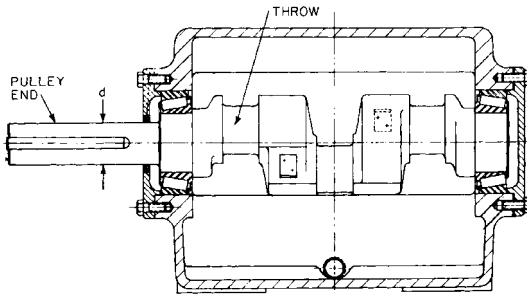


FIGURE 16 A cast crankshaft (Flowserve Corporation)

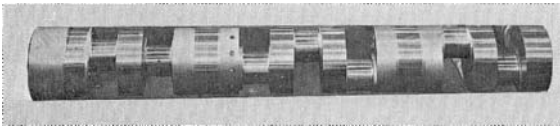


FIGURE 17 A crankshaft machined from a billet (Flowserve Corporation)

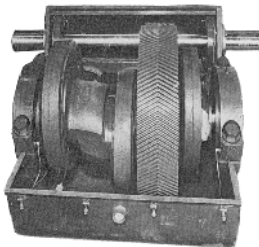
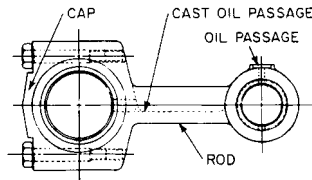


FIGURE 18 A cast crankshaft with integral gear (Continental-EMSCO)

TABLE 3 Order of Pressure Build-up or Firing Order

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

**FIGURE 19** The connecting rod (Flowsolve Corporation)

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

Many horizontal pumps under 250 hp (186 kW) are driven by means of a sheave or pulley and belt arrangement. Increasing the diameter of the pulley for a given horsepower will increase the stress in the crankshaft. The maximum allowable crankshaft stress will often limit the maximum diameter of the pulley that can be used. Stress levels of 2000 to 3000 lb/in² (14 to 20.7 MPa) are common, where d is the pulley end diameter of the crankshaft in inches (mm) and n is the pump speed in revolutions per minute.

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

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

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

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

Wrist Pin Located in the crosshead, the wrist pin transforms the oscillating motion of the connecting rod to a reciprocating motion. The maximum stress in the wrist pin from deflection should not exceed 10,000 lb/in² (69 MPa):

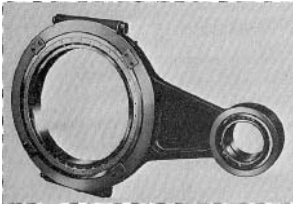


FIGURE 20 An eccentric strap (Gardner-Denver)

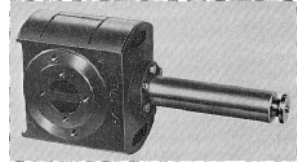


FIGURE 21 A crosshead (Gardner-Denver)

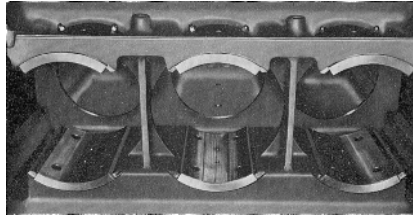


FIGURE 22 Crossways (Gardner-Denver)

In USCS units,

$$S = \frac{PL \times \ell}{8 \times 0.098d^3}$$

In SI units,

$$S = 1.25 \frac{PL \times \ell}{d^3}$$

where PL = plunger load, lb (N)

ℓ = length under load, in (mm)

d = diameter of pin, in (mm)

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

Crosshead The crosshead (see Figure 21) moves in a reciprocating motion and transfers the plunger load to the wrist pin. The crosshead is designed to absorb the side or radial load from the plunger as the crosshead moves linearly on the crossway. The side load is approximately 25% of the plunger load. For cast-iron crossheads, the allowable bearing load is 80 to 125 lb/in² (551 to 862 kPa). Crossheads are grooved for oil lubrication and have a 63 rms surface finish. Crossheads are piston types (full round) or partial-contact types. The piston type should be open or vented to prevent air compression at the end of the stroke.

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

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

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

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

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

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

SLEEVE BEARINGS

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

Wrist Pin Bearings These bearings are normally the highest loaded bearings in the pump. They can only perform an oscillating motion and therefore cannot develop the dynamic oil film common to fully rotational bearings. In single-acting pumps with low suction pressure, adequate reversals in loading take place on the bearing to permit replenishment of the oil film. High suction pressure in a horizontal pump increases the reverse loading. In vertical pumps, high suction pressure can produce a condition of no reverse loading, and in this case, special bearing materials, grooving, and pump derating may be required. Increasing oil pressure by itself will not solve this problem. Allowable projected area loading is 1200 to 1500 lb/in² (8.27 to 10.3 MPa) with bearing bronze.

Crank Pin Bearings The crank pin bearing is a rotating split bearing and develops a better oil film than the wrist pin bearing. It is clamped between the connecting rod and cap. The bearing is a bronze-backed babbitt metal or a tri-metal automotive type bearing. The allowable projected area load is 1200 to 1600 lb/in² (8.27 to 11.08 MPa).

Main Bearings The main bearings absorb the plunger load and gear load. The total plunger load varies during each revolution of the crankshaft. The triplex main bearings receive the greatest variations in loading because the crankshaft has the greatest relative span between bearings. Sleeve bearings are flanged to locate them in an axial position, and the flange absorbs residual axial thrust. In long-stroke vertical pumps, a main bearing can be found between every connecting rod. Split bearings are a bronze-backed babbitt metal with an allowable loaded area of 750 lb/in² (5.1 MPa).

ROLLING ELEMENT BEARINGS

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

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

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

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

LUBRICATION

Non-detergent SAE 30 to 50 oil is used for bearing lubrication. In splash lubrication, the cheeks of the crankshaft or oil scoops (refer to Figure 15) throw oil by centrifugal force against the frame wall. This oil is then distributed by gravity to the crosshead, wrist pin, and crank pin. At low speeds, there is not enough centrifugal force for proper oil distribution, so a partial force feed is then employed to maintain proper lubrication. Force-feed lubrication requires 0.5 to 1 gpm (1.9 to 3.8 l/m) per bearing with an oil pressure of 28 to 40 lb/in² (1.7 to 2.8 bar).

APPLICATIONS

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

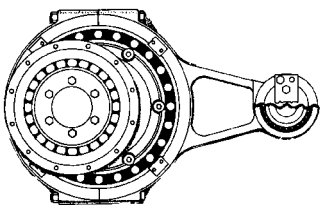


FIGURE 23 The main bearing mounted in a bearing holder (Gardner-Denver)

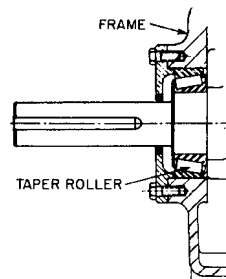


FIGURE 24 The main bearing mounted directly in the frame (Flowsolve Corporation)

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

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

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

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

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

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

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

RELIEF VALVES

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

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

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

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

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

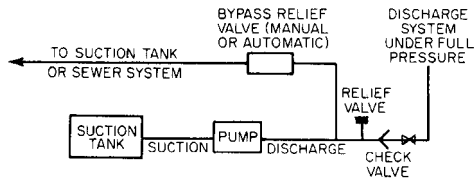


FIGURE 25 A bypass relief system (Flowserve Corporation)

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

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

PULSATION DAMPENERS

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

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