C • H • A • P • T • E • R • 3

DISPLACEMENT PUMPS

SECTION 3.1
POWER PUMP THEORY

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A power pump is a positive displacement machine consisting of one or more cylinders, each containing a piston or plunger. The pistons or plungers are driven through slider-crank mechanisms and a crankshaft from an external source. The capacity of a given pump is governed by the rotational speed of the crankshaft.

Unlike a centrifugal pump, a power pump does not develop pressure; it only produces a flow of fluid. The downstream process or piping system produces a resistance to this flow, thereby generating pressure in the piping system and discharge portion of the pump. The flow fluctuates at a rate proportional to the pump speed and number of cylinders. The amplitude of the fluctuations is a function of the number of cylinders. In general, the greater the number of cylinders, the lower the amplitude of the flow variations at a specific rpm.

All power pumps are capable of operating over a wide range of speeds, thereby making it possible to produce a variable capacity when coupled to a variable speed drive. Each pump has maximum suction and discharge pressure limits that, when combined with its maximum speed, determine the pump's power rating. The pump can be applied to power conditions that are less than its maximum rating but at a slight decrease in mechanical efficiency.

The power pump is a positive displacement device. When operating, it will continue to deliver flow independent of the pressure in the discharge piping system. Unlike a centrifugal pump, a power pump will not "deadhead" or "go back on its curve" in response to increasing discharge pressure. When this pressure exceeds the design limits of the pump, mechanical failure—often catastrophic—will result. For this reason, all piping systems incorporating power pumps must have discharge pressure relief devices to limit the pressure in the piping system and avoid pump failure. These devices must be located between the discharge connection on the pump and the first isolation valve in the piping system.

This section has been organized to provide sufficient power pump theory to properly select a pump for most applications. It is recognized that the proper pump metallurgy for the pumped fluid must be used, but due to the vast number of possible fluids, selection of pump metallurgy is outside the scope of this section.

The subject of Net Positive Suction Head (NPSH) is mentioned in several places in this section and is covered in more detail in Section 3.4, "Displacement Pump Performance, Instrumentation, and Diagnostics." For basic power pump selection, it is only necessary to understand that the NPSH available from the suction system must be sufficiently above the NPSH required by the pump to operate properly.

SELECTION THEORY

Power Brake horsepower (bhp) is a function of a pump's capacity, differential pressure, and mechanical efficiency. It is an essential criterion for selecting the drive components but is not valuable for pump selection. A large pump operating well below its design rating can meet the same horsepower requirements as a smaller pump running at a higher speed. Unless the application requires a derated pump, it is usually more economical to select a pump at the upper end of its design rating.

The brake horsepower for the pump is

In USCS units $$bhp = Q \times Ptd/1714 \times ME$$

In SI units $$kW = Q \times Ptd/36 \times ME$$

where Q = delivered capacity, gpm (m^3/h)
Ptd = differential pressure (discharge − suction), lb/in^2 (bar)
ME = mechanical efficiency, %

NOTE: One bar equals 10^5 Pa.

Capacity The capacity Q is the total volume of fluid delivered per unit of time. This fluid includes liquid, entrained gases, and solids at the specified conditions.

Displacement Displacement D, gpm (m^3/h), is the calculated capacity of the pump with no slip losses. For single-acting plunger or piston pumps, this is

In USCS units $$D = A \times m \times n \times s/231$$

In SI units $$D = A \times m \times n \times s \times 6 \times 10^{-5}$$

where A = cross-sectional area of plunger or piston, in^2 (mm^2)
m = number of plungers or pistons
n = rpm of pump
s = stroke of pump, in (mm) (half the linear distance the plunger moves in one revolution)
231 = in^3/gal

For double-acting plunger or piston pumps, this is

In USCS units $$D = (2A - a)m \times n \times s/231$$

In SI units $$D = (2A - a)m \times n \times s \times 6 \times 10^{-5}$$

where a is the cross-sectional area of the piston rod, in^2 (mm^2).

Pressure The pressure Ptd used to determine brake horsepower is the differential pressure or discharge pressure minus the suction pressure. In most applications, the suction pressure is small relative to the discharge pressure. However, when pumping some compressible liquids, such as methane and propane, the suction pressure may be 20 to 30% of the discharge pressure. For accurate brake horsepower calculations, always include the suction pressure. Figure 1 shows a typical performance curve for a power pump.

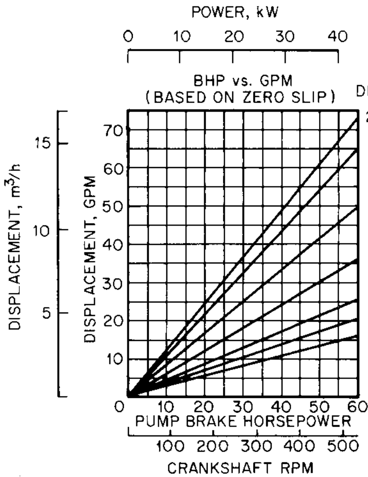


FIGURE 1 Typical power pump performance: brake horsepower versus gallons per minute (kilowatts versus m³h) based on zero slip (Flowserve Corporation)

TABLE 1 Slip in a pump with a plate valve

Centistokes	100	1,000	2,000	6,000	10,000	12,000
Slip, %	8	8.5	9.5	20	41	61

TABLE 2 Slip as a function of pump speed and pressure

	Slip, %		
Pressure, lb/in² (bar)	At 440 rpm	At 390 rpm	At 365 rpm
4000 (275)	11	22	34
3000 (207)	9	20	31
2000 (138)	7	18	30
1000 (69)	7	15	27.5

Slip Slip (S) is the loss of capacity due to internal and external pump leakage. External leakage occurs primarily through the stuffing box via the packing. Internal leakage is primarily the backflow past the suction and discharge valves. Backflow occurs when a valve remains open for a fraction of a second as the plunger or piston reverses direction. A small amount of leakage may occur across the piston in a double acting pump from the high pressure side to the low pressure side. Slip is expressed as a percentage loss of the suction capacity and is typically 1% to 4%:

$$S = B + V + L$$

where S = Slip, %

B = Leakage through the stuffing box

V = Backflow across the valves

L = Internal leakages

Fluid viscosity, pump speed, and discharge pressure can all have an effect on slip, which is shown in Tables 1 and 2.

Mechanical Efficiency The mechanical efficiency of a power pump is

In USCS units $ME = \text{power out/power in} = P_{out}/P_{in} = Q \times Pdt/1714 \times P_{in}$

In SI units $ME = \text{power out/power in} = P_{out}/P_{in} = Q \times Pdt/36 \times P_{in}$

where P_{in} is the input power from the driver, bhp (kW).

The mechanical efficiency of a power pump is the sum of all the frictional losses in the fluid and power ends. These include the plungers and packing, the crossheads, the rod seals, and the bearings. The efficiency of a single acting pump often exceeds 90%, while a double-acting piston pump will be 88% due to the additional piston and rod seals. If the pump is equipped with internal gearing, an additional 2% loss is common.

Most power pumps are designed to accept a range of plunger or piston sizes. When the larger plungers are used, the increased diameter of the packing/seals and plunger/liners will result in higher frictional losses than with smaller components. As a rule, doubling the plunger/piston diameter will decrease the mechanical efficiency by 8%. Mechanical efficiency is also affected by speed and, to a lesser extent, by developed pressure, as indicated in Tables 3 and 4.

Speed Pump speed, or, more correctly, stroke rate, is one of the most critical selection criteria for power pumps. The rotating and reciprocating parts of the power end, as designated, are often capable of speeds twice that of the actual pump rating. The maximum pump speed is determined by the design of the fluid end, the hydraulic capability of the anticipated suction system, and the required life of the plungers, packing, and valves. Most power pump standards limit the plunger speed from 140 to 280 ft/min (0.71 to 1.42 m/s). The plunger speed is

In USCS units $Sp = s \times n/6$

In SI units $Sp = s \times n/30{,}000$

where Sp = plunger or piston speed, ft/min (m/s).

s = stroke of the pump, in (mm) (half of the linear distance the plunger or piston moves in one revolution)

n = rpm of pump

All pumps have a minimum speed limit, usually determined by a decrease in the adequate lubrication to the bearings in the power end.

Volumetric Efficiency Volumetric efficiency (VE) is the ratio of the discharge volume to the suction volume, expressed as a percentage, plus the slip. It is proportional to the ratio r and the developed pressure where r is the ratio of the internal volume of fluid between valves when the plunger or piston is at the top of the peak of its back stroke (C + D) to the plunger or piston displacement (D) (see Figure 2).

TABLE 3 The effect of speed on mechanical efficiency at a constant developed pressure

% of full speed	44	50	73	100
***ME*, %**	93.3	92.5	92.5	92.5

TABLE 4 The effect of pressure on mechanical efficiency at constant speed

% of full-load developed pressure	20	40	60	80	100
***ME*, %**	82	88	90.5	92	92.5

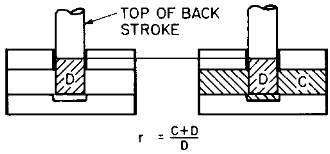


FIGURE 2 The ratio *r* (Flowserve Corporation)

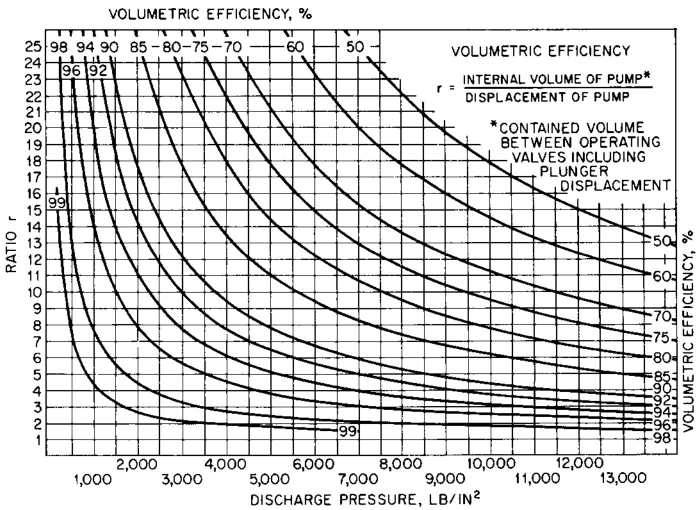


FIGURE 3 Volumetric efficiency (Flowserve Corporation) (lb/in^2 × 0.69 = bar)

Since the discharge volume cannot be readily measured at discharge pressure, it is taken at suction pressure. Taking the discharge volume at the suction pressure results in a higher volumetric efficiency than using the calculated discharge volume at discharge pressure because of fluid compressibility. Compressibility becomes important when pumping water or other liquids over 6000 lb/in^2 (414 bar), and it should be taken into consideration when determining the actual delivered capacity into the discharge system.

Figure 3 shows the approximate volumetric efficiency for water (not including slip).

Based on the expansion back to suction capacity,

$$VE = (1 - Ptd \times \beta \times r)/(1 - Ptd \times \beta) - S$$

Based on discharge capacity,

$$VE = (1 - Ptd \times \beta \times r) - S$$

where β is the compressibility factor of the liquid being pumped. Figure 4 shows approximate values of β for various liquids.

When the compressibility factor is not known, but the suction and discharge density can be determined in pounds per cubic foot, the following equations can be used for calculating *VE*:

Based on suction capacity,

$$VE = r - (p_d/p_s)(r - 1) - S$$

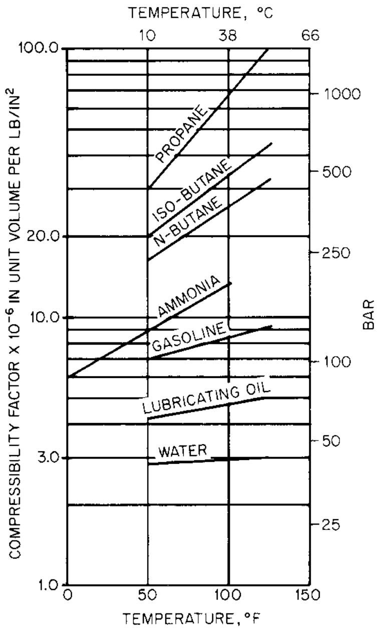


FIGURE 4 Compressibility factor (Flowserve Corporation) (bar × 14.5 = lb/in²)

Based on discharge capacity,

$$VE = 1 - r(1 - p_s/p_d) - S$$

where p_s = suction density
p_d = discharge density
S = slip

Torque The average torque required by a power pump is independent of the pump speed, assuming the suction and discharge pressures are held constant. This means that the pump is a "constant torque" device and, unlike a centrifugal pump, will have a flat speed-torque curve. The torque required at the input shaft is as follows:

In USCS units $M = p \times 5250/n$

In SI units $M = p \times 9.549/n$

where M = pump torque, lb-ft (N · m)
n = speed, rpm
p = power, bhp (kW)

Break-away torque, or start-up torque, is the torque required to initiate the motion of the pump to enable it to accelerate to a constant speed. If the pump is started against an open bypass to pump suction, the torque requirement is 25% of the running torque. If the pump is started against full discharge pressure, the torque requirement is 125% of the running torque. When selecting a drive system, the break-away torque of the pump must be considered. Special "high torque" motors are often required if the pump is to be started under the load.

TABLE 5 The pulses per revolution and pulse amplitude for various pump types

Pump Type	No. of Cylinders	Pulses per Rev.	Pulse Amplitude
Duplex, double-acting	4	8	+24% − 22%
Triplex, single-acting	3	6	+ 6% − 17%
Quintuplex, single-acting	5	10	+ 2% − 5%
Septuplex, single-acting	7	14	+1.2% − 2.6%
Nonoplex, single-acting	9	18	+0.6% − 1.5%

TABLE 6 Guidelines for acceptable plunger speeds

Fluid	Plunger Speed, ft/min (m/s)
Cold water	315–354 (1.6–1.8)
Hot water 140–194 °F (60–90 °C)	217–256 (1.1–1.3)
Hot water over 194 °F (90 °C)	157–217 (0.8–1.1)
Salt water	236–276 (1.2–1.4)
Cold oil	315–354 (1.6–1.8)
Hot oil	256–295 (1.3–1.5)
Crude oil	217–276 (1.3–1.5)
Liquid ammonia	197–236 (1.0–1.2)
Carbamate	138–158 (0.7–0.8)
Slurry	158–197 (0.8–1.0)
Fatty acid	236–295 (1.2–1.5)
Light hydrocarbons	177–236 (0.9–1.2)
Glycol	256–295 (1.3–1.5)
High viscosity liquids	138–197 (0.7–1.0)

If a variable frequency drive (VFD) is used for pump speed/capacity control, torque fluctuations or pulsations may require the drive manufacturer to provide special electronic dampening circuits. The normal pulsating flow of the pump causes the input power to fluctuate at approximately the same magnitude as the flow pulses. The frequency of these pulses per revolution of the crankshaft is twice the number of cylinders (see Table 5).

If the pump has internal gearing, or if the pump is being driven by an external gear reducer, the torque pulsation values listed should be divided by the gear ratio.

Derating Power pumps can be derated in order to achieve acceptable performance or part life in a variety of special applications. The most common derating is pump speed. Experience has shown that the life of fluid-end expendable parts, such as plungers, packing, and valves, can be extended if the pump speed/plunger velocity is reduced when pumping certain fluids. Some guidelines for acceptable plunger speeds are shown in Table 6.

Pumping high-viscosity fluids may also require the speed of a power pump to be derated. Some attempts have been made to establish standards for speed derating as a function of the fluid viscosity. Unfortunately, these methods have been based on power pump designs that were established for water or similar low-viscosity services. If a pump has been designed specifically for a high-viscosity service, only moderate speed derating may be required. Generally, pumps designed with large internal fluid passages, low-valve velocities, and larger plungers or pistons will need less derating.

Power pumps should be derated for a rod load if system or process “upsets” cannot be avoided. These are usually applications where the gas content of the pumped fluid can lead to abnormal pressure pulsations or cavitations in the fluid cylinder.

FLUID END THEORY

Pumping Cycle Unlike the relatively smooth, continuous flow of fluid through a centrifugal pump, the flow of fluid through a power pump occurs in a transitory dynamic condition called a *pumping cycle*. The event that initiates this cycle is the linear movement of the plunger or piston. In Figure 5, r is the radius of the crank in feet (meters), L is the length of the connecting rod in feet (meters), C equals L / r, and ω equals $(2\pi / 60) \times$ rpm. Thus, X , the linear movement of the piston or plunger is

$$X = r\left[1 - \cos\theta + L\left(1 - \sqrt{1 - \frac{r^2}{L^2}(\sin\theta)^2}\right)\right]$$

As the plunger (or piston) withdraws from the fluid cylinder or pumping chamber, the volume of the cylinder increases. The pressure in the cylinder decreases in response to the increased volume. Since most of the fluids handled by power pumps are relatively incompressible, very little plunger movement is required to cause a pressure drop. When the cylinder pressure drops sufficiently below suction pressure, the differential pressure begins to open the suction valve. The valve opens gradually and smoothly at the start of the suction stroke because the velocity and acceleration of the plunger are small.

Fluid flows through the suction valve assembly, following the plunger and filling the cylinder. As the plunger decelerates at the end of the suction stroke, the suction valve gradually returns to its seat. Ideally, the suction valve is completely closed as the plunger comes to a stop.

The motion of the slider-crank mechanism causes the plunger to reverse direction and start its discharge stroke. The fluid trapped in the fluid cylinder is compressed until the cylinder pressure exceeds the discharge pressure by an amount sufficient to begin to open the discharge valve. As with the suction valve, the discharge valve continues to open until it reaches its travel limit or until the velocity of fluid through the valve becomes constant. As the plunger decelerates, the valve moves back toward its seat. Again, ideally, the discharge valve closes when the motion of the plunger stops.

The number of pumping cycles in a single revolution of the crankshaft is the same as the number of cylinders in the pump. Every cylinder will "pump" in a sequence determined by the "firing order" of the crankshaft. The cylinders are arranged in parallel, with each one discharging into a common discharge manifold. In industry terms, the pump is usually identified by the number of plungers or pistons on the crankshaft. They are the same for single- or double-acting pumps (see Table 7).

Pulsations The pulsating characteristics of the fluid flowing into and out of power pumps are significantly influenced by the number of plungers or pistons. Discharge flow pulsations are the most critical because of the high energy potential generated when the system resistance reacts with the flow to create pressure. Since the magnitude of the discharge pulsation is mostly affected by the number of cylinders, increasing the number of cylinders will reduce the flow pulsations.

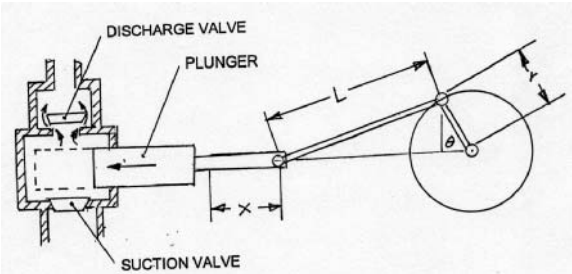


FIGURE 5 Slider-crank mechanism

TABLE 7 Industry terms for the number of plungers or pistons

Number	Term
1	Simplex
2	Duplex
3	Triplex
4	Quadruplex
5	Quintuplex
6	Sextuplex
7	Septuplex
9	Nonuplex

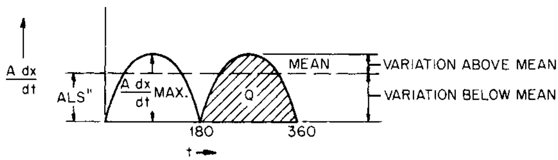


FIGURE 6 The discharge rate for a single double-acting pump (Flowserve Corporation)

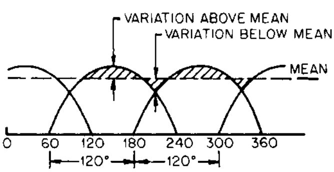


FIGURE 7 The discharge rate for a triplex single acting pump (Flowserve Corporation)

The acceleration and velocity of the plunger or piston will determine the rate that the fluid is discharged from the cylinder. The approximate velocity of the plunger or piston is

$$S = \frac{dx}{dt} = r\left(\sin\theta + \frac{\sin 2\theta}{2C}\right)\omega$$

The approximate acceleration of the plunger or piston is

$$Cp = \frac{d^2x}{dt^2} = r\left(\cos\theta + \frac{\cos 2\theta}{C}\right)\omega^2$$

The momentary rate of discharge capacity is the cross-sectional area A of the plunger or piston times velocity:

For single-acting plunger or piston $\quad A = 0.785 \times D^2$

For double-acting piston $\quad A - a = 0.785\,(D^2 - d^2)$

The total discharge Q is equal to ALS', where S' is the number of effective strokes in a given time. The quantity Q is the area of the curve, the mean height of which is

$$Total\ Q/t = ALS'/t = ALS''$$

where S'' is the number of strokes per second and t is time in seconds.

Figures 6 and 7 show discharge rates for two types of pumps. Tables 8 and 9 show how variations from the mean are related to the change in the number of plungers with a C of

TABLE 8 The effect of the number of plungers on variations in capacity from the mean (*C* is approx. 6:1)

Type	Number of Plungers	% Above Mean	% Below Mean	Total %	Plunger Phase
Duplex	2	24	22	46	180 deg.
Triplex	3	6	17	23	120 deg.
Quadruplex	4	11	22	33	90 deg.
Quintaplex	5	2	5	7	72 deg.
Sextuplex	6	5	9	14	60 deg.
Septuplex	7	1	3	4	51.5 deg.
Nonuplex	9	1	2	3	40 deg.

TABLE 9 The effect of change in *C* on variations in capacity from the mean for a triplex pump

C	% Above Mean	% Below Mean	Total %
4:1	8.2	20.0	28.2
5:1	7.6	17.6	25.2
6:1	6.9	16.1	23.0
7:1	6.4	15.2	21.6

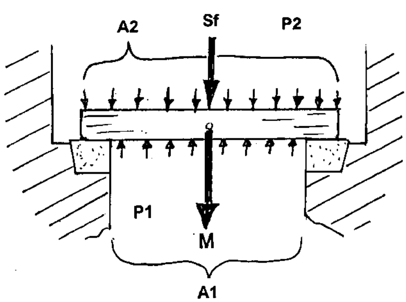


FIGURE 8 Force balance of a typical valve

approximately 6:1. This shows that pumps with an even number of cylinders have a higher flow variation than pumps with an odd number of cylinders. Table 9 shows the variations in a triplex with changes in C = L/R.

Valves A pump valve in its simplest form is a free-moving plug that is opened when the force of the liquid below the valve exceeds the force of the liquid above it. When the force above the valve becomes greater than the lower, the plug closes and forms an effective seal to fluid "backflow" and pressure loss. If the valve does not perform this function efficiently, the performance of the pump can be degraded to the point where no flow is produced.

A valve will be in equilibrium when the forces above and below the valve are balanced (see Figure 8):

$$\text{Equilibrium} = P_1 \times A_1 = P_2 \times A_2 + S_f + M$$

where P = the pressure below the valve

A = the area below the valve exposed to P

P = the pressure above the valve
A = the area above the valve exposed to P
S_f = the force of the valve spring (if any)
M = the mass of the valve and half the mass of the spring

The significance of this simple equation becomes apparent when you remember that the valve is operating in a dynamic environment of constantly changing pressures in a time frame that is measured in tenths of a second. Add to that the requirement that the valve must pass a volume of fluid in each opening cycle with minimal pressure drop, and it becomes clear that the pump valve is the most critical component in the fluid end in terms of pump operability.

The pump designer will size the valves to provide a flow area, called the *spill area*, that prevents pre-established velocity limits from being exceeded. The spill area of various valve types is given in Table 10. The velocity of the fluid flowing through the spill area is called the spill velocity, V, shown in Table 11:

TABLE 10 Types of valves and their applications (Flowserve Corporation)

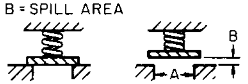
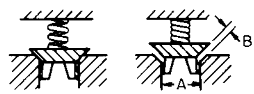
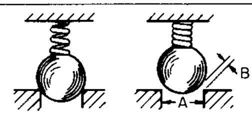
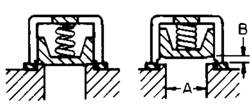
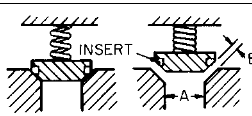
TYPE	SKETCH A = SEAT AREA B = SPILL AREA	PRESSURE, PSI (BAR)	APPLICATION
PLATE		5,000 (345)	CLEAN FLUID. PLATE IS METAL OR PLASTIC
WING		10,000 (690)	CLEAN FLUIDS. CHEMICALS
BALL		30,000 (2,069)	FLUIDS WITH PARTICLES. CLEAR, CLEAN FLUID AT HIGH PRESSURE. BALL IS CHROME PLATED
PLUG		6,000 (414)	CHEMICALS
SLURRY		2,500 (172)	MUD, SLURRY. POT DIMENSIONS TO API-12. POLYURETHANE OR BUNA-N INSERT

TABLE 11 Valve spill velocities

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

In USCS units $V\,(\text{ft/sec}) = \text{gpm through the valve} \times 0.642/\text{spill area of the valve, in}^2$

In SI units $V\,(\text{m/s}) = \text{m}^3\text{/h through valve} \times 555.6/\text{spill area of the valve, mm}^2$

The quantity 0.642 (555.6) is used because all the liquid passes through the valve in half the stroke.

Valve Dynamics Valve dynamics is the mechanical response of the valve to the changes of pressure across the valve. Using a suction valve as an example, the valve starts its cycle when the valve is closed and the plunger is at the start of its suction stroke (maximum insertion into the cylinder). As the plunger starts withdrawing from the cylinder, the internal volume in the cylinder starts to increase. This increasing volume results in decreasing cylinder pressure that, in turn, hydraulically unbalances the valve and the valve begins to open. There is usually a slight lag in the valve opening versus the start of the plunger motion of about 5 to 20 degrees of crankshaft rotation. Traditionally, the opening valve lag is attributed solely to the inertia required to set the valve in motion and the preload of the valve spring. In the last few years, the theory of *valve stiction* has been proposed as an additional cause of valve opening lag.

Valve stiction, as the name implies, can be an additional force to overcome when the valve is trying to lift off its seat. One version of the theory is that a cohesive force exists between the fluid molecules and the sealing surfaces of the valve and seat. This can be demonstrated by trying to separate two wet, highly machined plates. The second stiction theory focuses on possible fluid dynamic conditions that may occur as fluid starts to flow across the valve seat. Valve seats do not have "knife edge" sealing surfaces; they have a width that distributes the stresses in the valve and seat. The stiction theory postulates that the flow of fluid from the smaller area at the center of the seat to the larger area around the seat causes a pressure drop across the seat. The pressure drop is the result of the radial divergence of the fluid and is strong enough to momentarily prevent the valve from opening. Extreme cases have even resulted in a circumferential ring of cavitation located at the center of the sealing areas of the seat and valve.

Field testing valves with special grooves and radial slots in the sealing face has proven successful in reducing stiction and opening valve lag in some cases. Narrowing the sealing faces may also prove effective, although part life may decrease due to increased stress and decreased wear area. Additional computer modeling and testing is required before the stiction theory can be completely validated.

As the valve continues to open, the spill area increases proportionally. It must be remembered that fluid is now flowing through the valve at an increasing rate while trying to fill the expanding cylinder volume created by the plunger withdrawal. Ideally, the motion of the valve will exactly correspond to the flow rate of the fluid through the valve, and the valve follows a smooth trajectory to a full open position (see Figure 9).

This idyllic situation only occurs in very slow running pumps that are fitted with oversized valves. In modern higher speed pumps, the valve often accelerates at a rate fast enough to create a spill area greater than that required to maintain a constant flow velocity. At this point, the valve motion momentarily stops, and the valve may even start to close before the fluid flow once again matches the spill area. The valve then returns to the smooth trajectory until the maximum valve lift occurs.

After the plunger passes the mid point of its stroke, it starts decelerating. The flow of fluid through the valve is high enough to start to build pressure in the cylinder and unbalance the valve and start the valve closing. The valve spring has reached its maximum compression and, if properly designed, will help accelerate the valve back toward the seat. When the plunger reaches the end of the suction stroke, the valve should also be closing. The amount of crankshaft rotation that occurs between the time when the plunger reaches the end of its stroke and when the valve is completely closed is called the *delayed valve closing* and is normally 2 to 12 degrees of crank rotation. Although a delayed valve opening may have little effect on overall pump performance, a delayed valve closing certainly will. A valve that is partially open when the plunger reverses direction will result in backflow. Backflow, as the name implies, is the reverse flow of fluid back across the valve and results in lower pump volumetric efficiency.

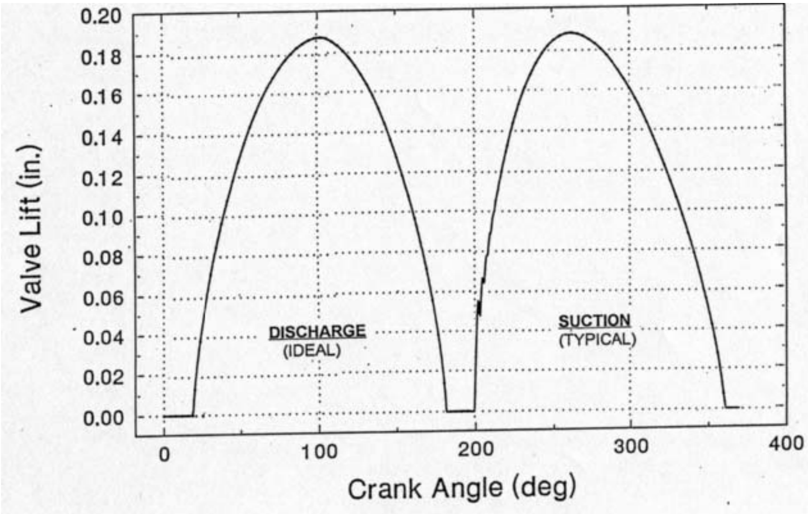


FIGURE 9 Valve lift versus crank angle (in × 2.54 = cm)

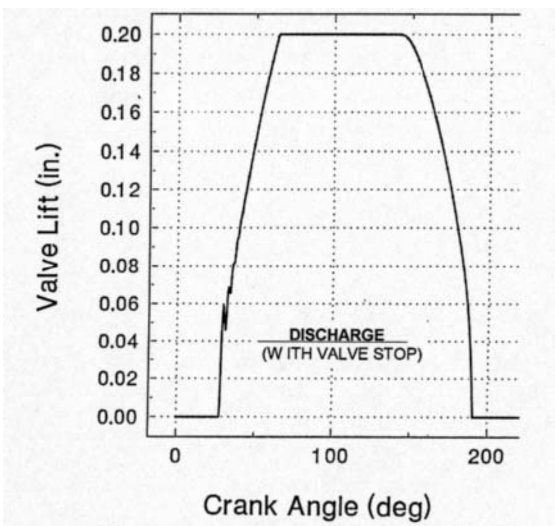


FIGURE 10 Valve lift versus crank angle for a valve with a mechanical stop (in × 2.54 = cm)

Many pump valves are designed with mechanical limits to the distance that the valve is allowed to open or lift. This is normally done to avoid overstressing the valve spring and to minimize the overall height of the valve assembly. Valve motion for a valve designed with a mechanical stop is shown in Figure 10. Another advantage of this design is that the valve does not have to travel as far during its closing cycle, compared to valves with no mechanical limits. The disadvantage is the potential for impact damage to the valve if it strikes the stop with excessive force.

The importance of valve mass to overall valve dynamics has been the subject of much debate among pump designers over the years. There is no question that the mass of the valve must be overcome in order to open the valve. Test data is also available that shows no appreciable difference in the performance of pumps fitted with hollow ball valves versus identical pumps with solid ball valves. This apparent contradiction can be explained by studying valve acceleration at various parts of the valve cycle using advanced computer modeling.

It has already been explained above why a valve can hesitate, or stop opening, during the opening portion of its cycle. It's also been stated that when the valve spill area is large enough to establish a hydraulic balance, the valve will stop opening. What actually occurs is the valve starts decelerating as it comes closer to hydraulic equilibrium. If the valve is properly designed, it will contact its mechanical stop just as its acceleration/deceleration is very low. In that case, the valve mass has little significance on the impact force of the valve on the stop. The same holds true of the impact force of the valve on the seat as it closes. If the valve is properly designed, a 62-lb (28 kg) solid ball valve operating at 100 cycles per minute can strike the seat with less than 20 lbs (9 kg) of force.

Although pump valves operate in a liquid medium, the shape of the valve is not an important design consideration for most applications. The relatively small distance that a valve travels is not sufficient for the fluid dynamics of a shape to have a measurable effect on valve dynamics. The only exception to this is valves in pumps handling high-viscosity liquids. For these applications, the ball valve, often spring-loaded, has proven to reduce closing valve lag and increase volumetric efficiency better than any other type of valve.

The most critical component in the optimization of valve dynamics is the valve spring. The pump designer normally selects a valve spring that will exert a certain amount of "pre-load" on the valve when it is closed. This pre-load helps the valve to close smoothly on the seat and avoid rebound (and possible backflow). Too high a preload in the suction valve may result in higher net positive suction head required (NPSHR). In the discharge valve, excessive preload can cause abnormally high pressure spikes in the fluid cylinder just before the valve opens.

The other valve spring design criterion is the spring rate. Every compression spring develops a predetermined resistance per unit length. This value is expressed in pounds per inch (kg per cm). As the valve is opening, the increasing spring force helps the valve obtain hydraulic balance faster. It also helps to limit the impact force of the valve on the stop. At the start of the closing cycle, the stored energy in the compressed spring helps the valve respond faster to the pressure changes in the fluid cylinder as the plunger starts to decelerate. Again, if the spring is properly designed, the closing valve lag will be minimized.

No published guidelines exist for the proper amount of valve spring pre-load or spring rate. Most pump designers use proprietary values, generated through a combination of in-house and field testing. However, although these values produce low NPSHR and high volumetric efficiency in most cases, the valve dynamics may not be close to being optimized. It should also come as no surprise that pumps fitted with "off-the-shelf" commercially available valves do not operate as well as pumps having optimized valve dynamics.

With the recent advent of advanced computer modeling of pump valves, it is now possible to optimize valve dynamics for a specific set of pump operating parameters. We may also see valve designs in the near future having variable rate valve springs, hydraulic dampening, and mechanisms that induce rotation as the valve opens and closes.

POWER END THEORY

The power end or drive end of a power pump consists of a crankshaft, connecting rods, crossheads, and bearings, all housed in a rigid structure referred to as the frame. Details of the design and construction of these components are covered in Section 3.2, "Power Pump Design and Construction." The slider-crank mechanism that converts rotational driving energy to the reciprocating motion that actuates the pistons or plungers can be found in reciprocating gas compressors, automotive engines, and stationary and marine engines. However, the stress loading pattern of power pump components is unique to this type of mechanism.

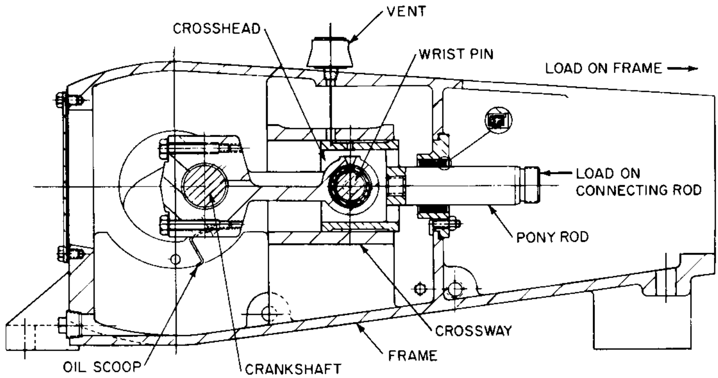


FIGURE 11 Power end, horizontal pump (Flowserve Corporation)

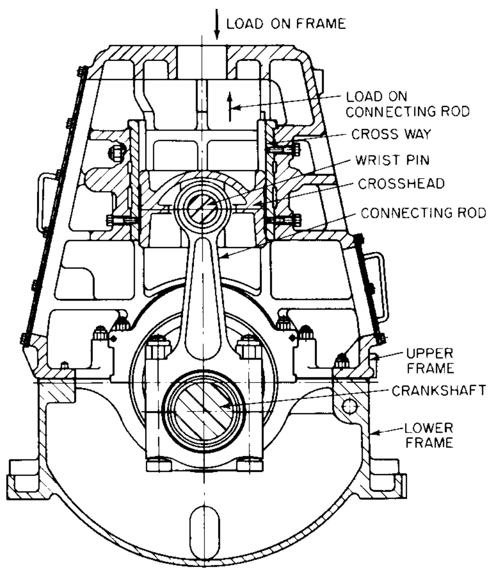


FIGURE 12 Power end, vertical pump (Flowserve Corporation)

Applying Rod Load to the Power End As stated earlier, the loading on the power end is called the rod load and is the product of the area of the plunger multiplied by the maximum discharge pressure. In a gas compressor or engine, this loading increases over 90 to 180 degrees of crank rotation before maximum loading is reached. In a power pump, the maximum loading is reached in less than 30 degrees of crank rotation due to the relative incompressibility of the pumpage. Since this loading cycle is repeated with each plunger stroke, the actual loading resembles a shock load more than a simple cyclic fatigue load. The design criteria for the stressed components of the power end must therefore include the material's capability to absorb shock loads and overall safety factors greater than 3:1.

Another critical factor in the capability of the power-end components to handle the rod load is how the load is applied. Figures 11 and 12 show the direction of the load on the frame and connecting rod for two different power end configurations. On vertical pumps with outboard packed stuffing boxes, the frame is in compression and the crosshead and connection rod are in tension. With horizontal, single-acting pumps, the frame is in tension, and the crosshead and connecting rod are in compression. The material selection and factors of safety must be altered accordingly.

Liquid Separation from the Plunger It has already been explained that fluid flows through the suction valve (and suction manifold and related piping) to fill the increasing cylinder volume caused by the withdrawal of the plunger. If the plunger accelerates faster than the flow of incoming liquid, the liquid will lose contact or separate from the plunger. The void that forms will be at a pressure lower than anywhere else in the cylinder. If the pumpage contains entrained gas, the gas may come out of a solution in this low-pressure area. The gas bubbles, when recompressed later in the plunger stroke, can cause cavitation damage to the plunger and surrounding cylinder.

The geometry of the slider-crank mechanism affects the point at which liquid separation occurs. As the ratio of the connecting rod length to the crank radius increases, the pump speed at which liquid separation occurs will decrease. Since liquid separation can be the determining factor in a pump's NPSHR, the pump designer must carefully evaluate the slider-crank geometry in order to optimize the pump's hydraulic performance. The pump speed at which water will separate from the end of the plunger can be calculated from the following:

In USCS units: $$\text{rpm} = 54.5\sqrt{\frac{(34 - h_s - h_f)A_s}{LR[l - (l/LR)]A_p}}$$

In SI units: $$\text{rpm} = 16.6\sqrt{\frac{(10.36 - h_s - h_f)A_s}{LR[l - (l/LR)]A_p}}$$

where h_s = suction head, ft (m)
h_f = piping frictional loss, ft (m)
A_s = area of suction pipe, in^2 (mm^2)
L = length of connecting rod, centerline to centerline, ft (m)
R = crank radius, ft (m)
l = length of pipe where resistance to flow is to be measured, ft (m)
A_p = area of plunger, in^2 (mm^2)

Unbalanced Forces Due to the relatively slow speed of a power pump, the inertia loads of the rotating/reciprocating parts are low enough to avoid the vibration problems associated with centrifugal pumps. For that reason, power pump crankshafts are not normally balanced. However, when the power pump is coupled to a drive containing a high-speed motor and gear reducer, a torsional analysis of the pump/drive unit may be required. For this analysis, the unbalanced forces of the rotating and reciprocating pump components can be calculated as follows:

Unbalanced Reciprocating Parts Force (F_{rec}) Parts are typically about one-third of the connecting rod weight (the crosshead, the crosshead bearing, the wrist pin, the pony rod, and the plunger). Additional parts on vertical pumps include the pull rods, yoke, and plunger nut.

$$F_{rec} = \frac{W}{g}\omega^2 R\left(\cos\theta + \frac{R}{\mathrm{L}}\cos\theta\right)$$

where F_{rec} = reciprocating parts force, lb (N)
W = weight of all reciprocating or rotating parts, lb (N)

g = 32.2 ft/s² (9.81 m/s²)

$\omega = (2\pi/60) \times n$ (n = pump rpm)

R = one-half of stroke, ft (m)

L = length of connecting rod, centerline to centerline, ft (m)

θ = crank angle; usually maximum force is at $\theta = 0°$, $\cos\theta = 1$

Unbalanced Rotating Parts Force (F_{rot}) Parts are typically about two-thirds of the connecting rod weight, crank end bearing, and crankpin, where the variables are as above.

$$F_{rot} = \frac{W}{g}\omega^2 R$$

Mechanical Efficiency The mechanical efficiency of a single-acting power pump without internal gears is typically 90 to 92%. Over half of the mechanical losses are due to the frictional drag of the plungers through the packing. The remaining losses are from the bearings, the crosshead-to-crosshead guide friction, and the extension rod-to-gland seal friction. If these components are properly lubricated, very few power-end design options are available that will produce a measurable increase in mechanical efficiency. Decreasing the diameter of the plungers and minimizing the number of packing rings will result in a small efficiency increase.

FURTHER READING

American National Standard for Reciprocating Power Pumps for Nomenclature, Definitions, Application and Operation, ANSI/HI 6.1-6.5-2000, Hydraulic Institute, Parsippany, NJ www.pumps/org

Positive Displacement Pumps—Reciprocating, API Standard 674. 2nd Edition, 1994, The American Petroleum Institute, Washington, D.C. www.api.org

Henshaw, T. L. *Reciprocating Pumps*. Van Norstrand Reinhold Company, Inc., New York, 1987.

Miller, J. E. *The Reciprocating Pump—Theory, Design and Use*. John Wiley & Sons, Inc., New York, 1987.