PUMP HANDBOOK

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Page 1 of 153

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Pump Handbook, Fourth Edition

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DISPLACEMENT PUMPS

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 3 of 153

SECTION 3.1
POWER PUMP THEORY

WILLIAM K. CHAPLIS
FREDERIC W. BUSE

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.

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 4 of 153

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 (flow rate), differential pressure, and overall 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 $$bhp = (Q \times Ptd \times 100)/(1714 \times OE)$$

In SI units $$kW = (Q \times Ptd \times 100)/(36 \times OE)$$

where Q = delivered capacity (flow rate), gpm (m^3/h)
Ptd = differential pressure (discharge − suction), lb/in^2 (bar)
OE = overall 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 (volume flow rate) 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.

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 5 of 153

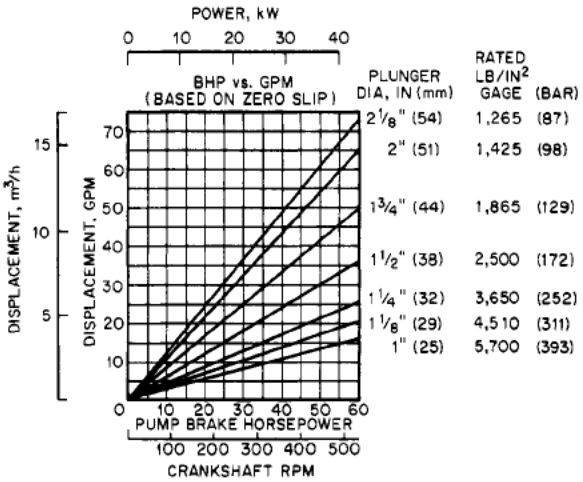


FIGURE 1 Typical power pump performance: brake horsepower versus gallons per minute (kilowatts versus m³h) based on zero slip (Flowserve Corporation)

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 displacement flow rate and is typically 1 to 4%:

$$S = 100 \times (B + V + L)/D$$

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 = displacement power/power in = $P_{displ}/P_{in} = D \times Pdt/1714 \times P_{in}$

In SI units ME = displacement power/power in = $P_{displ}/P_{in} = D \times Pdt/36 \times P_{in}$

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

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

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 6 of 153

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

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).

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

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

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 7 of 153

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

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,

$$VE = 100 \times (1 - Ptd \times \beta \times r)/(1 - Ptd \times \beta) - S$$

Based on discharge capacity,

$$VE = 100 \times (1 - Ptd \times \beta \times r) - S$$

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,

$$VE = 100 \times [r - (p_d/p_s)(r - 1)] - S$$

Based on discharge capacity,

$$VE = 100 \times [1 - r(1 - p_s/p_d)] - S$$

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)

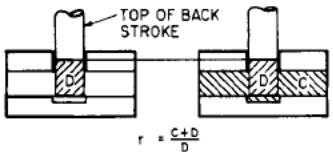


FIGURE 2 The ratio r (Flowserve Corporation)

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 8 of 153

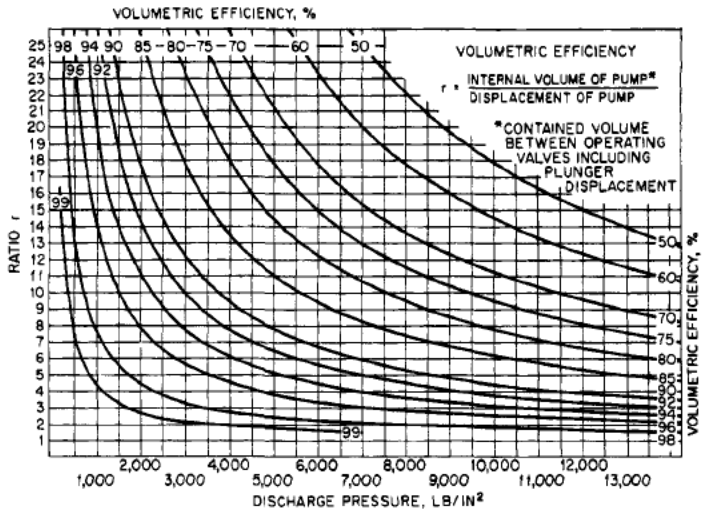


FIGURE 3 Volumetric efficiency (Flowserve Corporation) (lb/in² × 0.69 = bar)

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

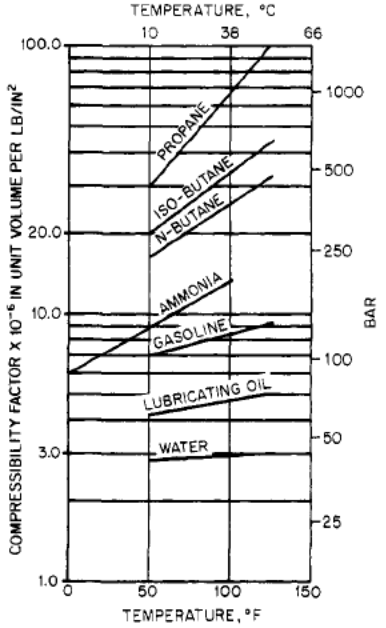


FIGURE 4 Compressibility factor (Flowserve Corporation) (bar × 14.5 = lb/in²)

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 9 of 153

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%

be considered. Special "high torque" motors are often required if the pump is to be started under the load.

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.

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)

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 10 of 153

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.

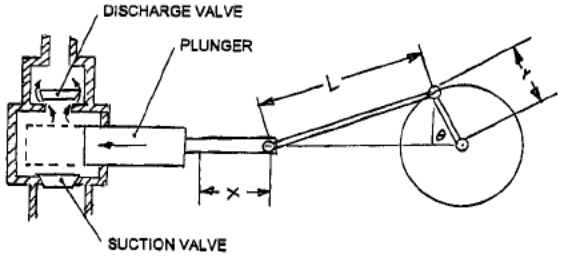


FIGURE 5 Slider-crank mechanism

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Petitioner
EX1010
Page 11 of 153

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

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 $A = 0.785 \times D^2$

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

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 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.

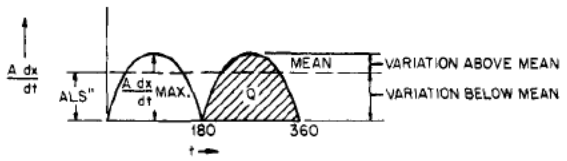


FIGURE 6 The discharge rate for a single double-acting pump (Flowserve Corporation)

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 12 of 153

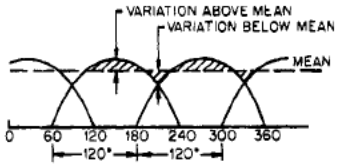


FIGURE 7 The discharge rate for a triplex single acting pump (Flowserve Corporation)

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 preestablished 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:

In USCS units V (ft/sec) = gpm through the valve × 0.642/spill area of the valve, in^2

In SI units V (m/s) = m^3/h through valve × 555.6/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

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.

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 13 of 153

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

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° 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).

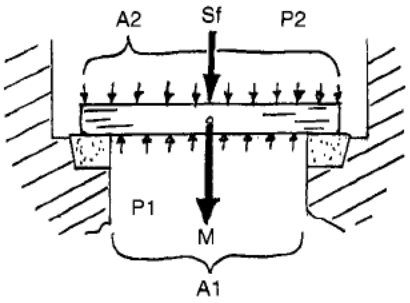
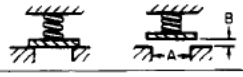
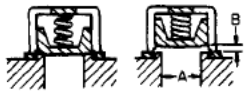
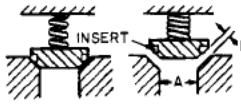


FIGURE 8 Force balance of a typical valve

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 14 of 153

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

TYPE	SKETCH	PRESSURE, PSI (BAR)	APPLICATION
PLATE	A = SEAT AREA B = SPILL AREA 	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

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° 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.

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)

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 15 of 153

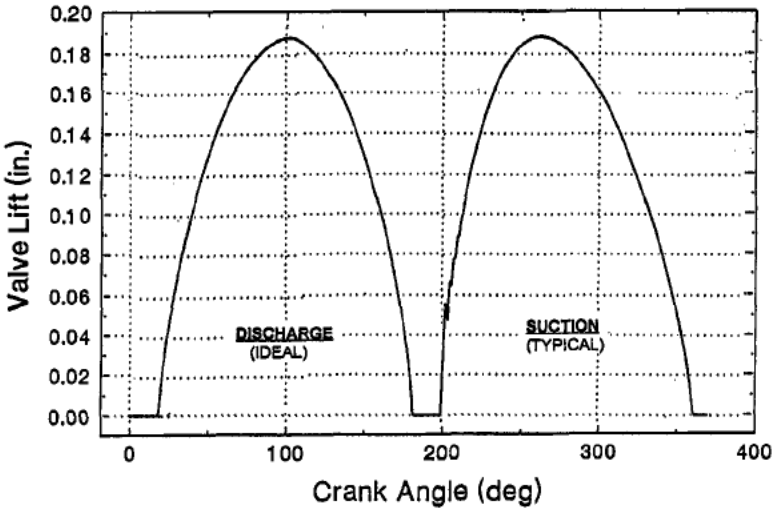


FIGURE 9 Valve lift versus crank angle (in × 2.54 = cm)

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.

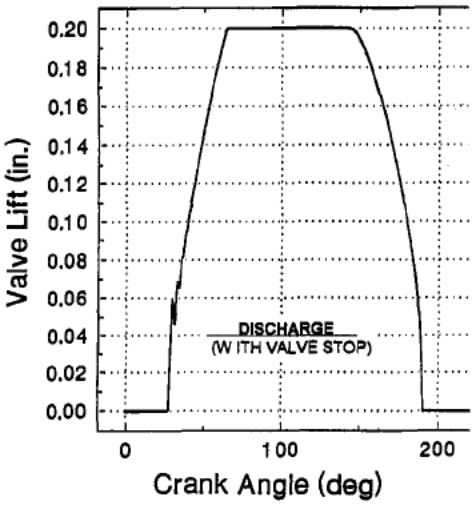


FIGURE 10 Valve lift versus crank angle for a valve with a mechanical stop (in × 2.54 = cm)

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 16 of 153

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 are 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 lb (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.

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 17 of 153

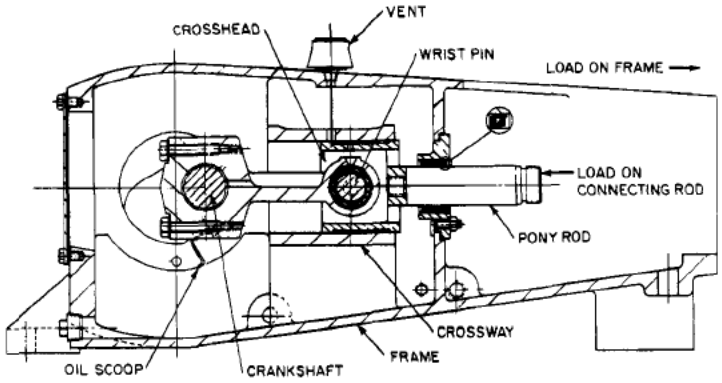


FIGURE 11 Power end, horizontal pump (Flowserve Corporation)

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° of crank rotation before maximum loading is reached. In a power pump, the maximum loading is reached in less than 30° 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

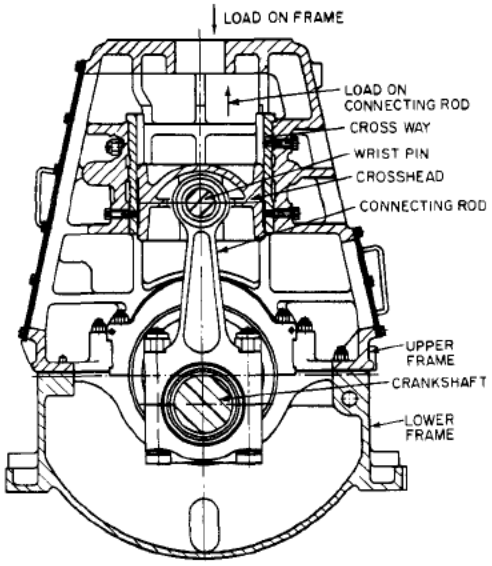


FIGURE 12 Power end, vertical pump (Flowserve Corporation)

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 18 of 153

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, in² (mm²)
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, in² (mm²)

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.

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

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

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 19 of 153

g = 32.2 ft/s² (9.81 m/s²)
ω = $(2\pi/60) \times n$ (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.

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GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 20 of 153

SECTION 3.2
POWER PUMP DESIGN AND CONSTRUCTION

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FREDERIC W. BUSE

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% component interchangeability with triplex

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 21 of 153

pumps. Triplex (three cylinder), quintuplex (five cylinder), septuplex (seven cylinder), and nonoplex (nine cylinder) pumps have been designed that have over a 90% 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 loosing 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.

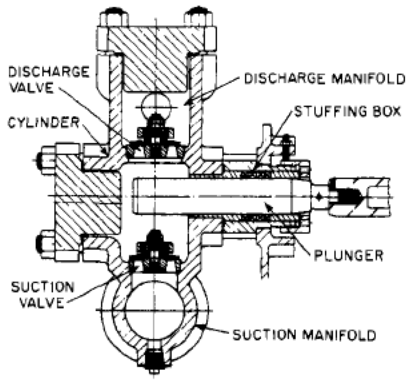


FIGURE 1 A liquid-end horizontal pump (Flowserve Corporation)

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 22 of 153

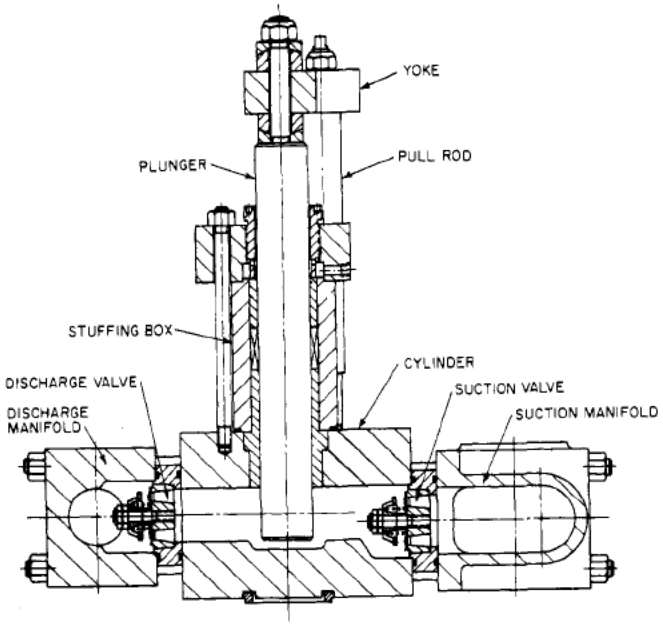


FIGURE 2 A liquid-end vertical pump (Flowserve Corporation)

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/in^2 (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/in^2 (69 to 172 Mpa), depending on the material of the cylinder, duty cycle, and liquid being

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)

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 23 of 153

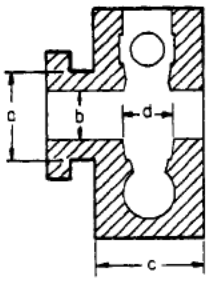


FIGURE 3 Stress dimensions for a horizontal liquid end (Flowserve Corporation)

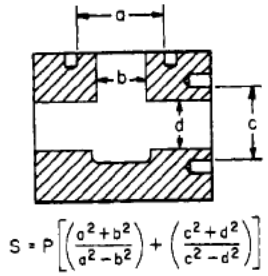


FIGURE 4 Stress dimensions for a vertical liquid end (Flowserve Corporation)

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.

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 24 of 153

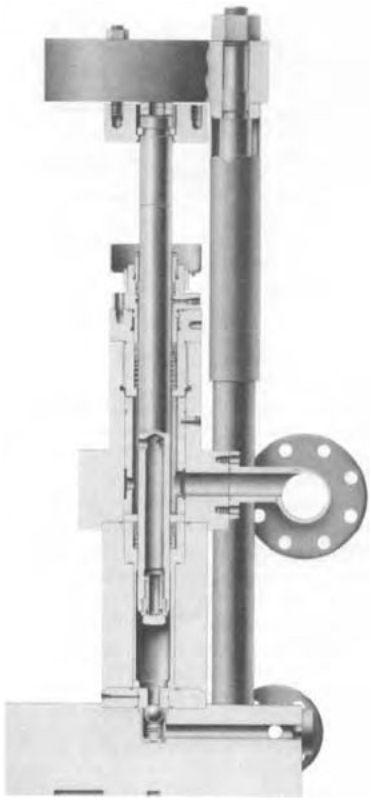


FIGURE 5 Special high-pressure liquid-end designs arrange the suction and discharge valves on the same axis as the plunger, thus eliminating intersecting bores (Flowserve Corporation).

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/in^2 (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

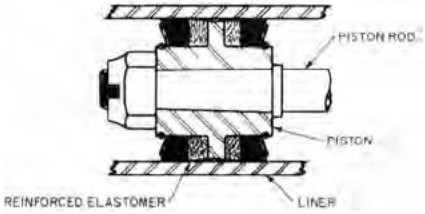


FIGURE 6 Elastomer face piston (FWI)

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 25 of 153

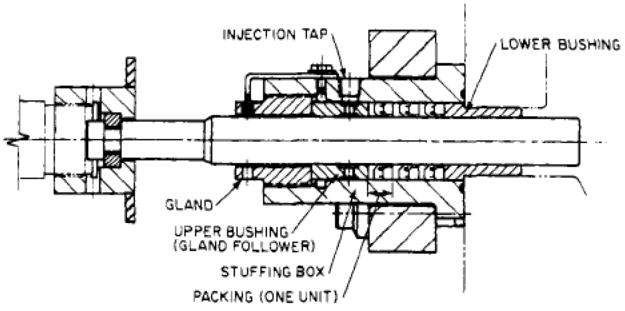


FIGURE 7 A stuffing box (Flowserve Corporation)

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

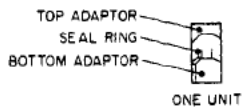


FIGURE 8 Chevron packing (Flowserve Corporation)

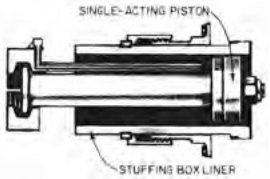


FIGURE 9 Single-acting piston stuffing box (Flowserve Corporation)

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 26 of 153

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 and 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, in^2

In SI units, V (m/s) = pump m^3/h $\times$ 277.8/cross-sectional area of the manifold, mm^2

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

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 27 of 153

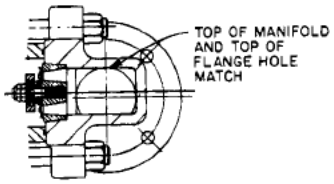


FIGURE 10 A suction manifold construction to eliminate air pockets (Flowserve Corporation)

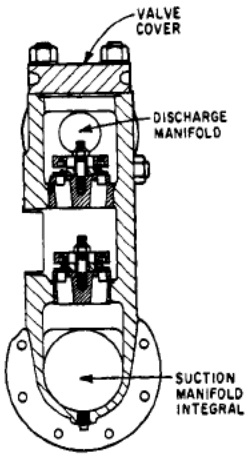


FIGURE 11 An integral suction and discharge manifold (Gardner-Denver)

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

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 28 of 153

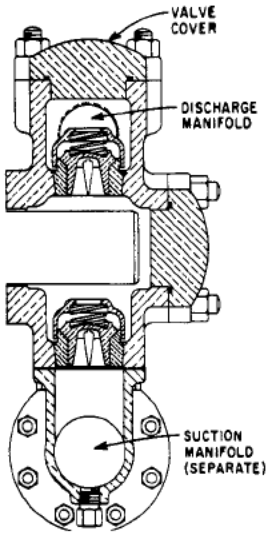


FIGURE 12 Separate suction with an integral discharge manifold (Gardner-Denver)

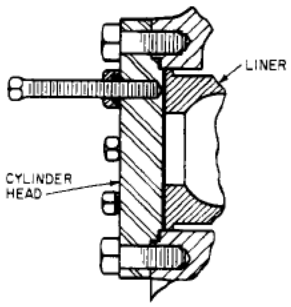


FIGURE 13 A plunger cover (FWI)

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

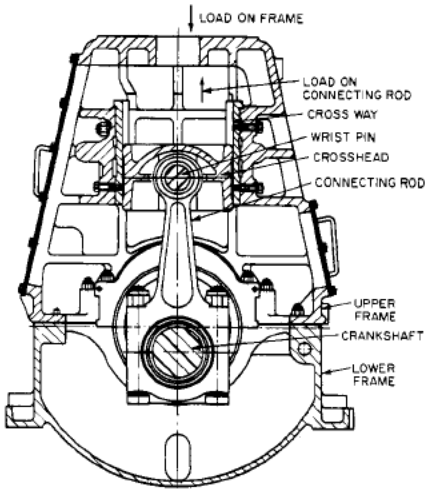


FIGURE 14 A vertical pump power end (Flowserve Corporation)

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 29 of 153

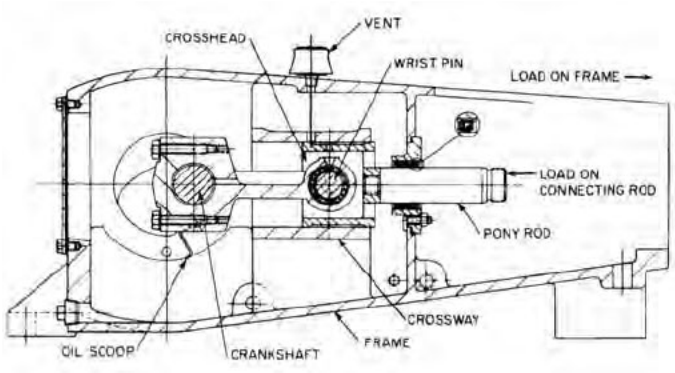


FIGURE 15 A horizontal pump power end (Flowserve Corporation)

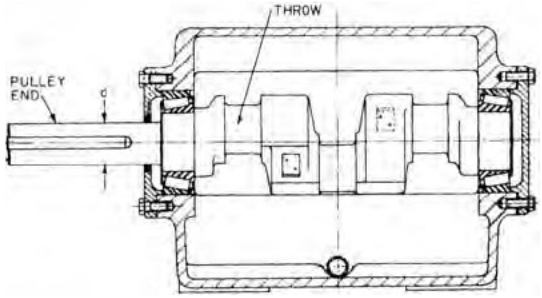


FIGURE 16 A cast crankshaft (Flowserve Corporation)

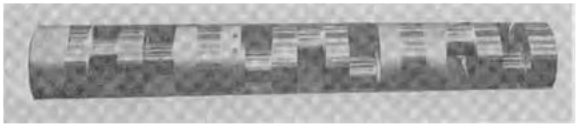


FIGURE 17 A crankshaft machined from a billet (Flowserve Corporation)



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

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 30 of 153

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

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/in^2 (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/in^2 (69 MPa):

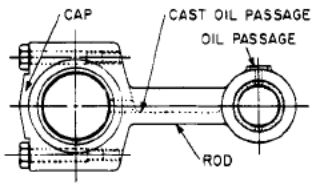


FIGURE 19 The connecting rod (Flowserve Corporation)

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 31 of 153



FIGURE 20 An eccentric strap (Gardner-Denver)

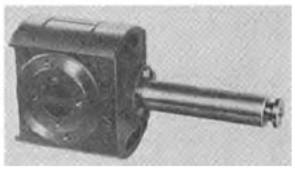


FIGURE 21 A crosshead (Gardner-Denver)

In USCS units, $$S = \frac{PL \times \ell}{8 \times 0.098 d^3}$$

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/in^2 (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

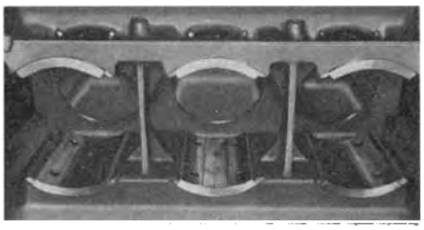


FIGURE 22 Crossways (Gardner-Denver)

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 32 of 153

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/in^2 (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/in^2 (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/in^2 (5.1 MPa).

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 33 of 153

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/in^2 (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

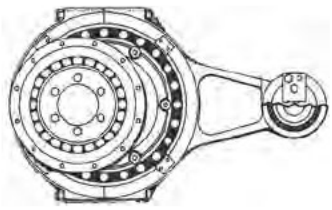


FIGURE 23 The main bearing mounted in a bearing holder (Gardner-Denver)

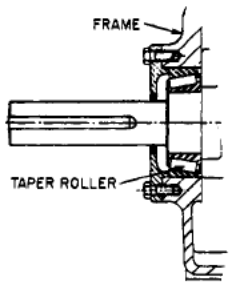


FIGURE 24 The main bearing mounted directly in the frame (Flowserve Corporation)

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 34 of 153

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% 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%
- Triplex and above: 10%

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

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 35 of 153

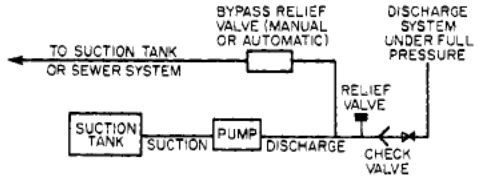


FIGURE 25 A bypass relief system (Flowserve Corporation)

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% of the system pressure. Although these units are relatively inexpensive, they can attenuate up to 90% of the pressure pulsations, below 50 Hz, in the interconnecting piping systems.

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 36 of 153

SECTION 3.3
STEAM PUMPS

ROBERT M. FREEBOROUGH

BASIC THEORY

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

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

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

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

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

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 37 of 153

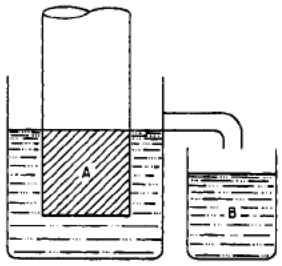


FIGURE 1 The volume of liquid displaced by a solid equals the volume of the solid

vacated by withdrawal of the plunger. During this phase of operation, the discharge check valve is held closed by the higher pressure in the discharged pipe. This portion of the pumping action of a reciprocating positive displacement pump is called the *suction stroke.*

The withdrawal movement must be stopped before the end of the plunger gets to the packing. The plunger movement is then reversed, and the *discharge stroke* portion of the pumping action is started, as illustrated in Figure 3.

Movement of the plunger into the cylinder causes an increase in pressure of the liquid contained therein. This pressure immediately becomes higher than suction pipe pressure and causes the suction valve to close. With further plunger movement, the liquid pressure continues to rise. When the liquid pressure in the cylinder reaches that in the discharge pipe, the discharge check valve is forced open and liquid flows into the discharge pipe. The volume forced into the discharge pipe is equal to the plunger displacement less very small losses. The plunger displacement is the product of its cross-sectional area and the length of stroke. The plunger must be stopped before it hits the bottom of the cylinder. The motion is then reversed, and the plunger again goes on suction stroke as previously described.

The pumping cycle just described is that of a *single-acting* reciprocating pump. It is called single-acting because it makes only one suction stroke and only one discharge stroke in one reciprocating cycle.

Many reciprocating pumps are *double-acting;* that is, they make two suction and two discharge strokes for one complete reciprocating cycle (Figure 4). Most double-acting pumps use as the displacing solid a piston that is sealed to a bore in the liquid cylinder or

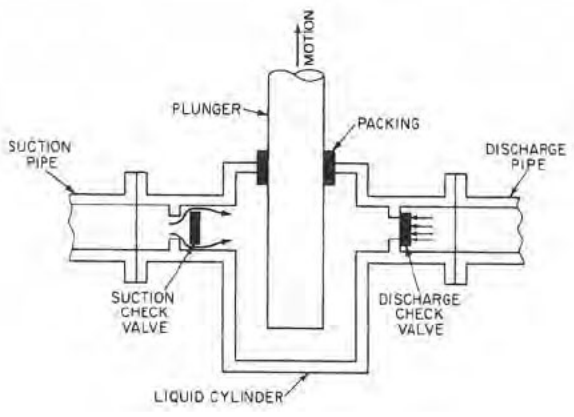


FIGURE 2 Schematic of the liquid end of reciprocating pump during the suction stroke

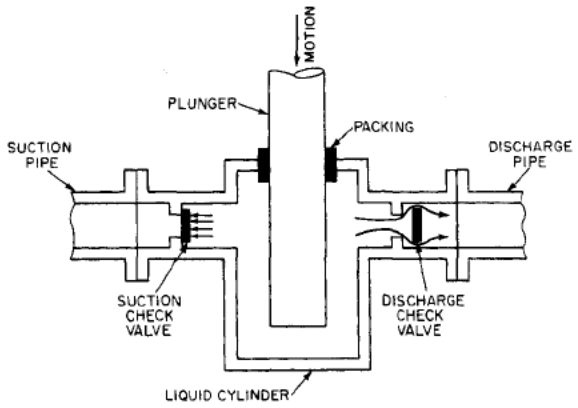


FIGURE 3 Schematic of the liquid end of a reciprocating pump during the discharge stroke

to a liquid cylinder liner by pistons with packing. It has two suction and two discharge valves, one of each on each side of the piston. The piston is moved by a piston rod. The piston rod packing prevents liquid from leaking out of the cylinder. When the piston rod and piston are moved in the direction shown, the right side of the piston is on a *discharge stroke* and the left side of the piston is simultaneously on a *suction stroke.* The piston packing must seal tightly to the cylinder liner to prevent leakage of liquid from the high-pressure right side to the low-pressure left side.

The piston must be stopped before it hits the right side of the cylinder. The motion of the piston is then reversed so the left side of the piston begins its discharge stroke and the right side begins its suction stroke.

A reciprocating pump is not complete with a liquid end only; it must also have a driving mechanism to provide motion and force to the plunger or piston. The two most common driving mechanisms are a reciprocating steam engine and a crank-and-throw device. Those pumps using the steam engine are called *direct-acting steam pumps.* Those pumps using the crank-and-throw device are called *power pumps.* Power pumps must be connected to an external rotating driving force, such as an electric motor, steam turbine, or internal combustion engine.

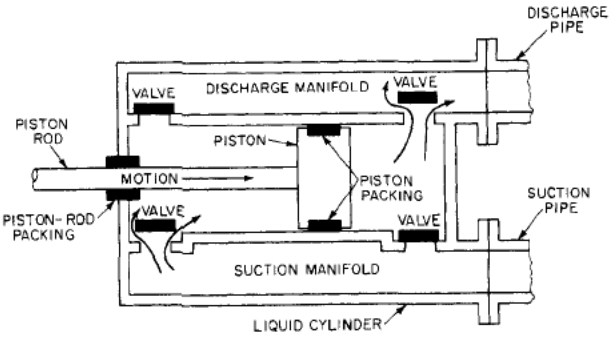


FIGURE 4 Schematic of a double-acting liquid end

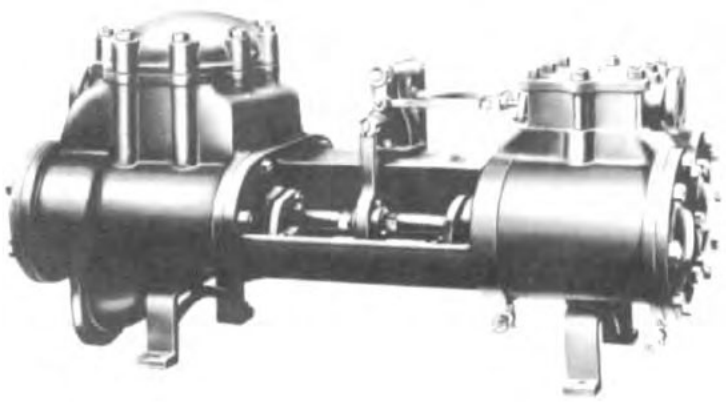


FIGURE 5 Duplex steam pump (Flowserve Corporation)

DIRECT-ACTING STEAM PUMPS

Direct-acting steam pumps are mainly classified by the number of working combinations of cylinders. For example, a duplex pump (Figure 5) has two steam and two liquid cylinders mounted side by side, and a simplex pump (Figure 6) has one steam and one liquid cylinder.

Additionally, simplex and duplex pumps may be further defined by (1) cylinder arrangement, whether horizontal or vertical; (2) number of steam expansions in the power end; (3) liquid end arrangement, whether piston or plunger; and (4) valve arrangement, that is, cap and valve plate, side pot, turret type, and so on.

Although this section will refer to steam as the driving medium, compressed gases such as air or natural gas can be used to drive a steam pump. These gases should have oil or mist added to them prior to entering the pump to prevent wear of the steam end parts.

STEAM END CONSTRUCTION AND OPERATION

The driving mechanism, or *steam end,* of a direct-acting steam pump includes the following components, as illustrated in Figure 7:

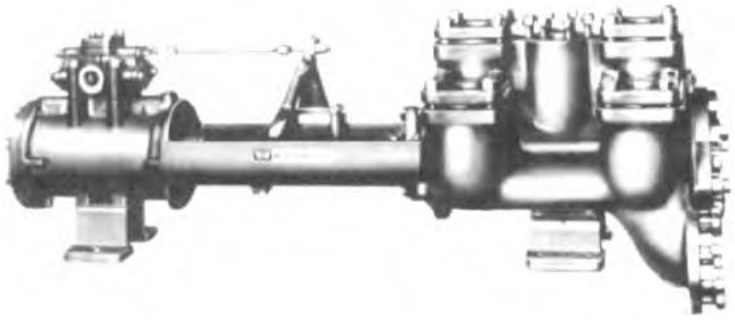


FIGURE 6 Simplex steam pump (Flowserve Corporation)

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 40 of 153

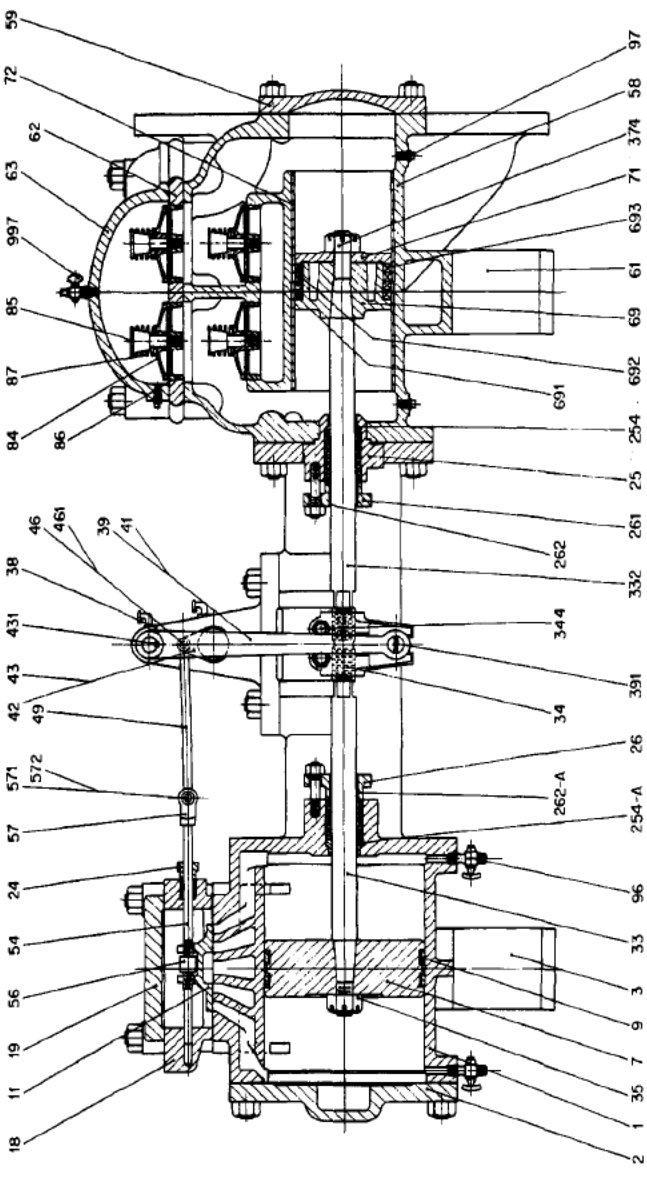


FIGURE 7 Typical section of duplex steam pump: (1) Steam cylinder with cradle, (2) steam cylinder head, (3) steam cylinder foot, (7) steam piston, (9) steam piston rings, (11) slide valve, (18) steam chest, (19) steam chest cover, (24) valve rod stuffing box gland, (25) piston rod stuffing box, liquid,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

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 41 of 153

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

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

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

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

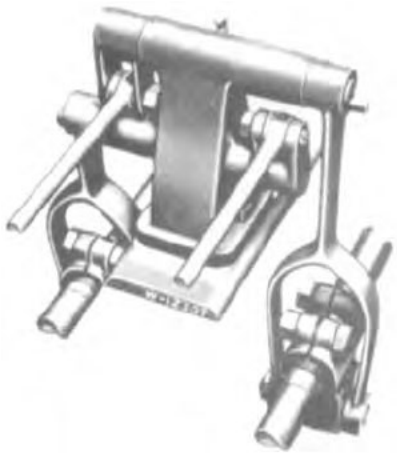


FIGURE 8 Steam valve actuating mechanism or a duplex pump (Flowserve Corporation)

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 42 of 153

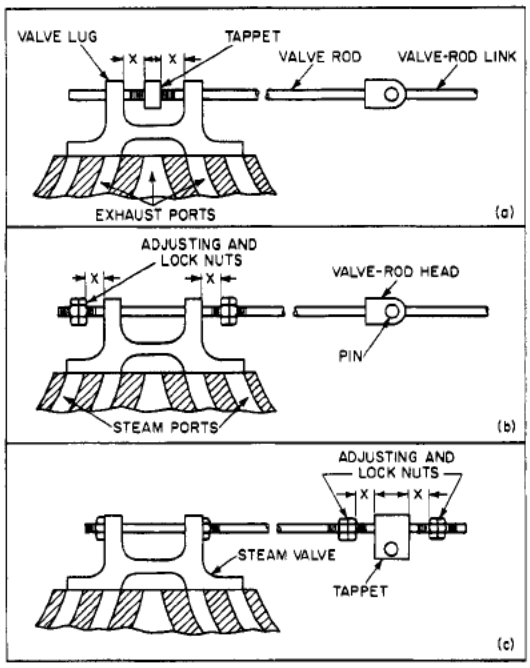


FIGURE 9A through C Duplex pump steam valve lost-motion arrangements

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

Flat Slide Steam Valves Steam enters the pump from the steam pipe into the steam chest on top of the steam cylinder. Exhaust steam leaves the pump through the center port of five ports, as shown in Figure 9. Most duplex pumps use a flat slide valve that is held against its seat by steam pressure acting upon its entire top area; this is called an *unbalanced* valve. The flat or D valve, as it is often called, is satisfactory for steam pressures up to approximately 250 lb/in^2 (17 bar*) and has a reasonable service life, particularly where steam end lubrication is permissible. On large pumps, the force required to move an unbalanced valve is considerable, and so a balanced piston valve, which will be discussed later, is used.

The slide valve shown in Figure 9 is positioned on dead center over the five valve ports. A movement of the valve to the right uncovers the left-side steam port and the right-side exhaust port, which is connected through the slide valve to the center exhaust port. The main steam piston will be moved from left to right by the admitted steam. Movement of the slide valve from dead center to the left would, of course, cause opposite movement of the steam piston.

The steam valves of a duplex pump are mechanically operated, and their movements are dependent upon the motion of the piston rod and the linkage of the valve gear. In order to ensure that one piston will always be in motion when the other piston is reversing at the end of its stroke, lost motion is introduced into the valve gear. Lost motion is a means by which the piston can move during a portion of its stroke without moving the steam valve. Several lost motion arrangements are shown in Figure 9.

*1 bar = 10^5 Pa.

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 43 of 153

If the steam valves are out of adjustment, the pump will have a tendency not to operate through its designed stroke. Increasing the lost motion lengthens the stroke; if this is excessive, the piston will strike the cylinder head. Reducing the lost motion shortens the stroke; if this is excessive, the pump will short-stroke, with a resulting loss in capacity.

The first step in adjusting the valves is to have both steam pistons in a central position in the cylinder. To accomplish this, the piston is moved toward the steam end until the piston strikes the cylinder head. With the piston rod in this position, a mark is made on the rod flush with the steam end stuffing box gland. Next, the piston rod is moved toward the liquid end until the piston strikes, and then another mark is placed on the rod halfway between the first mark and the steam end stuffing box gland. After this, the piston rod is returned toward the steam end until the second mark is flush with the stuffing box gland. The steam piston is now in central position. This procedure is repeated for the opposite piston rod assembly.

The next step is to see that both steam valves are in a central position with equal amounts of lost motion on each side, indicated by distance X in Figure 9.

Most small steam pumps are fitted with a fixed amount of lost motion, as shown in Figure 9a. With the slide valve centered over the valve ports, a properly adjusted pump will have the tappet exactly centered in the space between the valve lugs. The lost motion (X) on each side of the tappet will be equal.

Larger pumps are fitted with adjustable lost motions, such as are shown in Figure 9b. The amount of lost motion (X) can be changed by moving the locknuts. Manufacturers provide specific instructions for setting proper lost motion. However, one rule of thumb is to allow half the width of the steam port on each side for lost motion. A method to provide equality of lost motion is to move the valve each way until it strikes the nut and then note if both port openings are the same.

In some cases, it is desirable to be able to adjust the steam valves while the pump is in motion. With the arrangements previously mentioned, this cannot be done because the steam chest head must be removed. In a pump equipped with a lost-motion mechanism such as that shown in Figure 9c, all adjustments are external and can therefore be made while the pump is in operation.

Balanced Piston Steam Valve Thebalanced piston steam valve (Figure 10) is used on duplex steam pumps when the slide-type valve cannot be used because of size. The balanced piston valve can also be used without lubrication at pressure above 250 lb/in^2 (17 bar) and temperatures above 500°F (260°C). At higher pressures, wire drawing or steam cutting can occur as the piston slowly crosses the steam ports. To prevent wear and permanent damage to the steam chest and piston, piston rings are used on the steam valve and a steam chest liner is pressed into the steam chest to protect it.

Cushion Valves Steam cushion valves are usually furnished on larger pumps to act as an added control to prevent the steam piston from striking the cylinder heads when the pump operates at high speeds. As previously shown, the steam end has five ports, the outside ports are for steam admission, and the inside ports are for steam exhaust. As the steam piston approaches the end of the cylinder, it covers the exhaust port, trapping a

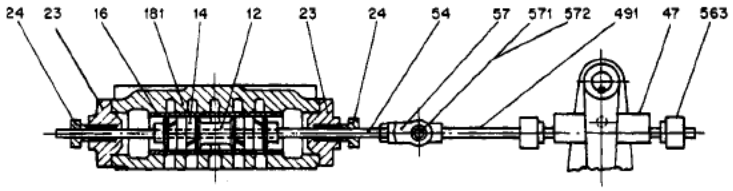


FIGURE 10 Balanced piston steam valve: (12) piston valve, (181) steam chest, (23) valve rod stuffing box, (24) valve rod stuffing box gland, (54) valve rod, complete, (57) valve rod head, (14) piston valve ring, (16) piston valve lining, (47) lost-motion block tappet, (491) valve rod link, (563) valve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

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 44 of 153

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

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

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

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

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

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

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

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

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

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

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

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 45 of 153

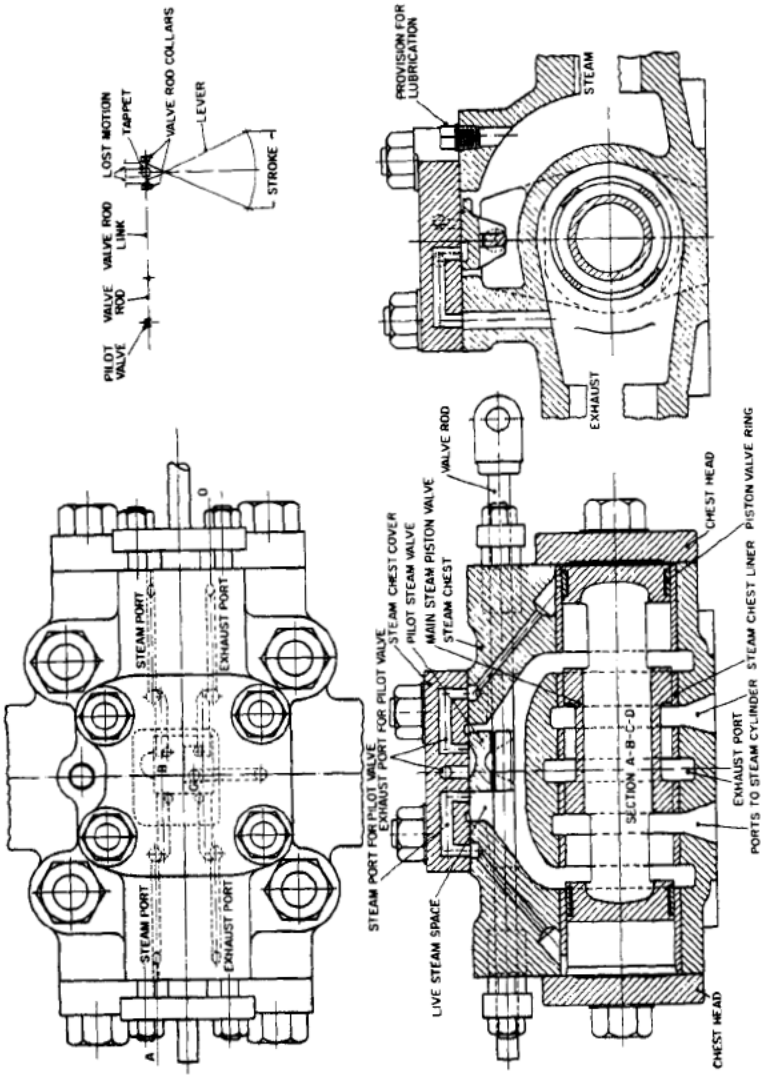


FIGURE 11 Simplex-type steam valve (Flowserve Corporation)

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 46 of 153

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

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

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

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

LIQUID END CONSTRUCTION

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

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

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

Piston-Type Liquid Ends The most generally used piston pump is the cap-and-valve plate design, illustrated in Figure 7. This is usually built for low pressures and temperatures, although some designs are used at up to 350 lb/in^2 (246 bar) of discharge pressure and 350°F (177°C). The discharge valve units are mounted on a plate separate from the cylinder and have a port leading to the discharge connection. A dome-shaped cap, subject to discharge pressure, covers the discharge valve plate. The suction valve units are mounted in the cylinder directly below their respective discharge valves. A passage in the liquid cylinder leads from below the suction valves down between the cylinders of a duplex pump to the suction connection.

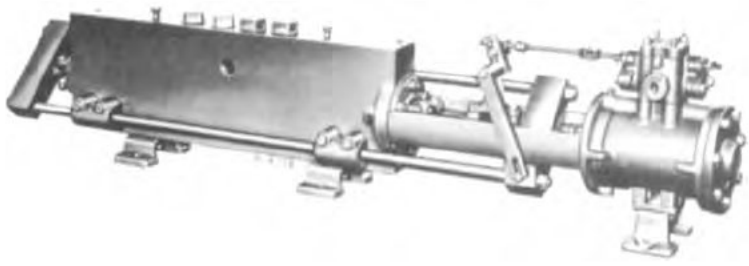


FIGURE 12 Simplex-type plunger pump (Flowserve Corporation)

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 47 of 153

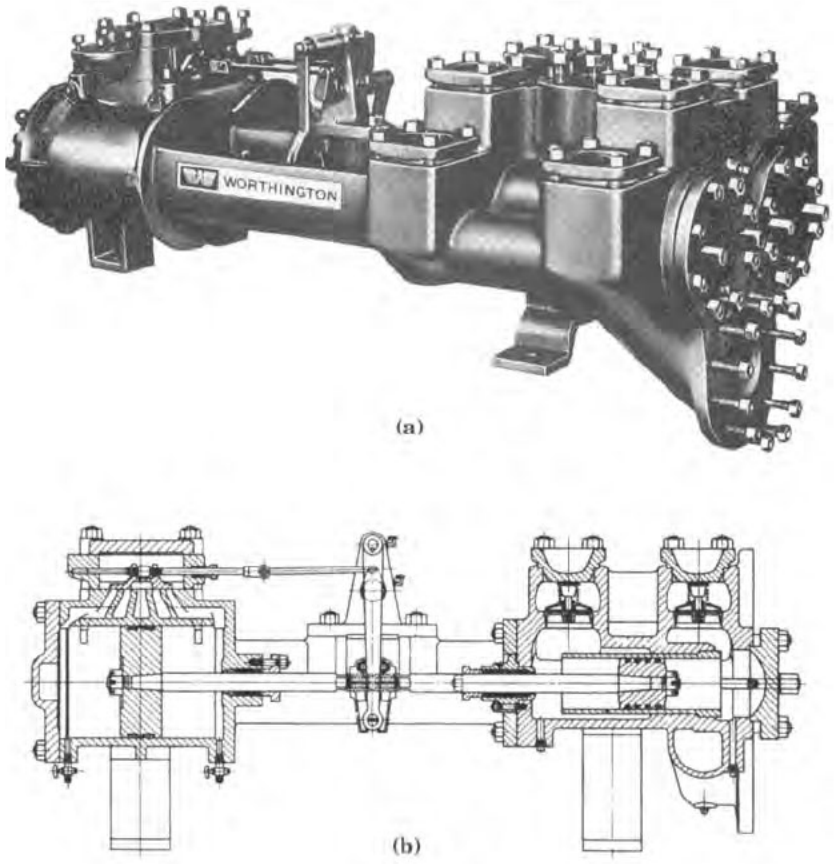


FIGURE 13A and B Side-pot piston pump (Flowserve Corporation)

Side-pot liquid ends are used where the operating pressures are beyond the limitations of the cap-and-valve-plate pump. Figure 13 illustrates this design. Suction valves are placed in individual pots on the side of the cylinders and discharge valves in the pots above the cylinders. Each valve can be serviced individually by removing its cover. The small area of the valve covers exposed to discharge pressure makes the sealing much simpler than is the case in the cap-and-valve design. Side-pot liquid ends are widely used in refinery and oil field applications. This design is commonly employed to the maximum pressure practicable for a piston pump.

There are several specially designed piston-type liquid ends that have been developed for specific applications. One of these is the close-clearance design illustrated in Figures 14 and 15. This pump can handle volatile liquids, such as propane or butane, or a liquid that may contain entrained vapors.

The close-clearance cylinder is designed to minimize the dead space when the piston is at each end of its stroke. The liquid valves are placed as close as possible to the pump chamber to keep clearance to a minimum. The suction valves are positioned below the cylinder at the highest points in the suction manifold to ensure that all the gases are passed into the pump chamber. Although these pumps are of close-clearance design, they are not compressors and can vapor-bind; that is, a large amount of gas trapped below the

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 48 of 153

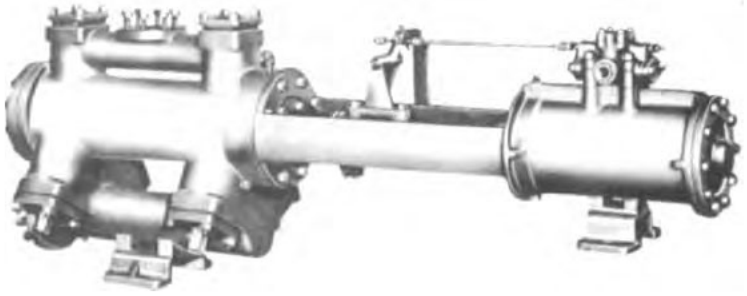


FIGURE 14 Close-clearance liquid end pump (Flowserve Corporation)

discharge valve will compress and absorb the entire displacement of the pump. When this occurs, the discharge valve will not open and this will cause a loss of flow. Hand-operated bypass or priming valves are provided to bypass the discharge valve and permit the trapped gases to escape to the discharge manifold. When the pump is free of vapors, the valves are closed.

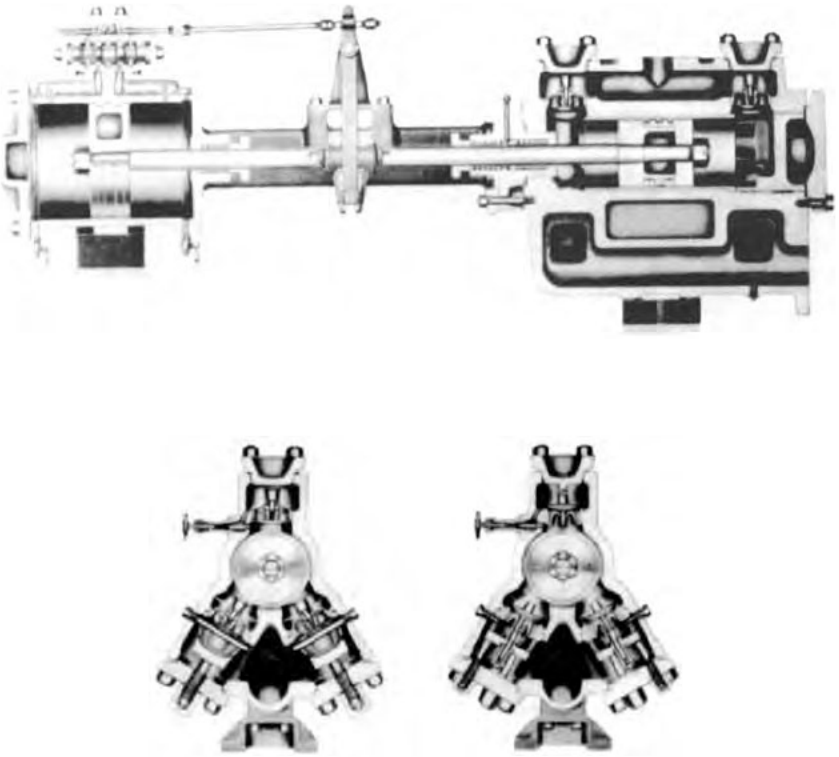


FIGURE 15 End views of close-clearance liquid end pump, showing disk valve assembly and wing valve assembly (Flowserve Corporation)

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 49 of 153

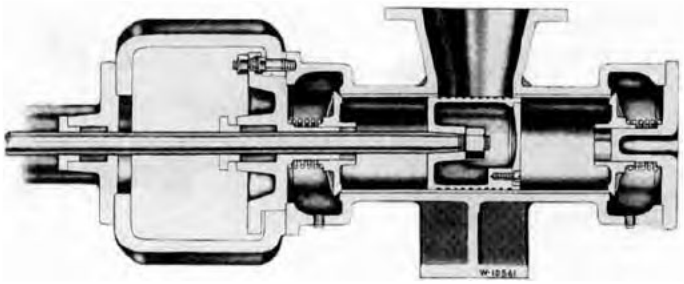


FIGURE 16 A pump for handling viscous liquid (Flowserve Corporation)

There are a number of other special piston pump designs for certain services in addition to the most common types just described. One of these special designs is the wet vacuum pump, which features tight-sealing rubber valves that permit the pump to handle liquid and air or non-condensable vapors. Another special design is made of hard, wear-resistant materials to pump cement grout on construction projects. Another design, shown in Figure 16, has no suction valves and is made for handling viscous products such as sugarcane pulp, soap, white lead, printer's ink, and tar. The liquid flows into the cylinder from above through a suction port that is cut off as the piston moves back and forth.

Piston Pump Liquid End Materials and Construction The materials used for piston pump liquid ends vary widely with the liquids handled. Most of the services to which these pumps are applied use one of the common material combinations listed in Table 1.

TABLE 1 Material and service specifications for pump liquid ends

Part	Regular Fitted (RF)	Bronze Fitted (BF)	Fully Bronze Fitted (FBF)	All Iron Fitted (AIF)	All Bronze (AB)
Cylinders	Cast iron	Cast iron	Cast iron	Cast iron	Bronze
Cylinder liners	Bronze	Bronze	Bronze	Cast iron	Bronze
Piston	Cast iron	Cast iron	Bronze	Cast iron	Bronze
Piston packing	Fibrous	Fibrous	Fibrous	Cast-iron, 3-ring	Fibrous
Stuffing box	Cast iron, bronze bushed	Cast iron, bronze bushed	Cast iron, bronze bushed	Cast iron	Bronze
Piston rod	Steel	Bronze	Bronze	Steel	Bronze
Valve service	Bronze	Bronze	Bronze	Steel	Bronze
Services for which most often used	Cold water; other cold liquids not corrosive to iron and bronze	Same as for RF, with reduced maintenance; continuous hot water	Boiler feed; intermittent hot-water; sodium chloride, brines	Oils and other hydrocarbons not corrosive to iron or steel; caustic solutions	Mild acids that would attack iron cylinders but not acid-resisting bronze

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 50 of 153

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

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

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

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

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

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

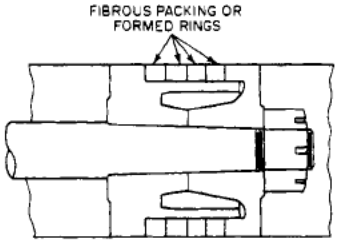


FIGURE 17 Body-and-follower piston

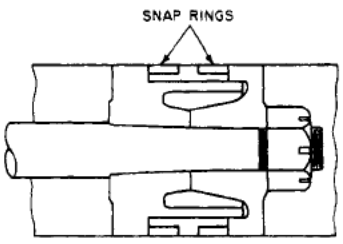


FIGURE 18 Body-and-follower piston with snap rings

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 51 of 153

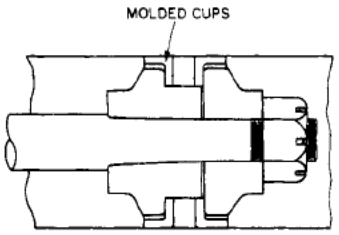


FIGURE 19 Body-and-follower piston with molded cups

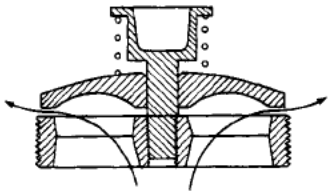


FIGURE 20 Stem-guided disk valve

installed in the center of the stuffing box with packing rings on both sides of it. A drilled hole is provided through which a lubricant, grease or oil, can be injected into the lantern ring from the outside of the stuffing box. At higher temperatures, approximately 500°F (260°C) or higher, a cooling water jacket is added to the outside of the stuffing box or as a spacer between the stuffing box and the liquid cylinder. The purpose of the cooling water jacket is to extend packing life by keeping the packing cool.

The liquid end valves of all direct-acting steam pumps are self-acting, in contrast to the mechanically operated slide valves in the steam end. The liquid end valves act like check valves; they are opened by the liquid passing through and are closed by a spring plus their weight.

Liquid end valves are roughly divided into three types: the disk valve for general service and thin liquids, the wing-guided valves for high pressures, and either the ball or the semispherical valve for abrasive and viscous liquids.

The valve shown in Figure 20 is typical of the disk type. This stem-guided design is commonly used in the cap-and-valve plate design. For hot-water boiler-feed and general service, the disk, seat, and stem are usually made of bronze, although other alloys may also be used. For lower temperatures and pressures, the disk may be made of rubber, which has the advantage of always forming a tight seal with the valve seat.

The wing-guided valve shown in Figure 21 is typical of the design use for high pressures. It derives its name from the wings on the bottom of the valve, which guide it in its seat. The beveled seating surfaces on the valve and seat tend to form a tighter seal than the flat seating surfaces on a disk valve. There is also less danger that a solid foreign particle in the liquid will be trapped between the seat and the valve. This type of valve is commonly made from a heat-treated chrome-alloy-steel forging, although a cast hard bronze and other materials may be used.

The ball valve, as its name suggests, is a ball that acts like a check valve. It is usually not spring-loaded, but guides and lift stops are provided as necessary to control its operation. The ball may be made of rubber, bronze, stainless steel, or other materials as service conditions require. The semispherical valve (Figure 22) is spring-loaded and can therefore be operated at higher speeds than the ball valve. Both the ball and the semispherical type

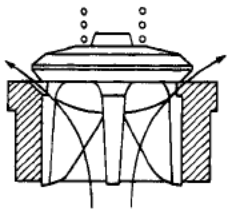


FIGURE 21 Wing-guided valve

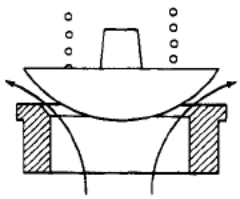


FIGURE 22 Semispherical valve

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 52 of 153

have the advantage of having no obstructions to flow in the valve seat (the disk valve seat has ribs and the wing-guided valve has vanes which obstruct the flow). The one large opening in the seat and the smooth spherical surface of ball and semispherical valves minimize the resistance to flow of viscous liquids. These types are also used for liquids with suspended solids because their rolling seating action prevents trapping of the solids between the seat and valve.

Plunger-Type Liquid Ends As mentioned previously, plunger-type pumps are used where dependability is of prime importance, even when the pump is operated continuously for long periods and where the pressure is very high. Cast liquid end plunger pumps are used for low and moderate pressures. Forged liquid end pumps (Figure 23), which are the most common plunger types, are used for high pressures and have been built to handle pressures in excess of 10,000 lb/in^2 (69 MPa).

Most of these designs have opposed plungers; that is, one plunger operating into the inboard end of the liquid cylinder and one into the outboard end. The plungers are solidly secured to inboard and outboard plunger crossheads. The inboard and outboard plunger crossheads are joined by side rods positioned on each side of the cylinder. With this arrangement, each plunger is single-acting; that is, it makes only one pressure stroke for each complete reciprocating cycle. The pump, however, is double-acting because the plungers are connected by the side rods.

PLUNGER PUMP LIQUID END MATERIALS The liquid cylinder of a forged liquid end plunger-type pump is most commonly made from forged steel, although bronze, Monel, chrome alloy, and stainless steels are also used. The stuffing boxes and valve chambers are usually integral with the cylinder (Figure 23), which is desirable for higher temperatures and pressures because high-temperature joints are minimized.

The liquid plungers may be made of a number of materials. The plungers must be as hard and smooth as possible to reduce friction and to resist wear by the plunger packing. Hardened chrome-alloy steels and steel coated with hard-metal alloys or ceramics are most commonly used.

The stuffing box packing used will vary widely depending upon service conditions. A soft, square packing cut to size may be used. However, solid molded rings of square, V-lip, or U-lip design are commonly used at higher pressures. Oil or grease is frequently injected into a lantern ring in the center of the stuffing box to reduce friction and reduce packing and plunger wear.

The liquid valves may be of any of the types or materials described above. However, the wing-guided valve with beveled seating surfaces is the most common because it is most suitable for high pressures.

DIRECT-ACTING STEAM PUMP PERFORMANCE

The direct-acting steam pump is a very flexible machine. It can operate at any point of pressure and flow within the limitations of the particular design. The speed of, and therefore the flow from, the pump can be controlled from stop to maximum by throttling the steam supply. This can be done by either a manual or an automatically operated valve in the steam supply line. The maximum speed of a particular design is primarily limited by the frequency with which the liquid valves will open and close smoothly. The pump will operate against any pressure imposed upon it by the system it is serving, from zero to its maximum pressure rating. The maximum pressure rating of a particular design is determined by the strength of the liquid end. In a particular application, the maximum liquid pressure developed may be limited by the available steam pressure and by the ratio of the steam piston and liquid piston areas.

STEAM PUMP CAPACITY

The flow to the discharge system is termed the pump *capacity*. The capacity, usually expressed in U.S. gallons per minute (cubic meters per minute), is somewhat less than the

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 53 of 153

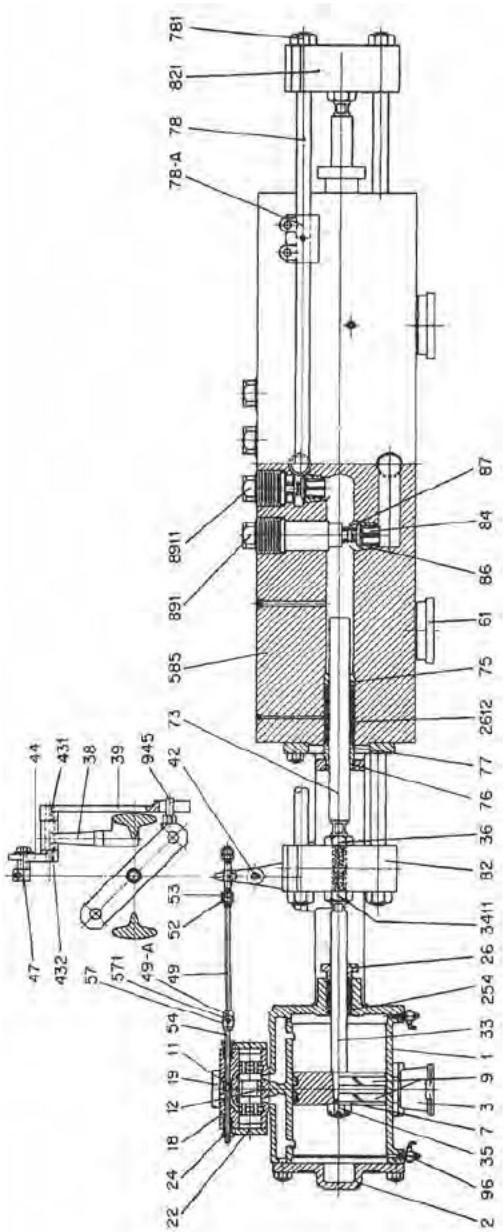


FIGURE 23 Simplex plunger pump with forged steel cylinder: (1) steam cylinder with cradle, (2) steam cylinder head, (3) steam cylinder foot, (7) steam piston, (9) steam piston rings, (11) slide valve, (12) piston valve, (18) steam chest, (19) steam chest cover, (22) steam chest head, (24) valve rod stuffing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

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 54 of 153

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

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

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

in SI units $$D = 1 \times 10^{-6} \times AS$$

where D = displacement, gpm (m³/min)
A = area of piston or plunger, in² (mm²)
S = piston speed, ft/min (m/min)
d = diameter of the liquid piston or plunger, in (mm)

For a duplex double-acting pump, D is multiplied by 2. This formula neglects the area of the piston rod. For very accurate calculations, it is necessary to deduct the rod area from the piston area. This is normally not done, and the resultant loss is usually considered part of the slip.

The slip also includes losses due to leakage from the stuffing boxes, leakage across the piston on packed-piston pumps, and leakage back into the cylinder from the discharge side while the discharge valves are closing. Slip for a given pump is determined by a test. For a properly packed pump, slip is usually 3 to 5%. As a pump wears, slip will increase, but this can be compensated for by increasing the pump speed to maintain the desired capacity.

PISTON SPEED

Although piston speed in feet per minute (meters per minute) is the accepted term used to express steam pump speed, it cannot easily be measured directly and is usually calculated by measuring the revolutions per minute of the pump and converting this to piston speed. One revolution of a steam pump is defined as one complete forward and reverse stroke of the piston. The relationship between piston speed and rpm is

in USCS units $$S = \frac{\text{rpm} \times \text{stroke}}{6}$$

in SI units $$S = 0.002 \times \text{rpm} \times \text{stroke}$$

where S = piston speed, ft/min (m/min)
rpm = revolutions per minute
stroke = stroke of pump, in (mm)

A steam pump must fill with liquid from the suction supply on each stroke, or it will not perform properly. If the pump runs too fast, the liquid cannot flow through the suction line, pump passageways, and valves fast enough to follow the piston. On the basis of experience and hydraulic formulas, maximum piston speeds that vary with the length of stroke and the liquid handled can be established.

Table 2 shows general averages of maximum speed ratings for pumps of specified stroke handling various liquids. Some pumps, by reason of exceptionally large valve areas or other design features, may be perfectly suitable for speeds higher than shown. From the table, it should be noted that piston speed should be reduced for viscous liquids. Unless the net positive suction head is proportionately high, viscous liquids will not follow the piston at high speeds because frictional resistance in suction lines and in the pump increases with viscosity and rate of flow. Pumps handling hot water are run more slowly to prevent boiling of the liquid as it flows into the low-pressure area behind the piston.

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 55 of 153

TABLE 2 Average maximum speed ratings

Stroke Length, in*	Piston speed, ft/min*: Cold Water; Oil to 250 SSU	Oil, 250–500 SSU	Oil, 500–1000 SSU	Oil, 1000–2500 SSU	Oil, 2500–5000 SSU	Boiler Feed 212°F (100°C)
3	37	35	33	29	24	22
4	47	45	42	36	31	28
5	53	51	47	41	35	32
6	60	57	53	46	39	36
7	64	61	57	49	42	39
8	68	65	61	53	45	41
10	75	72	67	58	49	45
12	80	77	71	62	52	48
15	90	86	80	69	57	54
18	95	91	85	73	62	57
24	105	100	94	81	68	63
36	120	115	107	92	78	72

*SI conversion factors: in × 25.4 = mm; ft/min × 0.3048 = m/min.

SIZE OF LIQUID END

The size of a steam pump is always designated as follows:

Steam piston diameter × liquid piston diameter × stroke

For example, a $\frac{1}{2} \times 5 \times 6$ steam pump has a $7\frac{1}{2}$-in (191-mm) diameter steam piston, a 5-in (127-mm) diameter liquid piston, and a 6-in (152-mm) stroke.

To determine the liquid piston diameter for a specified capacity, the following procedure is used. First, a reasonable stroke length is assumed and the maximum piston speed for this stroke and the type of liquid pumped is selected from Table 2. Then the desired capacity is increased by 3 to 5% to account for slip. The result is the desired displacement. Then for either a simplex or duplex pump, the liquid piston diameter can be calculated as follows:

In USCS units, for simplex pumps: $$d_l = 4.95\left(\frac{D}{S}\right)^{1/2}$$

For duplex pumps: $$d_l = 3.5\left(\frac{D}{S}\right)^{1/2}$$

In SI units, for simplex pumps: $$d_l = 1128.4\left(\frac{D}{S}\right)^{1/2}$$

For duplex pumps: $$d_l = 797.9\left(\frac{D}{S}\right)^{1/2}$$

where d_l = liquid piston diameter, in (mm)
D = displacement, gpm (m^3/min)
S = piston speed, ft/min (m/min)

Using the resultant liquid piston diameter, the next larger standard piston size is selected.

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 56 of 153

SIZE OF STEAM END

To calculate the size of the steam end required for a specific application, the basic principle of steam pump operation should be considered. A simple schematic of a steam pump is shown in Figure 24.

In order for the pump to move, the force exerted on the steam piston must exceed the force on the liquid piston that is opposing it. The force on the steam piston is the product of the net steam pressure and the steam piston area. The *net* steam pressure is the steam inlet pressure minus the exhaust pressure. The force acting on the liquid piston is the product of the net liquid pressure and the liquid piston area. The *net* liquid pressure is the pump discharge pressure minus the suction pressure or *plus* the suction lift. This may be expressed algebraically as follows:

$$P_s A_s > P_l A_l$$

where P_s = net steam pressure, lb/in² (bar)
A_s = steam piston area, in² (mm²)
P_l = net liquid pressure, lb/in² (bar)
A_l = liquid piston area, in² (mm²)

Because the pistons are circular, the squares of their diameters are directly proportional to their areas, and the above formula can be rewritten as

$$P_s d_s^2 > P_l d_l^2$$

where d_s = steam piston diameter, in (mm)
d_l = liquid piston diameter, in (mm)

In practice, it is necessary for the force on the steam piston to exceed the force opposing it on the liquid piston by a considerable amount. This is because of mechanical losses, which include stuffing box friction, friction between piston rings and cylinder of both liquid and steam ends, and the operation of the valve gear. These losses are determined by testing and are accounted for in size calculations by introduction of a mechanical efficiency figure. Mechanical efficiencies are expressed as a percentage, with 100% being a perfect balance of forces acting on the steam and liquid pistons as expressed in the previous formula. Because the efficiencies of two identical pumps may vary with stuffing box and piston ring packing tightness, the efficiencies published by manufacturers tend to be conservative.

With the mechanical efficiency factor inserted, the formula of forces becomes

$$P_s d_s^2 E_m = P_l d_l^2$$

where P_s = net steam pressure, lb/in² (bar)
d_s = steam piston diameter, in (mm)

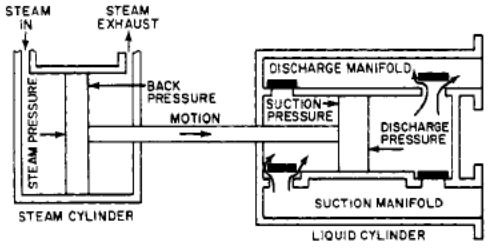


FIGURE 24 Schematic showing direction of forces on pistons

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 57 of 153

TABLE 3 Typical mechanical efficiency values

	Mechanical efficiency, %	
Stroke length, in (mm)	Piston Pump	Plunger pump
3 (76)	50	47
4 (102)	55	52
5 (127)	60	57
6 (152)	65	61
8 (201)	65	61
10 (254)	70	66
12 (305)	70	66
18 (457)	73	69
24 (610)	75	71

P_l = net liquid pressure, lb/in² (bar)
d_l = liquid piston diameter, in (mm)
E_m = mechanical efficiency, expressed as a decimal

This formula is commonly used to determine the *minimum* size of steam piston required when the liquid piston size has already been selected and the net steam and net liquid pressures are known. For this calculation, the formula is rearranged to the form

$$d_s = d_l\left(\frac{P_l}{P_s E_m}\right)^{1/2}$$

The efficiency of a long-stroke pump is greater than that of a short-stroke pump. Although mechanical efficiency varies with stroke length, any two pumps of the same size are capable of the same efficiency.

Table 3 shows typical mechanical efficiencies that can be used to determine required steam end size.

STEAM CONSUMPTION AND WATER HORSEPOWER

After determining the proper size of other pump types, the next concern usually is to calculate the maximum brake horsepower so the proper size of driver can be selected. With a steam pump, the next step is usually to determine the steam consumption. This must be known to ensure that the boiler generating the steam is large enough to supply the steam required by the pump as well as that required for all its other services.

To determine the steam consumption, it is necessary first to calculate the water horsepower as follows:

In USCS units $$whp = \frac{Q \times P_l}{1715}$$

In SI units $$\text{kW} = \frac{Q \times P_l}{36}$$

where whp = water horsepower
Q = pump capacity, gpm (m³/h)
P_l = net liquid pressure, lb/in² (bar)

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 58 of 153

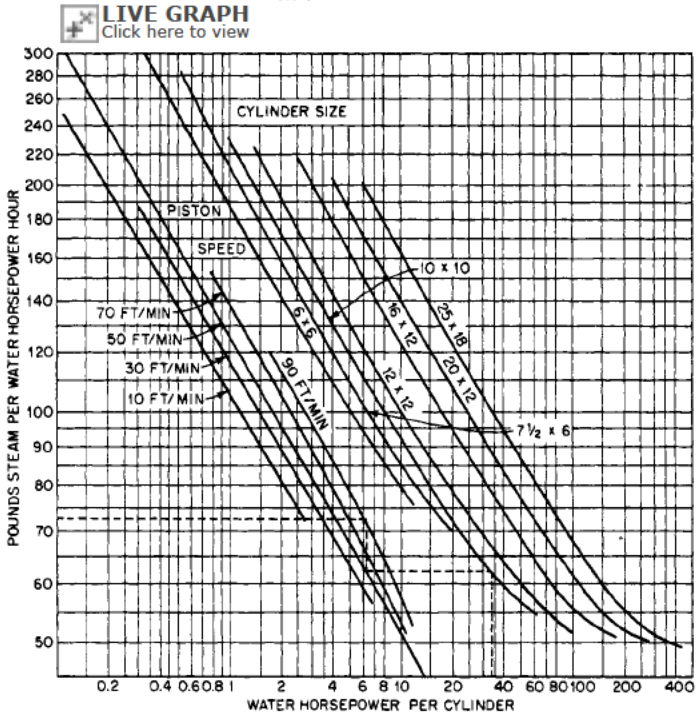


FIGURE 25 Approximate steam consumption for steam pumps (pounds of steam per water horsepower-hour = 1.644 kg of steam per water kilowatt-hour; water horsepower per cylinder = 1.341 wkW per cylinder; ft/s = 0.3047 m/s; in = 25.44 mm) (Hydraulic Institute Standards, 12th Edition, 1969—out of print)

A steam consumption chart (Figure 25) affords a means of quickly obtaining an approximate figure for the steam rate of direct-acting steam pumps. For duplex pumps, divide the water horsepower by 2 before applying it to the curves. These curves were made up on the basis of water horsepower per cylinder; if the above procedure is not followed, the results will be inaccurate.

Starting with the water horsepower per cylinder:

1. Move vertically to the curve for steam cylinder size.
2. Move horizontally to the curve for 50-ft/min (15.15-m/min) piston speed. This is the basic curve from which the other curves were plotted.
3. Move vertically to the piston speed at which the pump will run.
4. Move horizontally to the steam rate scale and read it in pounds per water horsepower-hour (kilograms per kilowatt-hour).
5. Multiply the result by total water horsepower to obtain the steam rate in pounds per hour (kilograms per hour).

For steam cylinders with diameters as shown, but with longer stroke, deduct 1% from the steam rate for each 20% of additional stroke. Thus, a 12 × 24 steam end will have a steam consumption about 5% less than a 12 × 12 steam end. For $5\frac{1}{4} \times 5$ and $4\frac{1}{2} \times 4$ steam ends, the 6 × 6 curve will give approximate figures. For cylinders of intermediate diameters, interpolate between the curves.

To correct for superheated steam, deduct 1% for each 10° of superheat. To correct for back pressure, multiply the steam rate by a correction factor equal to

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 59 of 153

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

where P = net steam pressure to drive pump, lb/in² (bar)
BP = back pressure, lb/in² (bar)

Direct-acting steam pumps have inherently high steam consumption. This is not necessarily a disadvantage, however, when the exhaust steam can be used for heating the boiler-feed water or for building heating or process work. Because these pumps can operate with a considerable range of back pressure, it is possible to recover nearly all the heat in the steam required to operate them. Because they do not use steam expansively, they are actually metering devices rather than heat engines and as such consume heat from the steam only as the heat is lost via radiation from the steam end of the pump. These pumps act, in effect, like a reducing valve to deliver lower pressure steam that contains nearly all its initial heat.

SUCTION SYSTEMS AND NET POSITIVE SUCTION HEAD

A majority of pump engineers agree that most operating problems with pumps of all types are caused by failure to supply adequate suction pressure to fill the pump properly.

The steam pump industry uses the term *net positive suction head required* (NPSHR) to define the head, or pressure, required by the pump over its datum, usually the discharge valve level. This pressure is needed to (1) overcome frictional losses in the pump, (2) overcome the weight and spring loadings of the suction valves, and (3) create the desired velocity in the suction opening and through the suction valves. The NPSHR of a steam pump will increase as the piston speed and capacity are increased. The average steam pump will have valves designed to limit the NPSHR to 5 lb/in² (0.3 bar) (34.5 MPa) or less at maximum piston speed.

If the absolute pressure in the suction system minus the vapor pressure is inadequate to meet or exceed the NPSHR, the pump will cavitate. Cavitation is the change of a portion of the liquid to vapor, and it causes a reduction in delivered capacity, erratic discharge pressure, and noisy operation. Even minor cavitation will require frequent refacing of the valves, and severe cavitation can cause cracked cylinders or pistons or failure of other major parts.

FLOW CHARACTERISTICS

The flow characteristics of duplex and simplex pumps are illustrated in Figure 26. The flow from a simplex pump is fairly constant except when the pump is at rest. However,

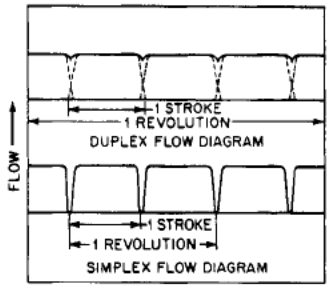


FIGURE 26 Flow characteristics of simplex and duplex pumps

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 60 of 153

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

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

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GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 61 of 153

SECTION 3.4
DISPLACEMENT PUMP PERFORMANCE, INSTRUMENTATION, AND DIAGNOSTICS

J. C. WACHEL
FRED R. SZENASI

The most common operational and reliability problems in reciprocating positive displacement pump systems are characterized by

- Low net positive suction head (NPSH), pulsations, pressure surge, cavitation, waterhammer
- Vibrations of pump or piping
- Mechanical failures, wear, erosion, alignment
- High horsepower requirements, high motor current, torsional oscillations
- Temperature extremes, thermal cycling
- Harsh liquids: corrosive, caustic, colloidal suspensions, precipitates (slurries)

Although any component in a pump system may be defective, most operational problems are caused by liquid transient interaction of the piping system and pump system or by purely mechanical interaction of the pump, drive system, foundation, and so on. This section discusses hydraulic and mechanical problems and suggests measurement and diagnostic procedures for determining the sources of these problems.

HYDRAULIC AND MECHANICAL PUMP PROBLEMS

Inadequate NPSH Net positive suction head available (NPSHA) is the static head plus atmospheric head minus lift loss, frictional loss, vapor pressure, and acceleration head available at the suction connection centerline.

Acceleration head can be the highest factor of NPSHA. In some cases, it is 10 times the total of all the other losses. Data from both the pump and the suction system are required

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 62 of 153

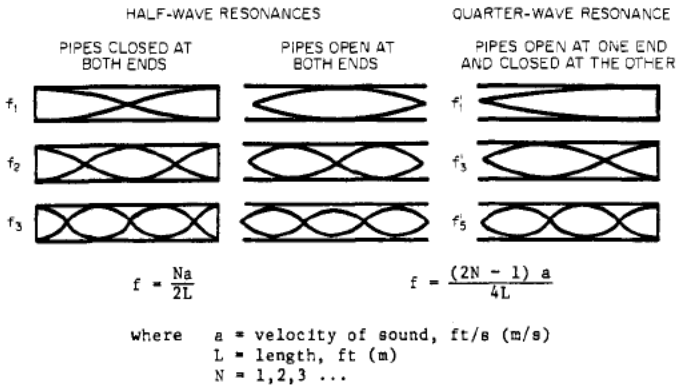


FIGURE 1 Organ pipe resonant mode shapes

to determine acceleration head; its value cannot be calculated until these data have been established. Inadequate NPSH can cause cavitation, the rapid collapse of vapor bubbles, which can result in a variety of pump problems, including noise, vibration, loss of head and capacity, and severe erosion of the valves and surfaces in the adjacent inlet areas. To avoid cavitation of liquid in the pump or piping, the absolute liquid static pressure at pumping temperatures must always exceed the vapor pressure of the liquid. The pressure at the pump suction should include sufficient margin to allow for the presence of pulsations as well as pressure losses due to flow.

Positive Displacement Pump Pulsations The intermittent flow of a liquid through pump internal valves generates liquid pulsations at integral multiples of the pump operating speed. For example, a 120-rpm triplex pump generates pulsations at all multiples of pump speed (2 Hz, 4 Hz, and so on); however, the most significant components will usually be multiples of the number of plungers (6 Hz, 12 Hz, 18 Hz, and so on). Resultant pulsation pressures in the piping system are determined by the interaction of the generated pulsation spectrum from the pump and the acoustic length resonances of liquid in the piping. For variable-speed units, the discrete frequency components change in frequency as a function of operating speed and the measured amplitude of any pulsation harmonic can vary substantially with changes in the location of the measurement point relative to the pressure nodes and antinodes of the standing wave pattern.

Piping System Pulsation Response Because acoustic liquid resonances occur in piping systems of finite length, these resonances will selectively amplify some pulsation frequencies and attenuate others. Resonances of individual piping segments can be described from organ pipe acoustic theory. The resonant frequencies of standing pressure waves depend upon the velocity of sound in the liquid being pumped, pipe length, and end conditions. The equations for calculating these frequencies are shown in Figure 1. All of the integral multiples (N) of a resonance can occur, and it is desirable to mismatch the excitation frequencies from any acoustical resonances. A 2:1 diameter increase or greater would represent an open end for the smaller pipe. Closed valves, pumps, or a 2:1 diameter reduction represent closed ends. For example, a 2-in (51-mm) diameter pipe that connects radially into two 8-in (203-mm) diameter volumes would respond acoustically as an open-end pipe.

Complex piping system responses depend upon the termination impedances and interaction of acoustical resonances and cannot be handled with simplified equations. An electroacoustic analog[1] or digital computer can be used for the more complex systems.

Velocity of Sound in Liquid Piping Systems The acoustic velocity of liquids can be determined by the following equation:

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 63 of 153

$$a = C_1\sqrt{\frac{K_s}{\text{sp. gr.}}} \tag{1}$$

where a = velocity of sound, ft/s (m/s)
C_1 = 8.615 for USCS units, 1.0 for SI units
K_s = isentropic bulk modulus, lb/in² (kPa)
sp. gr. = specific gravity

In liquid piping systems, the acoustic velocity can be significantly affected by pipe wall flexibility. The acoustic velocity can be adjusted by the following equation:

$$a_{\text{adjusted}} = a\sqrt{\frac{1}{1 + \frac{DK_s}{tE}}} \tag{2}$$

where D = pipe diameter, in (mm)
t = pipe wall thickness, in (mm)
E = elastic modulus of pipe material, lb/in² (kPa)

Pipe wall radial compliance can reduce the velocity of sound in liquid in a pipe as shown in Figure 2.[2]

The bulk modulus of water can be calculated with the following equation[3] for temperatures from 0 to 212°F (0 to 100°C) and pressures from 0 to 4.4×10^4 lb/in² (0 to 3 kbar*):

$$K_s = K_0 + 3.4P \tag{3}$$

where K_s = isentropic bulk modulus, kbar
K_0 = constant from Table 1, kbar
P = pressure, kbar
(1 kbar = 10^5 kPa = 14,700 lb/in²)

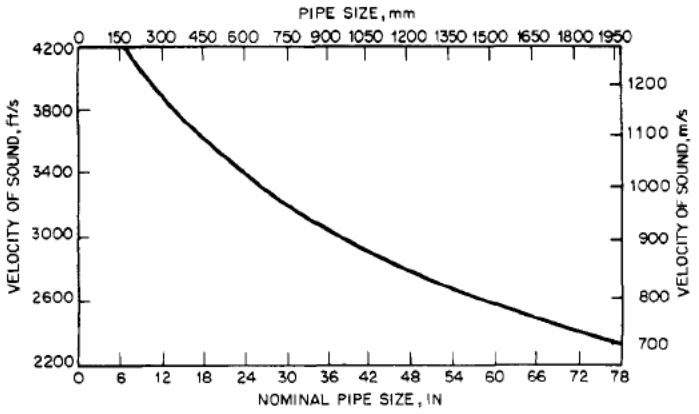


FIGURE 2 Velocity of sound in water at 14.6 lb/in² (1 bar), 60°F (15.6°C) versus nominal pipe size with 0.25-in (6.35-mm) wall thickness

*1 bar = 10^5 Pa.

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 64 of 153

TABLE 1 Constant K_0 for evaluation of isentropic bulk modulus of water from 0 to 3 kbar

Temperature, °C (°F)		Isentropic constant K_0, kbar[a]
0	(32)	19.7
10	(50)	21.0
20	(68)	22.0
30	(86)	22.7
40	(104)	23.2
50	(122)	23.5
60	(140)	23.7
70	(158)	23.7
80	(176)	23.5
90	(194)	23.3
100	(212)	22.9

[a]1 kbar = 14,700 lb/in².
Source: Reference 3.

The calculation of the isentropic bulk modulus of water is accurate to ±0.5% at 68°F (20°C) and lower pressures.[4] At elevated pressures (greater than 3 kbar) and temperatures (greater than 100°C), the error should not exceed ±3%.

The bulk modulus for petroleum oils (hydraulic fluids) can be obtained at various temperatures and pressures by using Figures 3 and 4, which were developed by the American

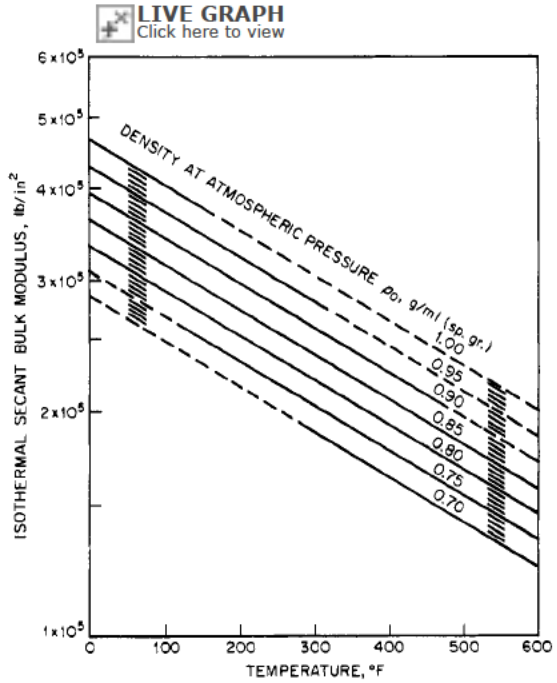


FIGURE 3 Isothermal secant bulk modulus at 20,000 lb/in² gage for petroleum oils (1 lb/in² = 6.895 kPa; °C = (°F − 32)/1.8).

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 65 of 153

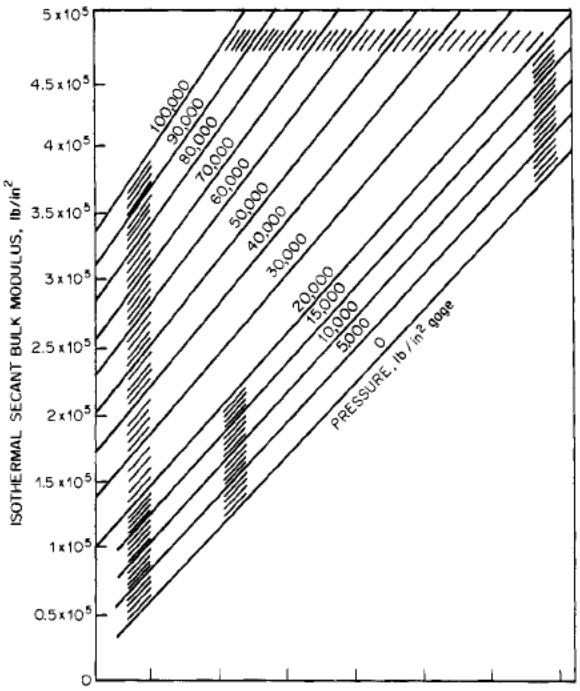


FIGURE 4 Pressure correction for isothermal secant bulk modulus for petroleum oils (1 lb/in² = 6.895 kPa)

Petroleum Institute (API).[4-7] Figure 3 relates density (mass per unit volume) and temperature to the isothermal secant bulk modulus at 20,000 lb/in² (137,900 kPa), and Figure 4 corrects for various pressures.

The isentropic tangent bulk modulus is needed to calculate the speed of sound in hydraulic fluids and can be readily obtained from Figures 3 and 4 as follows:

1. Read the isothermal secant bulk modulus at the desired temperature from Figure 3.
2. Using the value for isothermal secant bulk modulus obtained from Figure 3, go to Figure 4 and locate the intersection of the pressure line with that value. Move vertically to the pressure line representing twice the normal pressure. Read the adjusted isothermal secant bulk modulus for the double value of pressure.
3. Multiply the adjusted isothermal secant bulk modulus by 1.15 to obtain the value of the isentropic tangent bulk modulus (compensation for the ratio of specific heats).

The isothermal tangent bulk modulus has been shown to be approximately equal to the secant bulk modulus at twice the pressure[5] within ±1%. The relationship between isothermal bulk modulus K_t and isentropoic bulk modulus K_s is

$$K_s = K_t c_p / c_v \tag{4}$$

The value of c_p/c_v for most hydraulic fluids is approximately 1.15.

Pulsation Control Pulsation control can be achieved by judicious use of acoustic filters and side branch accumulators. Acoustic filters are liquid-filled devices consisting of volumes and chokes that use reactive filtering techniques to attenuate pulsations. Side

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 66 of 153

branch resonators are of two types: quarter-wavelength resonant stubs and gas-charged accumulators. (The terms *accumulators, dampeners,* and *dampers* are used interchangeably in the liquid filter industry.)

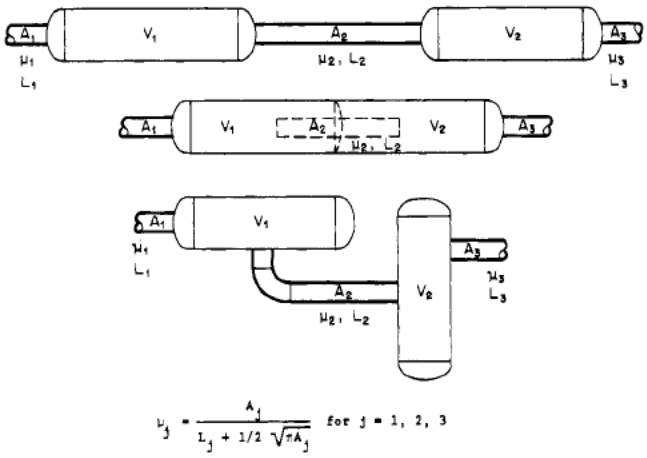
Acoustic Filters An acoustic filter consisting of two volumes connected by a small-diameter choke can significantly reduce the transmission of pulsations from the pump into the suction and discharge piping systems. The equations given in Figure 5 can be used to calculate the resonant frequency for a simple volume-choke-volume filter. The filter should be designed to have a resonant frequency no more than one-half the lowest frequency desired to be reduced, referred to as the cutoff frequency. Such a filter is called a low-pass filter because it attenuates frequencies above the cutoff frequency,

A special case for symmetric liquid-filled filters can be obtained by choosing equal chamber and choke lengths. This reduces the equation in Figure 5 to

$$f = \frac{ad}{\pi\sqrt{2}CD} \tag{5}$$

where d = choke diameter, in (mm)
L = chamber and choke length, ft (m)
D = chamber diameter, in (mm)

Normally, a good filter design will have a resonant frequency less than one-half the plunger frequency and will have a minimal pressure drop. For example, a triplex pump running at 600 rpm generates pulsations at all multiples of 10 Hz. The largest amplitudes



$$\mu_j = \frac{A_j}{L_j + 1/2\sqrt{\pi A_j}} \quad \text{for } j = 1, 2, 3$$

$$f = \frac{a}{2\pi}\sqrt{\frac{1}{2}\left[\frac{\mu_1+\mu_2}{V_1} + \frac{\mu_2+\mu_3}{V_2} \pm \sqrt{\left(\frac{\mu_1+\mu_2}{V_1} - \frac{\mu_2+\mu_3}{V_2}\right)^2 + \frac{4\mu_2^2}{V_1 V_2}}\right]}$$

FOR EQUAL VOLUMES, THE RESONANT FREQUENCY IS APPROXIMATELY:

$$f = \frac{a}{\sqrt{2}\,\pi}\sqrt{\frac{\mu_2}{V_1}}$$

FIGURE 5 Two-chamber resonator system with both ends open. V = volume, ft³ (m³); f = resonant frequency, Hz; L = choke tube length, ft (m); A = choke tube area, ft² (m²); a = acoustic velocity, ft/s (m/s); m = acoustic parameter

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 67 of 153

would normally be at 30 Hz, 60 Hz, 90 Hz, and so on. The filter resonant frequency should be set at 15 Hz or lower. For water, in which the velocity of sound is 3200 ft/s (976 m/s), and a volume bottle size inner diameter of 19 in (482.6 mm),

in USCS units
$$\frac{L}{d} = \frac{3200}{\pi\sqrt{2}\,(15)(19)} = 2.53\,\frac{\text{ft}}{\text{in}}$$

in SI units
$$\frac{L}{d} = \frac{976}{\pi\sqrt{2}\,(15)(482.6)} = 0.03\,\frac{\text{m}}{\text{mm}} \qquad (6)$$

If the choke diameter is selected to be 1.049 in (26.64 mm), the length of each volume bottle and choke tube is 2.65 ft (0.81 m). See also Reference 8.

Side Branch Accumulators Liquid-filled, quarter-wavelength, side branch accumulators reduce pulsations in a narrow frequency band and can be effective on constant/speed positive displacement pumps. However, in variable-speed systems, accumulators with or without a bladder can be made more effective by partially charging them with a gas (nitrogen or air) because the gas charge cushions hydraulic shocks and pulsations. If properly selected, located, tuned, and charged, a wide variety of accumulators (weight- or spring-loaded; gas-charged) can be used in positive displacement pump systems to prevent cavitation and waterhammer, damp pulsations, and reduce pressure surges.[9,10] Improper sizing or location can aggravate existing problems or cause additional ones. Typically, the best location for accumulators is as close to the pump as possible.

Gas-charged dampeners, or accumulators, such as those depicted in Figure 6, are most commonly used and can be quite effective in controlling pulsations. These devices are commercially available from several sources. Their location and volume and the pressure of the charge are important to their effectiveness. When gas-charged dampeners are used, the gas pressure must be monitored and maintained because the gas can be absorbed into the liquid. The system pressure can sometimes be lower than the gas charging pressure, such as on start-up; therefore, a valve should be installed to shut off the accumulator during start-up to eliminate gas leakage to the primary liquid. When the valve is closed, the accumulator is decoupled from the system and is not effective. Accumulators with integral check valves should be adjusted so pressure transients do not close the check valve and render the accumulator ineffective. Accumulators that have bladders (Figures 6b, d, and e) to separate the gas charge from the liquid have some distinct advantages, particularly if gas absorption is a problem. Accumulators with flexible bladders must be carefully maintained because failure of a bladder could release gas into the liquid system and could compromise the effectiveness of the dampener.

The in-line gas dampener (refer to Figure 6e) has a cylinder around the pipe containing a gas volume and bladder. The liquid enters the dampener through small holes in the circumference of the pipe and impinges upon the bladder, which produces the same acoustic effect as a side branch configuration.

It is not always possible to design effective pulsation control systems using simplified techniques. For complicated piping systems with multiple pumps, an electroacoustic

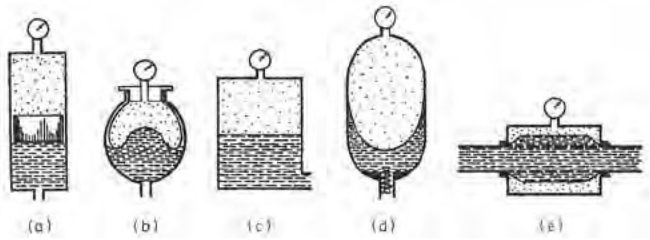


FIGURE 6 Types of accumulators: (a) piston, (b) diaphragm, (c) gas-charged, (d) bladder, (e) in-line

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 68 of 153

LIVE GRAPH
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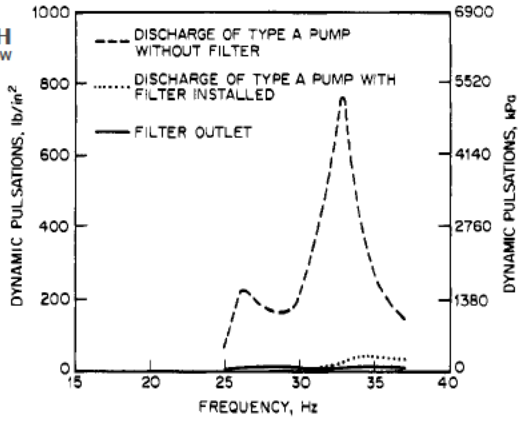


FIGURE 7 Effect of acoustic filter on pulsations

analog[1] is recommended for designing optimum filters or accumulators. This tool has become widely accepted for designing reliable piping systems for reciprocating liquid pumps and gas compressor units. The reduction in pulsations in a liquid pump system in a nuclear plant is shown in Figure 7. These results were obtained with a two-volume acoustic filter system designed with the electroacoustic analog. The volume diameter was 19.3 in (49 cm), and the length was 4 ft (1.2 m). The choke tube diameter was 0.8 in (2 cm), and its length was 7 ft (2.1 m). The speed of the triplex (2-cm) pump was 360 rpm, and the filter resonant frequency was set at 8.1 Hz for a velocity of sound of 4550 ft/s (1390 m/s).

Piping Vibrations When mechanical resonances are excited by pulsations, vibrations in the pump and piping can sometimes be 20 times higher than under off-resonant conditions. When the mechanical resonances coincide with the acoustic resonances, an additional amplification factor as high as 300 can be encountered.

Piping system mechanical natural frequencies can be calculated using simplified design procedures to provide effective detuning from known excitation sources. A nomogram (see Figure 8) for calculating the lowest natural frequency of uniform steel piping spans[11] can be used in designing piping systems and in diagnosing and solving vibration problems. For example, welded between two bottles, a 4-in (102-mm) pipe that is 10 ft (3 m) long and has an inner diameter of 3.826 in (97.18 mm) would have a mechanical frequency of 74 Hz. If the pipe was an equal-leg L bend ($L = 5$), the natural frequency would be 50 Hz.

To minimize piping vibration problems, all unnecessary bends (considering routing and thermal flexibility) should be eliminated because they provide a strong coupling point between pulsation excitation forces and the mechanical system. When bends are needed, use the largest enclosed angle possible and locate restraints near each bend. Piping should also have supports near all piping size reductions and at large masses (valves, accumulators, flanges, and so on). Small auxiliary piping connections (vents, drains, pressure test connections, and so on) should be designed so the mass of the valve and flange is effectively tied back to the main piping, thus eliminating relative vibration.

Diagnostics and Instrumentation The diagnosis of vibrations in positive displacement pumps should usually include dynamic pressure measurements in the cylinders and piping near the pump. These measurements can be obtained by the use of piezoelectric or strain gage pressure transducers. If cavitation or flashing is suspected, a pressure transducer capable of measuring static and large dynamic pressures should be used; even then, cavitation-produced pressure shocks may damage the transducer.

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 69 of 153

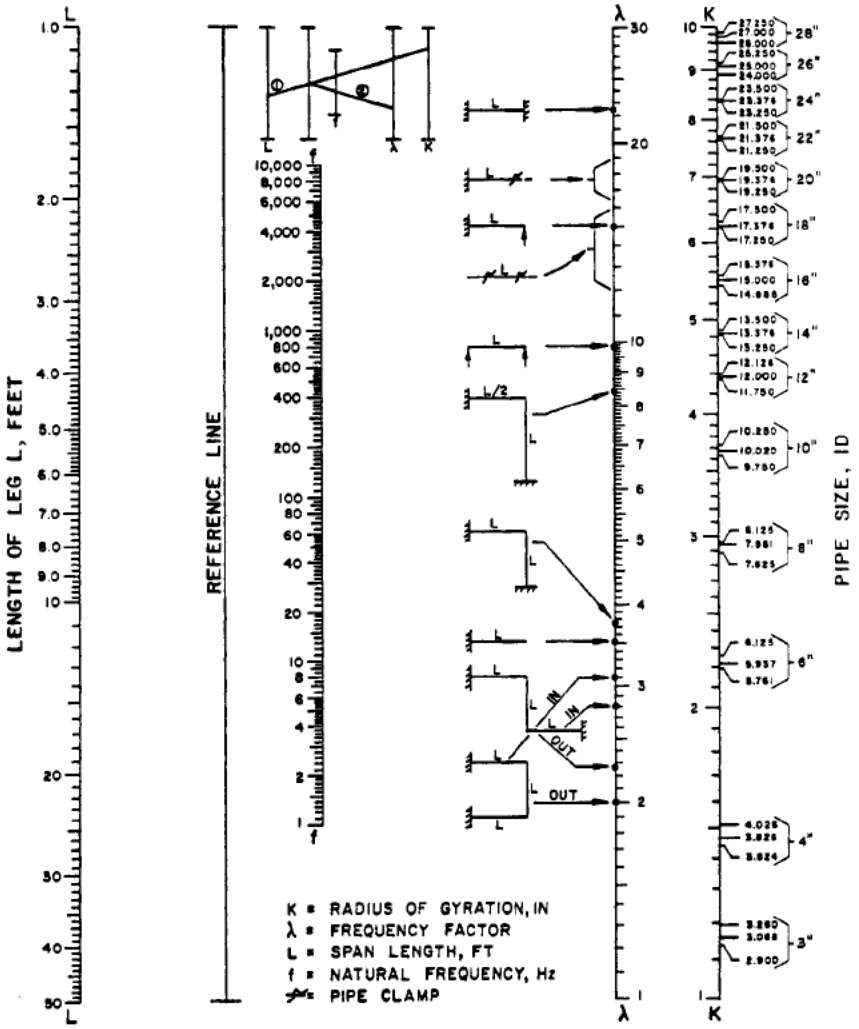


FIGURE 8 Natural frequency of uniform steel piping spans (1 ft = 0.3048 m; 1 in = 2.54 cm)

Accelerometers with low frequency characteristics may be used with electronic integrators to obtain accurate vibration displacement data from the pump case, cylinders, or piping. Similarly, velocity or seismic pickups may be employed. Maximum vibrations usually occur at the middle of piping spans and at unsupported elbows (out of plane).

Real-time analyzers and oscilloscopes may be used to display the resulting signals. A field example showing the diagnosis of cavitation at the pump suction using a strain gage diaphragm pressure transducer is given in the oscilloscope trace of Figure 9. The vapor pressure (gage) for this system was 25 lb/in^2 (172 kPa). Note that the negative half of the cycle is flattened when vapor pressure is reached and that very high amplitude pressure spikes are apparent. For liquids with dissolved gases, lower pulsation amplitudes can produce cavitation; however, the cavitation is usually less severe.

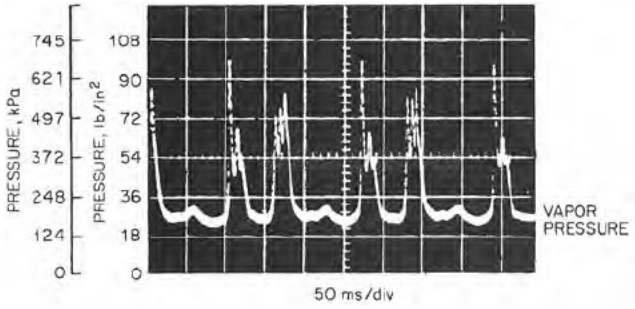


FIGURE 9 Complex wave data showing cavitation effects of pressure wave in a liquid piping system

Typical field pulsation data obtained on a three-plunger pump are shown in Figure 10, which is a frequency spectrum of the pulsations made by a real-time analyzer. Each spike represents a frequency multiple of running speed. Note that the third spike, representing the plunger frequency, has the largest amplitude; however, the components at one and two times pump speed are also significant, which means that these frequency components should be considered for pulsation control.

The vibration frequencies of the piping should be compared with pulsation frequencies to evaluate potential pulsation excitation of mechanical resonances. A check of the piping mechanical natural frequencies from the nomogram (refer to Figure 8) should be made to evaluate the possibility of a mechanical resonance. High vibrations produced by low-level pulsations at a particular frequency are indicative of a mechanical resonance, which can usually be corrected by additional piping restraints or snubbers. After the causes of the pulsations and vibrations are diagnosed, the techniques presented previously can be used to develop solutions.

Thermal Problems In pump systems with high thermal gradients, large forces and moments on the pump case can cause misalignment of the pump and its driver as well as pump case distortion resulting in vibrations, rubbing (wear), bearing failure, seal leakage, and so on. High stresses can be imposed on the piping, resulting in local yielding or damage to the piping restraints, snubbers, or support system. Misalignment problems commonly exhibit a high second-order component of the shaft vibrations. Proximity

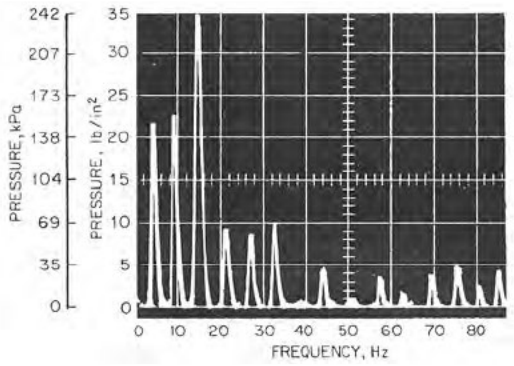


FIGURE 10 Typical field data recorded on a three-plunger pump

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 71 of 153

probes can be used at the bearings to measure movement of the shaft relative to the bearing centerline.

Diagnosis of Shaft Failures Pump and driver shafting can experience high stresses during start-up and normal operation because of the uneven torque loading of the positive displacement pumping action. Shaft failures are strongly influenced by the torsional resonances of the system, which are the angular natural frequencies of the system.

Torsional vibrations can be measured using velocity-type torsional transducers that mount on a stub shaft. Alternatively, they may be gauged by measuring the gear tooth passing frequency with a magnetic transducer or proximity probe and using frequency-to-voltage converters to give the change in tooth passing frequency (the torsional vibrational velocity). Spectral analysis of these signals defines the torsional amplitudes and natural frequencies. The stresses can be calculated by using the mode shape of the specific resonant natural frequency and combining all the torsional loads. Torsional natural frequencies, mode shapes, and stresses can be calculated by using either the Holzer technique or digital computer programs.[12]

Torsional problems can usually be solved by changing the coupling stiffness between the driver and pump or by using a flywheel in an effective location. The addition of a flywheel will tend to smooth the torque oscillations. Pumps with a greater number of cylinders and equal cylinder phasing usually operate more smoothly with lower shaft stresses.

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GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 72 of 153

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Positive Displacement Pumps: Controlled Volume, API Standard 675, 2nd ed., 1994, American Petroleum Institute, Washington, D.C. www.api.org.

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 73 of 153

SECTION 3.5
DISPLACEMENT PUMP FLOW CONTROL

WILL SMITH

FLOW CONTROL IN INDIVIDUAL PUMPS

An inherent characteristic of positive displacement pumping of relatively incompressible liquids is that flow rate is proportional to displacement rate and independent of pressure levels. The capacity of a centrifugal pump operating at constant speed varies from a maximum flow at no developed pressure to zero flow at a definite limiting pressure known as shutoff head. The *average* capacity of positive displacement pumps at constant speed is, within design limits of pressure, practically constant, even though flow rate pulsations do occur as individual displacements are forced into the discharge pipe.

Flow control of positive displacement pumps is accomplished by

- Changing the displacement rate
- Changing the displacement volume
- Changing the proportion of the displacement delivered into the piping system

Throttle Control in Direct-Acting Steam Pumps Direct-acting pumps are controlled by speed change, which is effected by throttling the flow of steam (or other motive gas) to the drive cylinder. The magnitude of excess force on the drive piston over that required for the driven, or pumping, piston to force pumpage into the piping system dictates the stroke rate and therefore the capacity of the pump. A sensor that detects the desired result of pumping (pressure, level, flow rate, and so on) may be used to modulate the steam throttle valve.

Speed Control in Power-Driven Pumps Speed modulation is the most common means of flow control for power-driven pumps. The most rudimentary speed control is intermittent (start-stop) operation. The average capacity over relatively long time periods depends upon the percentage of time the pump operates at 100% versus the percentage of time it

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 74 of 153

operates at zero flow. Of course, consideration must be given to the frequency of starts because electric motors may overheat if there is insufficient time for cooling after the inrush of starting current.

When continuous modulation of flow is desired, means of varying the driver speed or the ratio of speed reduction between the driver and pump are employed.

DRIVER SPEED CONTROL

1. Multispeed constant-torque motors offer limited steps in speeds.
2. Variable-frequency motor controls permit a wide range of fully modulated speeds but, at present, are limited in maximum horsepower (Section 9.2.2).
3. Direct current and wound rotor alternating current motors permit modulation of speeds within definite limits of torque capability (Section 9.2.2).
4. Gasoline and diesel engines provides speed variation capabilities within the limits where driver torque is adequate to satisfy the constant-torque requirements of the pump (Section 9.1.3).
5. Steam or gas turbines can operate over a limited speed range within their particular output torque limits (Sections 9.1.2 and 9.1.5).

SPEED CHANGER CONTROL Because *most* positive displacement pumps operate at significantly lower speed than *most* drivers, a speed reduction unit is coupled between driver and pump. Capacity control by varying the ratio of the speed reduction is quite convenient. A number of methods are used.

1. *Variable-ratio belt drives* change the speed by changing the pitch diameter of driver and driven sheaves in response to capacity requirements.
2. *Hydraulic torque converters* (hydrodynamic drives) vary the speed by regulating the *amount* of active drive fluid in the coupling. Because these converters relate torque to speed, they are sensitive to the driven torque requirement change that may derive from system pressure changes (Section 9.2.3).
3. *Hydroviscous drives* vary the slip between input and output shafts by adjusting the *distance* between elements in a viscous fluid environment (Section 9.2.3).
4. *Eddy-current couplings* also vary the slip between driver and driven elements but do so by varying the strength of a magnetic field (Section 9.2.1).

Variable Displacement Flow Control Rather than by speed control (changes in the number of displacements in a given period of time), flow may be varied by changing the *volume* displaced per stroke.

Variable strokes are generally limited to small metering or "controlled volume" pumps. Although large-capacity variable-stroke pumps have been built, the complexity of the mechanism involved and the development of other effective and economical means of flow control have precluded their use.

Changing Delivery to the System Another means of capacity control is *not* to vary the pump capacity but to alter the amount of pumpage delivered to the system. There are two popular means of doing this.

SUCTION VALVE UNLOADING Suction valve unloaders cause the displacement to be ineffective during the strokes for which the unloaders are activated. They command either full or zero delivery while the pump is kept running. Thus, current inrush problems from frequent motor starts are avoided. Because discharge pressure is not developed when suction valves are unloaded, energy consumption is held to a minimum.

A pump valve is unloaded by mechanically preventing the valve plate from returning to its seat. If the suction valve is held away from its seat, liquid will ebb and flow through the valve from suction header to pump cylinder and from cylinder to header during the stroking of the plunger or piston.

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 75 of 153

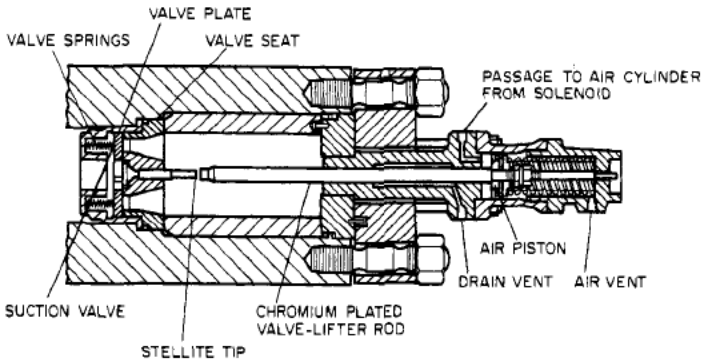


FIGURE 1 Synchronized unloading keeps the valve open through the use of a suction valve unloader

The unloader mechanism does not move the valve. The valve is lifted off its seat by the pressure differential created by normal pump operation. In synchronized unloading, the suction valve unloader (Figure 1) is retracted only when it is certain that the incoming flow will keep the valve open.

The unloader never resists pressure in the cylinder; it merely holds the valve plate against valve springs that had already been compressed and moves away from the valve while the springs are still compressed. This permits the valve to close when the plunger begins its discharge stroke.

For smooth transition between full flow and zero flow, the unloading and loading are accomplished in an immediate and identical sequence of the pumping order of the plungers. It is the function of the synchronized suction valve unloader distributor (Figure 2) to control unloading events in that proper and immediate sequence. Similarity can be drawn to the gasoline engine distributor in its function of assuring the proper and immediate sequence of firing of the spark plugs.

The distributor is driven directly by the crankshaft so its position is positively indexed to the stroking of the plungers. It provides an electrical signal to a solenoid valve, which

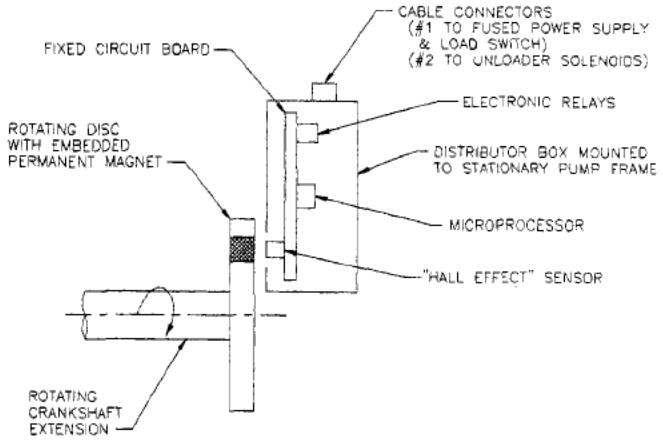


FIGURE 2 "Hall effect" microprocessor distributor system

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 76 of 153

admits motive fluid pressure to the unloader chamber when the suction valve to the particular cylinder has been opened. Then, in sequence, each of the remaining suction valves is retained so, in the first revolution of the pump after a control signals the distributor to initiate unloading, all suction valves are unloaded and the pump capacity is reduced to zero. When the control again signals for capacity, each suction valve is released, in sequence, during a single revolution of the crank.

While unloaded, the discharge valves do not function; they remain closed and serve as check valves against the pressurized discharge system. The suction valves remain open. Liquid enters from and is dispelled back into the suction manifold through the open suction valves. Therefore, no work is applied to the liquid, not even that required to open the suction and discharge valves. When the control calls for loading, the load is applied in steps equal to the number of plungers. Thus the pumping horsepower, as well as the liquid inertia in the pumping system, is changed in a number of steps equal to the number of plungers over the time span of one crank revolution.

The unloader consists of a spring-loaded pneumatic cylinder attached to the suction valve cover. The spring advances holding fingers that keep the valve away from its seat. Pneumatic pressure, applied through a solenoid valve in response to the distributor signal, creates a force on a piston that opposes the spring force and allows the suction valve to load (close). The system is normally arranged to fail in the unloaded position on either electrical or pneumatic pressure failure.

Distributor systems have evolved from a) direct mechanical linkages between reciprocating parts and the suction valves, to b) electromechanical interfacing, to c) electronic proximitor systems, to d) a "Hall effect" sensor signaling a microprocessor. The objective of each of these systems is to cause suction valve unloader solenoids to activate or deactivate in the appropriate sequence and at the appropriate position of the plunger during the suction stroke.

BYPASS CONTROL Ultimate delivery of pumpage to the system may be controlled by a discharge-to-suction bypass valve. Pump operation is continuous, with the possibility of modulating the flow to the system by regulating the portion "bypassed" to suction. A serious disadvantage is the full power utilization and full-load wear and tear even when at zero system capacity.

Evaluation of Flow Control Options Proper selection of the means of capacity control requires consideration of system control requirements and then evaluation of the options available to satisfy those requirements.

Precision, responsiveness, and degree of modulation required may limit options. First cost, maintenance requirements, and the effect on pump operating cost should then be weighed for viable options. To evaluate the effect of operating costs, it is necessary to determine (or reasonably assume) the expected pattern of operation throughout the capacity range. For example, if practically all operation except infrequent start-ups is to be at full pump capacity, little penalty should be assigned to the full power usage of a bypass valve or to the part-load losses inherent with hydroviscous or eddy-current drives.

If smooth modulation of flow is not essential, start-stop or suction valve unloading may be the preference. If smooth modulation *and* considerable operation at part capacity are necessary, variable-frequency motors or variable-pitch V-belt drives may be indicated.

Table 1 and Figure 3 provide some general guidelines. Each application should be evaluated on the merits of requirements and options.

FLOW CONTROL IN COMBINED DISPLACEMENT AND CENTRIFUGAL PUMPS

Systems that involve both centrifugal and reciprocating positive displacement pumps deserve some special consideration. Centrifugal pumps are often used as suction boosters to overcome acceleration head requirements peculiar to reciprocating pumps but are rarely used to supplement flow. Some unique characteristics of each type of pump that affect the other type must be considered in the design, operation, and control or the interrelated system.

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 77 of 153

TABLE 1 Comparison of several capacity control schemes for positive displacement pumps

Control Method	Degree of Modulation	First Cost	Operating Cost	Comments
		Directing-acting pumps		
Steam throttle	Full: 0 to 100%	Low	Low	Steam pressure required to balance liquid piston force and overcome breakaway friction. Steam volume throttled to produce desired capacity.
		Power pumps		
Start-stop	0 *or* 100%	Low	Low	Limited in frequency of starts because of temperature rise from inrush.
Multispeed motors	Steps dependent on motor winding	Medium	Low	Cost of motor controller switch gear motors must be assessed.
Variable frequency	Full: 0 to 100% +	High	Low	Limited by current-handling capacity of solid-state controller.
Direct current	Full: 0 to 100%	High	Medium	Drive is torque-speed sensitive, pump is torque-pressure sensitive at all speeds. Check drive for required torque at minimum and maximum speeds.
Wound rotor motors	Full: 0 to 100%	High	Low	Drive is torque-speed sensitive. Pump is torque-pressure sensitive at all speeds. Check drive for required torque at minimum and maximum speeds.
Combustion engines	Variable for limited range	High	Low	Torsional analysis required to avoid high torsional stresses.
Steam or gas turbines	Variable for limited range	Medium	Medium	Drive is torque-speed sensitive. Pump is torque-pressure sensitive at all speeds. Check drive for required torque at minimum and maximum speeds.
Hydraulic torque converter	Full: 0 to 100%	High	High	Low full speed efficiency.

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 78 of 153

TABLE 1 Comparison of several capacity control schemes for positive displacement pumps (*Continued*)

Control Method	Degree of Modulation	First Cost	Operating Cost	Comments
		Power pumps		
Hydroviscous speed control	Full: 0 to 100%	High	High at part load	Efficiency proportional to capacity. Loss to heat exchanger.
Fluid coupling	Full: 0 to 100%	Medium	High	Full-speed slip loss greater than hydroviscous. Loss to heat exchanger.
Magnetic (eddy) coupling	Full: 0 to 100%	High	High	Full-speed slip loss.
Suction valve unloaders	0 *or* 100%	Low	Low	Synchronization with suction stroke avoids start-stop and shock problems.
Bypass valve	Limited	Low	High	Uses full power at zero capacity with full valve wear.
Variable-pitch belt	Limited	Low	Low	Limited to belt drive horsepower.

Pertinent Positive Displacement Pump Characteristics

1. The discharge pressure produced is a function of the system requirement only and is independent of pump capacity.
2. The flow rate pulses between maximum and minimum values for each revolution of the crankshaft.

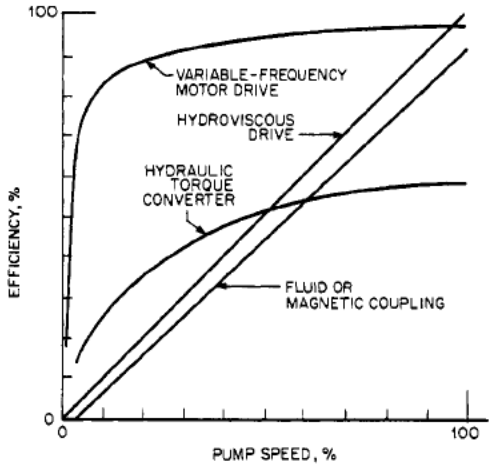


FIGURE 3 Relative efficiencies of speed control options

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 79 of 153

3. The pulsating flow imposes an acceleration head that adds to the net positive inlet pressure required (Section 3.1).
4. Being more energy-efficient, the positive displacement pump is normally the lead pump and the centrifugal pump is the supplement when operated in parallel.

Pertinent Centrifugal Pump Characteristics

1. The dynamic head (or pressure rise) produced is a function of pump capacity as well as of system requirements.
2. For satisfactory operation, the flow must be kept within a limited range of the best efficiency capacity (Section 2.1.3).
3. When used as a suction booster, the centrifugal pump must be designed so it cannot introduce air into the gas-intolerant reciprocating pump. (Gas in positive displacement pumps, like liquids in positive displacement compressors, may cause severe hydraulic and mechanical shock.)
4. Centrifugal pumps do not generate acceleration heads that impose on net positive suction head required. However, if a centrifugal pump is connected in parallel to a common suction line with a reciprocating pump, some of the system acceleration head loss from the positive displacement pump may affect the centrifugal pump.

Only two cases need to be considered: centrifugal pumps feeding (1) in series into reciprocating pumps to increase suction NPSHA to the reciprocating pump or (2) in parallel to augment the delivered capacity. It would be most unusual to encounter a positive displacement pump feeding into the suction of a centrifugal pump because of the high pressure that could be imposed on the centrifugal pump suction and because of the amplification of flow pulsations resulting from interaction of the characteristics of the two pumps, which could be deleterious to both pumps.

Series Operation, Suction Boost The flow rate is always determined by the positive displacement pump capacity and is independent of the centrifugal booster pump. The capacity is therefore controlled as with any positive displacement pump.

The purpose of the centrifugal pump is to supply sufficient pressure to satisfy the suction requirements of the positive displacement pump. The centrifugal pump must be sized to provide the maximum rate of flow of the positive displacement pump at a pressure high enough to meet the NPSH requirement of the latter.

With a flow-controlled positive displacement pump, speed control of the booster might be considered, as illustrated in Figure 4. The system pressure requirement can be satisfied by the positive displacement pump at all flow rates; therefore, total energy requirements may be reduced by using the less efficient centrifugal pump to develop no more than the required suction head.

If the centrifugal booster pump is run at constant speed, its share of the total head requirement will increase as system capacity is reduced.

Parallel Operation To increase the capacity of an existing pumping system, it is sometimes convenient to consider paralleling a positive displacement pump with a centrifugal pump. The interactions of the two pumps and the system must be considered and controlled for satisfactory operation.

Suction stabilizers and discharge dampeners, desirable for most reciprocating pump installations, become mandatory when a centrifugal pump and a positive displacement pump are connected in parallel. These devices are used to isolate, as much as possible, the effects of positive displacement pump flow pulsations from the sensitive centrifugal pump operating characteristics.

Centrifugal pumps in parallel with positive displacement pumps can supplement the system capacity only while they generate sufficient discharge pressure to satisfy the system requirement at the higher total flow. Undampened flow pulsations shift the centrifugal shutoff flow point relative to system flow at the frequency of the reciprocating

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 80 of 153

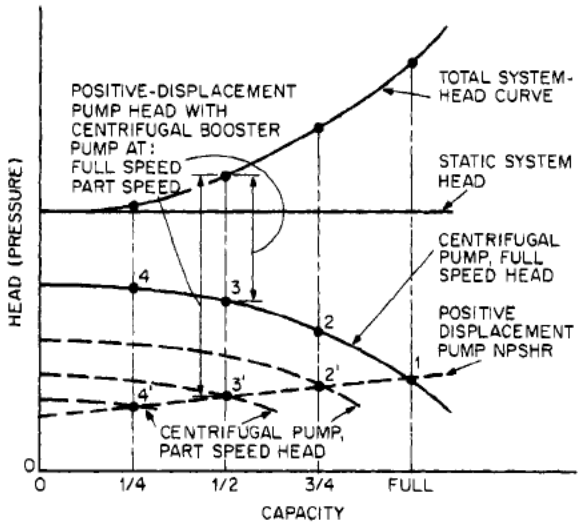


FIGURE 4 A centrifugal booster pump can satisfy the NPSH requirement of a positive displacement pump by operating either on its characteristic full speed curve 1-2-3-4 or with speed reduction along curve 1′-2′-3′-4′

pulsation. From the system standpoint, the total flow varies along curve 1-2-3-2-1 in Figure 5. From the centrifugal pump standpoint, it varies along its characteristic curve, 1′-2′-3′-2′-1′. The reciprocating pump operates along curve 1″-2″-3″-2″-1″.

For purely static system-head curves, the centrifugal pump theoretically would not "sense" the pulsating flow. For dynamic system heads, the centrifugal pump flow varies

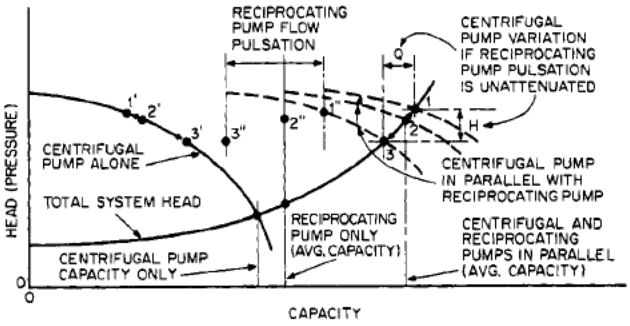


FIGURE 5 Parallel operation of undampened reciprocating and centrifugal pumps

1. The reciprocating pump will operate at a capacity corresponding to its speed regardless of its discharge pressure.
2. If the system head is only static (constant head), the centrifugal pump capacity is constant and the system flow variation corresponds to the reciprocating pump variation.
3. If the system head is partially dynamic (as illustrated), the centrifugal pump will experience flow variations.
   a. The flatter the centrifugal pump curve, the greater the capacity and smaller the head variations on the centrifugal pump.
   b. The flatter the system curve, the lower the capacity and greater the head variations on the centrifugal pump.

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 81 of 153

contracyclically with the reciprocating pump pulsations. The magnitude of the centrifugal pump flow variation due to the positive displacement pump pulsations is a function of the system head curve, the centrifugal pump head-capacity curve, the degree of flow pulse imposed on the centrifugal pump discharge header, and the liquid bulk modulus (or compressibility).

FLOW CONTROL IN POSITIVE DISPLACEMENT PUMPS IN SERIES

Whenever positive displacement pumps are installed in series (such as for multiple-station pipeline service), some variable-capacity capability is mandatory. Otherwise, any variation in capacity between series pumps or series groups of pumps would result in disastrous interstation pressures.

The flow is determined by the first pump in the line. Subsequent pumps should be equipped with capacity controls responsive to their inlet pressures. The preferred control is a continuous-modulation type, such as variable speed, unless large reservoir capacity is available between pumps.

Recirculation from discharge to suction through a throttling bypass valve may be used, at the expense of energy, to match the delivery from downstream pumps to upstream pumps. Excessive bypass flow in relation to through flow can result in significant temperature rise of the liquid.

When downstream pump capacity is less than the upstream delivery, the interpump pressure will rise, indicating a need for increased capacity. When the downstream pump capacity exceeds the upstream delivery, the downstream pump will "draw down" the interpump pressure to the point where net positive inlet pressure difficulties are ensured.

Reservoir accumulators between pumps permit a limited degree of mismatch in capacities, allowing the use of incremental capacity controls, such as suction valve unloading and intermittent pump bypass.

In any event, effective discharge dampeners on the upstream pump and suction stabilizers on the downstream pump are necessary to prevent the inherent pressure pulsations and control modulation variances from adversely affecting operation.

FLOW CONTROL IN POSITIVE DISPLACEMENT PUMPS IN PARALLEL

Because the capacity of positive displacement pumps is virtually independent of head, there is no problem from the head-capacity characteristics in parallel operation—even for pumps of different sizes.

Particular attention to suction acceleration head requirements is necessary when more than one pump is taking suction from a common suction pipe. Because it cannot be assured that two pumps will not, at some point, have simultaneous suction strokes, the acceleration head requirements of each pump must be added for that portion of the suction line that furnishes the total flow. This suggests extra large suction lines, individual ones from the liquid source, or suction stabilizers sized for the total flow with individual outlet connections for each pump.

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 82 of 153

SECTION 3.6
DIAPHRAGM PUMPS

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Diaphragm pumps are a class of displacement pumps featuring flexible membranes in combination with check valves that are used to move fluids into and out of pumping chamber(s). These pumps are used extensively in transfer and metering applications requiring flows of up to 300 gallons per minute (1150 liters per minute). They are quite versatile, handling a wide variety of fluids including chemicals, dry powders, food additives, glues, paints, pharmaceutical products, slurries, tailings, and wastewater. A distinguishing feature of all diaphragm pumps is the absence of seals or packing, meaning they can be used in applications requiring zero leakage. There are three main categories of diaphragm pumps: 1) mechanically driven, 2) hydraulically driven, and 3) air-operated.

MECHANICALLY DRIVEN DIAPHRAGM PUMPS

Many industries are served by mechanically driven diaphragms pumps. They are used in construction, chemical, and water treatment applications.

Construction Industry Mechanically driven diaphragm pumps are widely used in the construction industry for dewatering applications where pumps may ingest rocks or other debris. A popular make of this type of pump contains a spring on the plunger rod (see Figures 1 and 2). If the operating pressure exceeds the maximum recommended pumping pressure, the spring compresses and does not move the diaphragm. The spring can compress and thus keep a rock from being pushed through the wall of the pumping chamber or cause the drive mechanism to fail.

In single-diaphragm pumps, the pumped liquid can have a lot of inertia if the suction and discharge lines are relatively long. A simple accumulator on the suction (inlet) side of the pump enables the pump to draw liquid from the accumulator while it simultaneously draws liquid through the suction line.

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 83 of 153

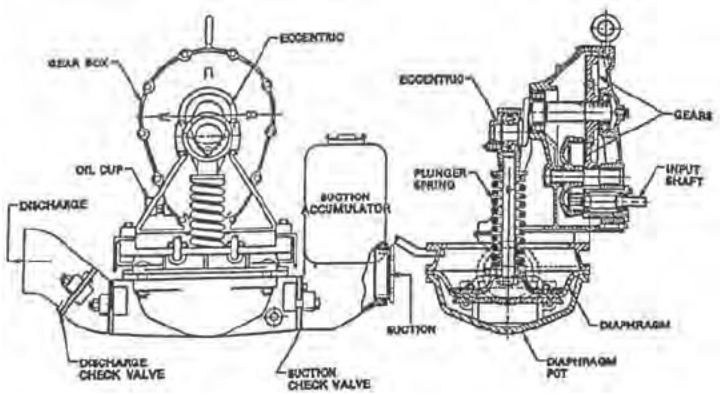


FIGURE 1 Cross-section of a mechanically driven single-diaphragm pump for the construction industry (Gorman-Rupp)

During the discharge stroke, the accumulator can refill with liquid from the suction line. If the discharge line from the pump is relatively long, the inertia of the liquid can be great, as mentioned earlier, and can impose severe loads on the diaphragm and drive mechanism. The spring on the plunger rod can absorb some of the drive energy early in the discharge stroke and "give it back" during the latter part of the discharge stroke, greatly reducing the inertia loading on the diaphragm and drive mechanism.

Mechanically driven diaphragm pumps in the construction industry operate by a reciprocating plunger, usually secured to plates on both sides of the diaphragm. The diaphragms are customarily fabric-reinforced elastomers (usually synthetic rubbers) similar in many ways to the fabric-reinforced materials used in pneumatic tires. The diaphragms are normally molded with a convoluted section between the central clamped area and the clamped periphery. This convoluted section permits longer strokes than would be possible otherwise.

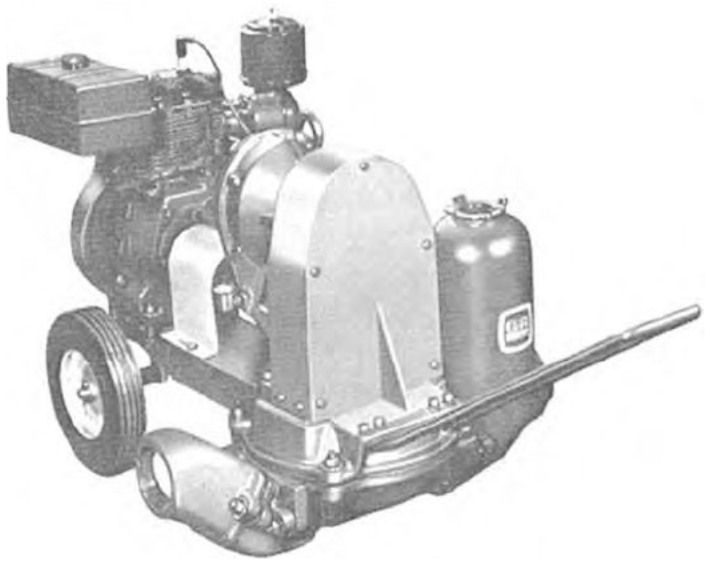


FIGURE 2 Mechanically driven single-diaphragm pump, engine-powered (Gorman-Rupp)

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 84 of 153

These pumps are sometimes duplexed so that the reciprocating means acts alternately on two diaphragms with one on a suction stroke, while the other is on a discharge stroke and vice versa. A connector called a *walking beam* is pivoted between two diaphragms. As one diaphragm is pushed down on a discharge stroke, the other diaphragm is simultaneously pulled up on a suction stroke. The pumping chambers with inlet and outlet check valves are manifolded together to a common inlet and a common outlet. The principle advantage of the duplex diaphragm pump is its more constant flow (two pressure pulsations per cycle).

Mechanically driven diaphragm pumps are used in the construction industry for dewatering foundations and cofferdams, as well as in sewage treatment plants for pumping lime slurries. They are normally limited to 50 ft (15.2 m) of differential head and are capable of suction lifts of as much as 25 ft (7.6 m).

Chemical and Water Treatment Industries Another type of mechanically driven diaphragm pump is used for the injection or transfer of chemicals into process streams at pressures up to 250 lb/in^2 (17 bar). These pumps are designed to enable the easy adjustment of their capacities, so precise volumes of chemicals can be injected. Typically, capacities can be adjusted through a 20:1 range. Injection repeatability is generally plus or minus 3%.

A wide range of chemicals can be handled. Wetted materials include PVC, PVDF, Polypropylene, 316SS, Alloy 20, and Alloy C22. Diaphragms are PTFE or PTFE with elastomeric backing. Ball type check valves are usually employed.

Applications for this type of pump include the injection of acids and bases for pH control, biocides, chlorination, coagulants, and fertilizers. There are two basic configurations for pumps in this class: *electromagnetic pumps* (solenoid) and *motor-driven pumps*.

Electromagnetic (electronic) pumps (see Figure 3) are used in a variety of low-power applications with flows from 0.026 to 26 gallons per hour (0.1 to 100 liters per hour) at pressures up to 250 lb/in^2 (17 bar). These metering pumps employ an electronic control circuit that pulses an electromagnet that, in turn, generates the linear motion of an armature-shaft-diaphragm assembly. Each electronic pulse results in one discharge stroke of the pump. At the end of the stroke, a set of springs returns the diaphragm assembly to its initial position, drawing more fluid into the pump chamber in preparation for the next stroke.

These pumps are inherently safe, as they can be run indefinitely in the stalled condition without damage to the pump or overpressuring most systems. An additional feature of certain electronic pumps is the regulation of pulse strength through electronic power control, which leads to smoother fluid injection. Capacity is usually controlled by adjusting

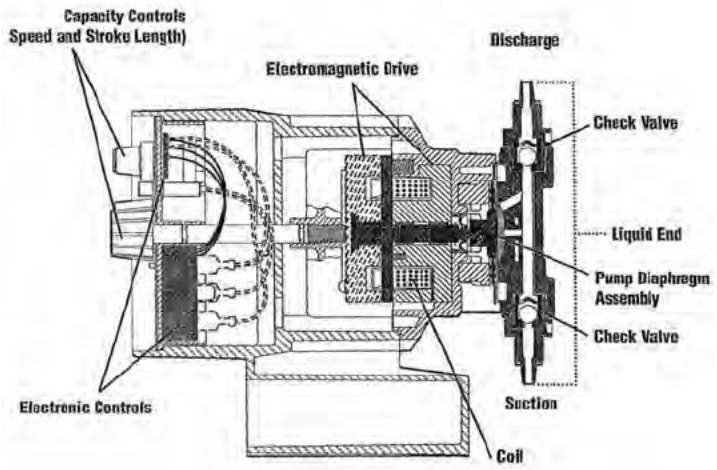


FIGURE 3 Electromagnetic diaphragm pump (Milton-Roy, subsidiary of Sundstrand Corp.)

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 85 of 153

the stroke rate, but the stroke length can also be adjusted. Combining these adjustments provides a wide range of outputs.

Advances in the electronic controls have led to the capability to control output manually from 4 to 20 mA process signals, digital pulses from external sources (such as flow meters), or serial data communications signals from computers. Yearly maintenance is recommended for low-pressure applications, but as pressures rise, diaphragms and check valves will need to be replaced more frequently.

Motor-driven, mechanically actuated diaphragm pumps are used in applications for flows from 2 to 300 gallons per hour (approximately 10 to 1000 liters per hour) again at pressures up to 250 lb/in^2 (17 bars). Since motor-driven, mechanically actuated diaphragm pumps are positive displacement pumps, capacities cannot be adjusted by the use of a throttling valve.

Three techniques are used in the industry to adjust motor-driven pump capacities. Some designs employ the use of a "mechanical lost motion" mechanism in which an adjustable mechanical stop interrupts the spring-loaded crosshead from following the cam for a portion of the stroke. This decreases the effective stroke length (see Figure 4). Other designs employ the use of an adjustable crank machined with a "variable eccentricity." The adjustment of the crank's position changes the pump's effective stroke length (see Figure 5). The third design technique is the use of AC or DC variable speed motors.

At higher flow rates, attention should be paid to the system design to ensure the proper handling of pressure pulsations, due to the acceleration and deceleration of the process fluid in the lines. This is especially true for pumps having a mechanical lost motion configuration, due to the diaphragm's rapid starts and stops. In the higher flow systems, pulsation dampeners are frequently employed in the discharge lines and occasionally need to be used in the suction lines to control pulsations. When discharge pressures are low, backpressure valves are employed to provide a system pressure sufficient to decelerate the fluid in the suction line at the end of every suction stroke. If the flow of the process fluid in the suction line has not stopped by the beginning of the discharge stroke, the accuracy of the injection is diminished. Yearly maintenance is recommended for low-pressure applications, but as pressures rise, diaphragms and check valves will need to be replaced more frequently.

Other Mechanically Driven Diaphragm Pumps Automotive fuel pumps are usually of the diaphragm type. The diaphragm is moved mechanically by a cam on a suction stroke

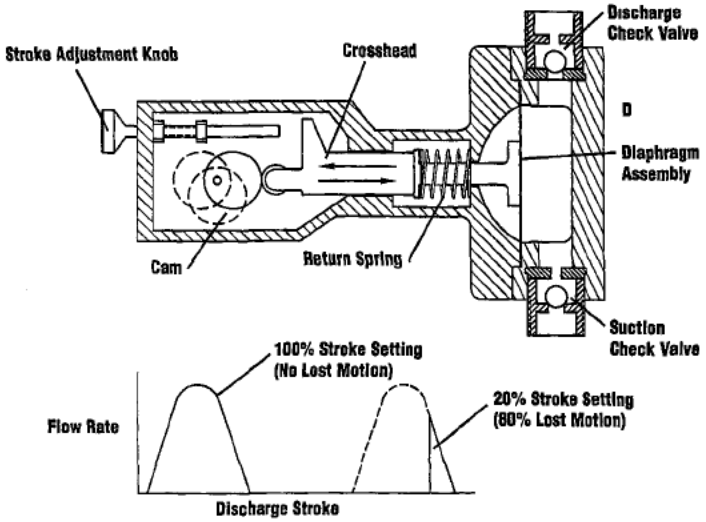


FIGURE 4 Diaphragm of a typical mechanical lost motion mechanism (Milton-Roy, subsidiary of Sundstrand Corp.)

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 86 of 153

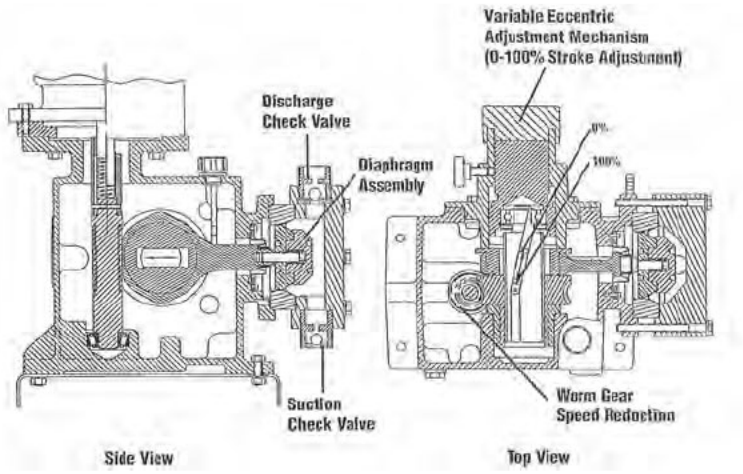


FIGURE 5 Cross-section of a variable eccentric mechanically driven diaphragm pump (Milton-Roy, subsidiary of Sundstrand Corp.)

and is returned by a spring for the discharge stroke. Thus, the spring determines the discharge pressure and provides a nearly constant pump pressure to the carburetor regardless of the pump (engine) speed or the rate of fuel consumption. Hand-operated diaphragm pumps are used extensively as bilge pumps on sailboats where the loss of power is a major concern and hand operation is essential.

HYDRAULICALLY DRIVEN DIAPHRAGM PUMPS

Hydraulically driven diaphragm pumps are used in applications for the transfer or injection of chemicals into process streams at pressures up to 7500 lb/in² (approximately 500 bar). Because the diaphragm is pressure-balanced, the stresses in the diaphragms are low. Therefore, these pumps tend to require minimal maintenance. The pump's capacities can be adjusted to match the specific process requirement by adjusting the effective stroke length or stroking speed of the pump. Effective stroke lengths are adjusted by either a hydraulic lost motion, a mechanical lost motion, or by varying the eccentric's offset. The repeatability of the injected flow is plus or minus 1% or better.

Applications range from 0.26 to 26,000 gallons per hour (1 to 100,000 liters per hour). At flows above 26 gallons per hour (100 liters per hour), most pump models employ capacity adjustments based on variable eccentric or variable speed technology to avoid significant pressure spikes due to the rapid acceleration and deceleration of the fluid in the pipes.

As with the mechanical diaphragm pumps, a wide range of chemicals can be handled. Wetted materials include PVC, Polypropylene, PVDF, 316 SS, Alloy 20, Alloy C-22, Titanium, and Inconel. Diaphragms for pressures up to 4350 lb/in² (300 bar) are typically composed of PTFE or PTFE with an elastomeric backing. Diaphragms above 4350 lb/in² (300 bar) are typically 316 SS, Alloy C, or PEEK. Optional features include fluid temperature control jackets, diaphragm rupture detection capabilities, and remote diaphragm head designs. Typical applications include the injection of acids and bases for pH control, corrosion inhibitors, methanol, coagulants, primary process blending, process slurries, and drag reducers. Three types of liquid ends are used: the disc diaphragm, shown in Figure 6, the tubular diaphragm in Figure 7, and the high performance diaphragm in Figure 8.

The disc diaphragm pump is equipped with process-side and suction-side restraining plates to prevent overdisplacement of the diaphragm during system upsets. When the

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 87 of 153

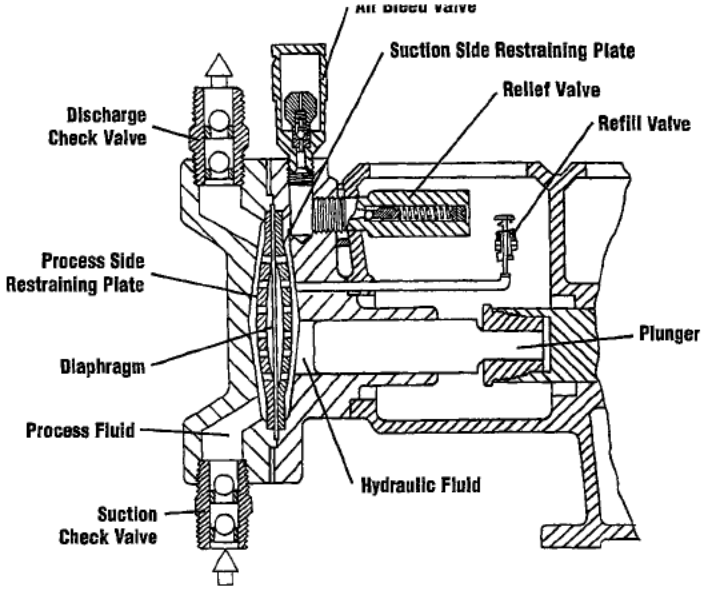


FIGURE 6 Diagram of a disc diaphragm pump (Milton-Roy, subsidiary of Sundstrand Corp.)

diaphragm reaches the suction-side restraining plate, the hydraulic oil pressure drops, causing the refill valve to open and replenish the oil. When the diaphragm hits the process-side restraining plate, the hydraulic pressure rises, causing the relief valve to open, venting some oil. The fluid volume between the restraining plates is typically 150% of the maximum displaced volume of the pump. Therefore, the diaphragm does not contact both restraining plates during the same stroke.

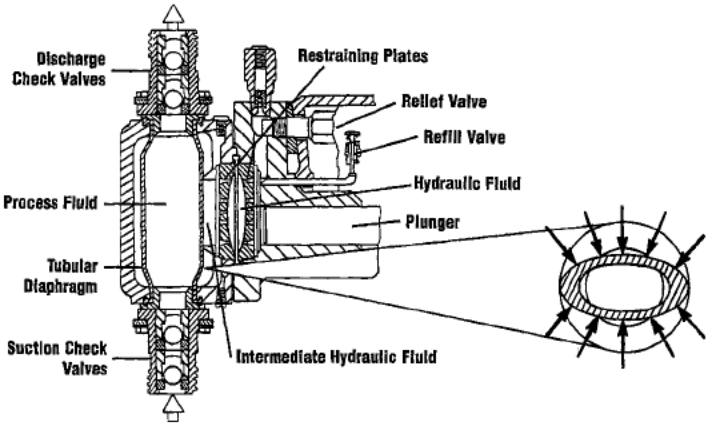


FIGURE 7 Diagram of a tubular diaphragm configuration (Milton-Roy, subsidiary of Sundstrand Corp.)

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 88 of 153

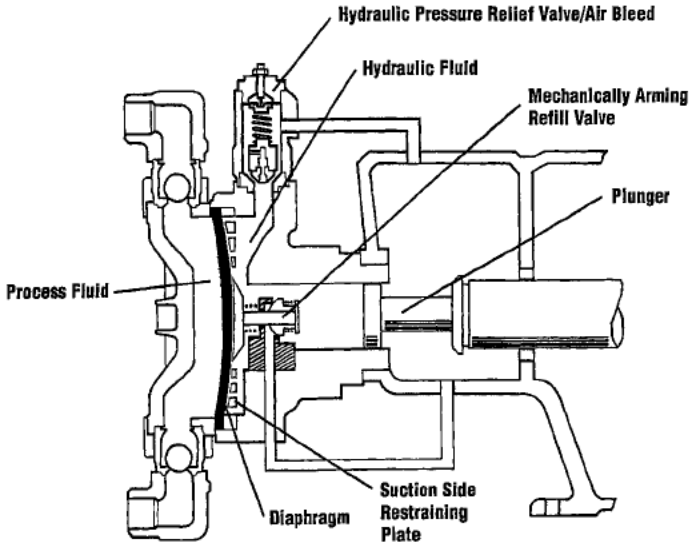


FIGURE 8 Diagram of a high performance diaphragm design (Milton-Roy, subsidiary of Sundstrand Corp.)

The tubular diaphragm configuration is a variation of the disc diaphragm design. A diaphragm shaped in the form of a tube is placed in a chamber in front of the disc diaphragm assembly. This design eliminates the process fluid flowing through the front-restraining plate, reducing viscous losses and wear in case of slurries. The chamber must be filled with a precise amount of hydraulic fluid to avoid overdisplacing the tube.

The high-performance diaphragm configuration eliminates the use of a process-side restraining plate providing the throughflow performance of a tubular design while eliminating the possibility of overdisplacing the tube during startup and maintenance. With a mechanically arming, pressure-sensitive refill valve, the hydraulic fluid can only be replenished when the diaphragm is in the most rearward position. This eliminates the possibility of overfilling the hydraulic chamber and therefore overdisplacing the diaphragm during system upsets (blocked suction or discharge lines).

Most problems with hydraulic diaphragm pumps occur due to incorrect system designs. Pressures above 9 lb/in^2 (0.6 bar) should be maintained in the pump diaphragm heads during the suction stroke to stop vapor buildups in the hydraulic or process-side cavities. Pressures at 3 lb/in^2 (0.2 bar) can be handled in some applications with modified designs and special hydraulic fluids. NPSH calculations should include viscous losses in the check valves and contour plates (if so equipped).

In addition, since hydraulic diaphragm pumps are reciprocating machines, acceleration losses also have to be considered. Peak acceleration/deceleration losses occur at the beginning and end of the stroke, while peak viscous losses occur at midstroke. The losses are not additive. The manufacturer should be contacted to provide guidance in performing these calculations.

As with mechanically driven diaphragm pumps, at higher flows, pulsation dampeners should be considered to ensure the proper handling of pressure pulsations due to the acceleration and deceleration of the process fluid in the lines. This is especially true for pumps having a mechanical lost motion configuration due to the diaphragm's rapid starts and stops. When discharge pressures are low, backpressure valves are employed to provide a system pressure sufficient to decelerate the fluid in the suction line at the end of every suction stroke. If the flow of the process fluid in the suction line has not stopped by the beginning of the discharge stroke, the accuracy of the injection is compromised.

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 89 of 153

AIR-OPERATED DIAPHRAGM PUMPS (AODPS)

In general, diaphragm pumps of all types are sealless, have no dynamic seals or packing, are self-priming, and have an infinitely variable flow rate and pressure rate within the pressure and capacity ranges of the pump. *Air-operated diaphragm pumps* (AODPs) can also run dry indefinitely, and the discharge can be throttled to zero flow indefinitely.

The most common types of AODPs are the double-diaphragm pumps (duplex pumps). These contain two diaphragm chambers and two flexible diaphragms. The diaphragms are connected to each other through a connecting rod and are clamped at the outer edges of the diaphragm. The shaft-connected diaphragms move in the same linear direction simultaneously. Compressed air directed to the back side of the left diaphragm moves both diaphragms to the left, while air is exhausted to the atmosphere from the back side of the right diaphragm. After completion of a stroke, an air distribution valve directs compressed air from the supply to the back side of the right diaphragm and exhausts air to the atmosphere from the left chamber. This continuous reciprocating motion, along with properly operating internal check valves, creates an alternating intake and discharge of pumped liquid into and out of each chamber that results in a nearly continuous pumping action from the combined chambers.

A diaphragm pump air motor contains an air distribution valve that shifts positions at the end of each stroke of the pump. The air distribution valve alternately directs supply air pressure to one chamber and exhausts the other. Air motors often use a two-stage valve to control the reciprocating motion of the pump (see Figure 11). The pilot valve supplies a pilot air pressure signal to the air distribution valve throughout the entire stroke of the pump, even though pressure oscillations in the pumping system may occur. The pilot valve is not connected directly to the diaphragm's connecting rod, which provides a "deadband" to prevent the power valve from erroneously shifting just after the end of each stoke. The two views contained in Figure 11 depict the position of the moving parts just before the pilot valve is moved by its contact with the diaphragm washer. Depending on its position, the pilot rod alternately pressurizes and exhausts the large end of the air distribution valve. Other valve design configurations pressurize and exhaust both ends of the air distribution valve.

The two common types of liquid check valves that are used in a diaphragm pump are the *flap valve* and the *ball valve*. A flap valve pump (see Figures 9 and 10) can handle nearly marble-sized solids. Because the discharge is from the bottom of the diaphragm chambers, the pump is ideally suited for pumping solids in suspension that may tend to settle out, particularly when the pumping rate is reduced or when the pump is shut down. The bottom outlet enables foreign matter to be easily pumped out of the chambers.

The popular air-operated, double-diaphragm pump with ball valves (see Figures 12 and 13) features the inlet at the bottom of the diaphragm chambers, and the outlet is at the top. The top discharge arrangement has the advantage of enabling air or vapors to be easily expelled from the chambers. Trapped air or vapors in pumps having bottom outlets can reduce the volumetric displacement of the pumps as the air or vapor is alternately compressed and expanded, instead of the liquid being displaced. This can be a concern in low-flow applications requiring relatively high pumping pressure and that handle viscous liquids. In higher flow applications, a sufficient turbulence is present and air or vapors mix with the pumped liquid to purge the pumping chambers of the gases.

The performance chart of a typical 2-in (51-mm) air-operated, double-diaphragm pump (see Figure 14) is similar to that of other pump types but contains air consumption rather than horsepower consumption. With a constant supply pressure of compressed air, work and energy relationships in the air chambers, as well as liquid flow losses within the pump, result in a downward sloping head-capacity curve, similar to a centrifugal pump. The following are noteworthy features of air-operated diaphragm pumps:

- With the pump shut off, there is no power consumption. Air consumption is approximately proportional to the flow rate; there is zero air consumption at a zero flow rate and maximum air consumption at a maximum flow rate. This feature enables diaphragm pumps to be used in applications where the flow rate must be varied over a wide range or from no flow to a high flow rate.

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 90 of 153

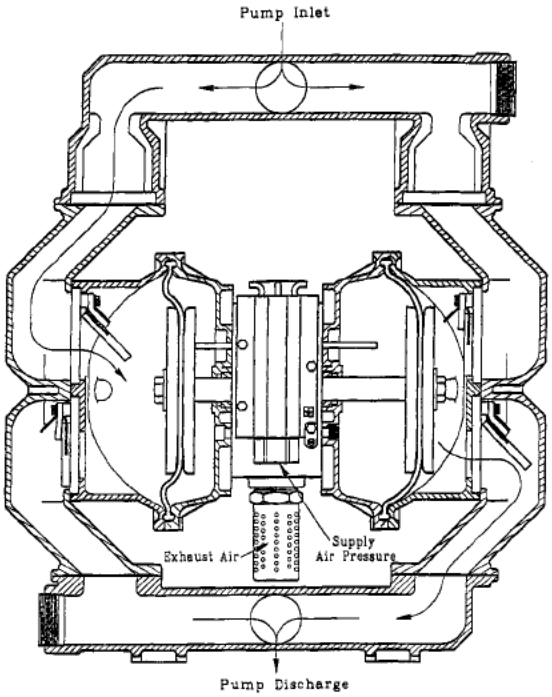


FIGURE 9 Cross-section of an air-operated double-diaphragm pump with flap check valves (Ingersoll-Rand Fluid Products)

- The pump discharge pressure remains the same for a given capacity and air inlet pressure regardless of the specific gravity of the liquid being pumped. The discharge head varies with the specific gravity of the pumped fluid, because pressure and not head is primarily maintained by the air pressure. For centrifugal pumps, the discharge pressure is directly proportional to the specific gravity of the liquid being pumped, while head is fixed at a given point on the performance curve. The supplied air pressure, pump flow rate, and NPSH set the discharge pressure for a given air operated pump, not the head, which is a characteristic of a centrifugal pump operated at a fixed speed.
- The NPSH required for an air-operated diaphragm pump is defined in the same manner as with any reciprocating pump. The NPSH required is determined to occur when the volumetric efficiency drops by a measurable amount (normally 3%) at a fixed speed or flow rate.

MATERIALS OF AIR-OPERATED DIAPHRAGM PUMPS

Check valves for diaphragm pumps are of three types: flap, ball, and poppet (see Figure 15). Flap valves preferably hang in a vertical position with a horizontal flow through the valves. Their principle advantage is the capability to handle large objects in suspension. Flap valves with elastomeric hinges are extensively used. Ball valves are arranged to provide a vertical flow through the valve seats. Poppet valves are usually guided by a valve stem and are spring-loaded. The poppet type of valve is not position-sensitive and can

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 91 of 153

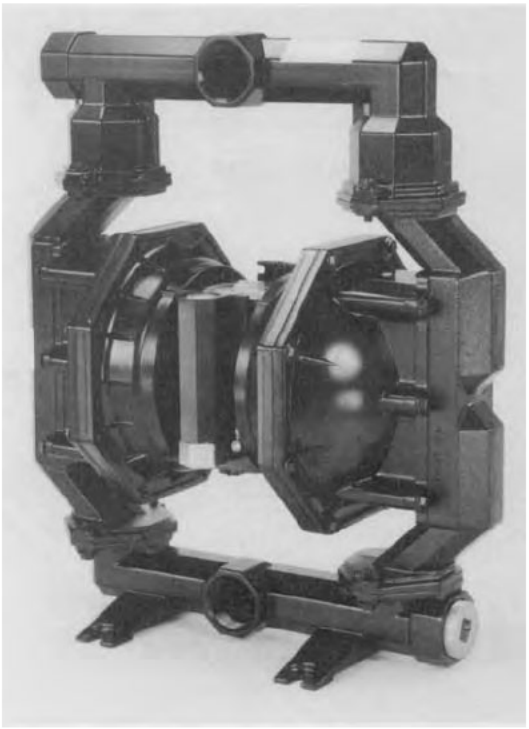


FIGURE 10 ARO air-operated double-diaphragm pump with flap check valves (Ingersoll-Rand Fluid Products)

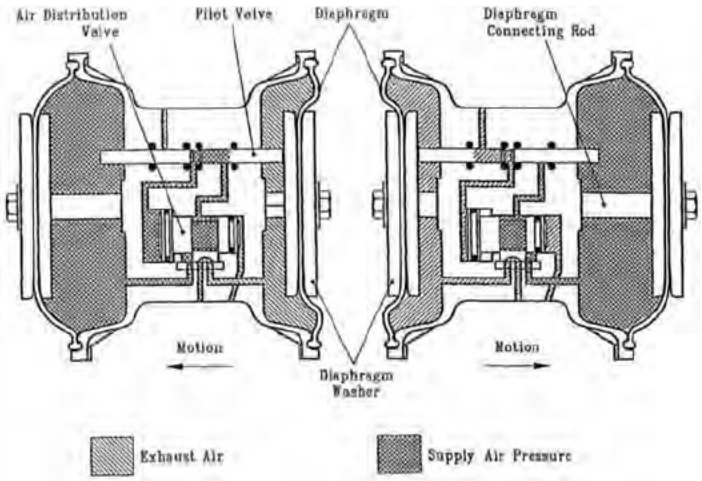


FIGURE 11 Cross-section of an air-operated double-diaphragm pump air motor operation (Ingersoll-Rand Fluid Products)

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 92 of 153

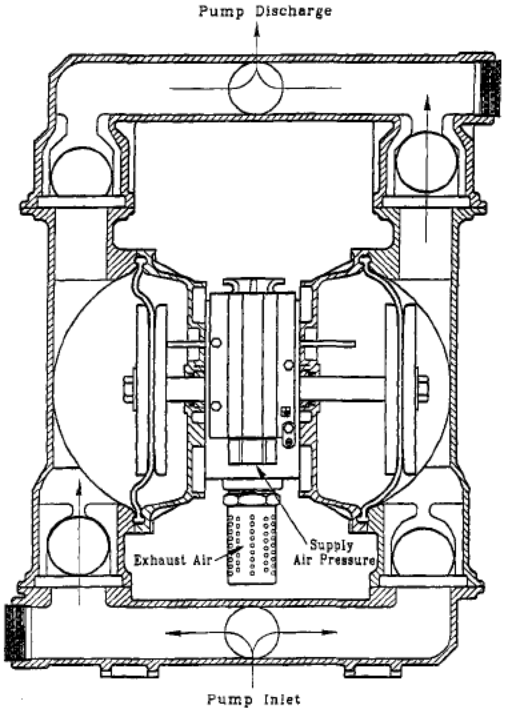


FIGURE 12 Cross-section of an air-operated double-diaphragm pump with ball check valves (Ingersoll-Rand Fluid Products)

operate in any orientation. Since many diaphragm pumps are used to pump abrasive slurries, the valves usually have elastomeric faces or are elastomeric balls. Large ball check valves may have metal cores covered by a thick wall of synthetic rubber. They may also be made of solid rubber, a thermoplastic elastomer, or Teflon®.

Diaphragms are customarily made of fabric-reinforced synthetic rubber, thermoplastic elastomers, or PTFE. Diaphragm materials include most of the synthetic rubbers: neoprene, Buna N, EPR, Viton®, thermoplastic elastomers such as Hytrel® and Santoprene® as well as Teflon®. Many combinations of pump and diaphragm materials can cover a wide range of pumped products. For some solvents and aggressive acids or alkalis, Teflon® diaphragms can be used either directly or as an overlay on the conventional diaphragm.

The diaphragms in air-operated, double-diaphragm pumps are essentially balanced and act simply as membranes separating the compressed air from the product being pumped. The only unbalance occurs during the suction stroke of one diaphragm while it is being pulled by the shaft-connected other diaphragm. When the suction lift is minimal, the unbalance may be negligible.

Pump case materials include cast iron, aluminum, stainless steel, Carpenter 20, Hastelloy C, and plastics reinforced with glass or metal fibers.

APPLICATIONS OF AIR-OPERATED DIAPHRAGM PUMPS

General Diaphragm pumps are ideally suited for handling abrasive slurries because 1) the liquid velocity through the check valves and pumping chambers does not exceed the

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 93 of 153



FIGURE 13 ARO air-operated double-diaphragm pump with ball check values (Ingersoll-Rand Fluid Products)

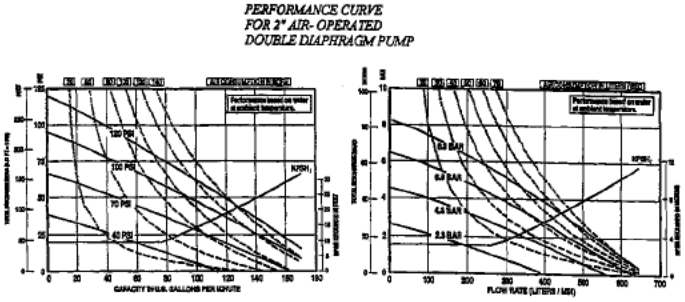


FIGURE 14 Performance curve for a 2-in air-operated double-diaphragm pump

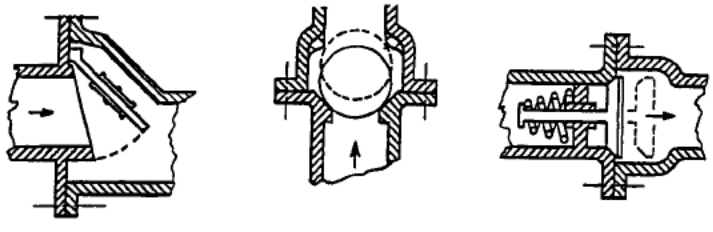


FIGURE 15 Three basic types of check valves

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 94 of 153

pipeline velocity and 2) the scouring and abrasions from the slurry are minimal. Because no close-fit sliding or rubbing parts exist and velocities are low, these pumps can be used for liquids with viscosities up to 50,000 SSU (11,000 cSt). Because the turbulence and mixing are minimal, they are ideally suited for shear-sensitive materials, such as latex.

Pumping Dry Powders Since diaphragm pumps can pump air as well as liquids, they are used successfully to pump dry powders. The air acts as a fluid medium for the powders in suspension, and the pump moves the air containing the suspended powder. Sometimes it is necessary to inject air into the powder to lower the apparent specific gravity and to get the powder into suspension.

Pump Controls An air pressure regulator in the compressed air supply line can control the pumping pressure. An air line valve can control the pumping rate. Thus, the pressure and capacity are easily controlled. Although air-operated diaphragm pumps are displacement pumps, they are not positive ones. The maximum pumping pressure cannot exceed the pressure of the compressed air powering the pumps. Table 1 describes the control of diaphragm pumps with the features inherent with each drive type.

Liquids Handled and Applications A variety of liquids can be handled by AODPs, yet the compatibility of pump case materials, valve materials, and diaphragms is the basic application limitation. Liquids and slurries handled by diaphragm pumps include ceramic slurry, paint, cement grout, chemicals, glue, resins, petroleum products, driller's mud, mill scale, ore concentrates, printer's ink, sewage, filter aids, latex, waste oils, wood preservatives, core washes, asphaltic coatings, bilge waste, radioactive waste, lapping compounds, porcelain frit, mine tailings, volatile solvents, coolants with metal fines, varnish, acids, coatings, soapstone slurries, explosives, lime slurries, yeast, chocolate, and wine. Typical applications of AODPs include tank and container loading and unloading, fluid filtration, spray painting, adhesive application, process mixing and batching, cutting oil, machine coolant, lubrication, sump pumping, dewatering, and waste water treatment.

LIMITATIONS AND ADVANTAGES OF AIR-OPERATED DIAPHRAGM PUMPS

This section outlines the pros and cons of AODPS. Table 1 also details the controls of the different types of pumps. The limitations of AODPs are as follows:

TABLE 1 Control of diaphragm pumps

Feature	Electro-Magnetically Driven	Mechanically Driven	Hydraulically Driven	Air-Operated
Can run dry:	Yes	Yes	Yes	Yes
Self-priming:	Yes	Yes	Yes	Yes
Discharge can be shutoff without damage:	Yes	No	Yes	Yes
Relief valve required:	No	Yes	Integral to pump	No
Flow controlled by:	Motor speed or stroke-length adjustment	Motor speed or stroke-length adjustment	Motor speed or stroke-length adjustment	Motor speed or discharge valve

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 95 of 153

- AODPs are not practical for pumping rates above about 300 gpm (1150 ℓ/m).
- AODPs are not manufactured for operating air pressures above 125 lb/in^2 gauge (8.6 bar). However, some versions are available that increase the pressure ratio by 2:1 or 3:1.
- Ice formation in air motors can occur (basic to the physics of operation), but the effect can be minimized by proper application and design.
- Diaphragms have a finite life. Fluids with abrasives and higher process temperatures can limit a diaphragm's life, but many material choices are available, including Teflon®, and several thermoplastic elastomers.

An AODP's advantages are as follows:

- They are self-priming from a dry start.
- The pumps have an infinitely variable flow rate and pressure-within-pressure and capacity ranges.
- AODPs have no dynamic seals or packing.
- They can run dry indefinitely.
- The discharge can be throttled to zero flow indefinitely.
- No air is used when the AODP is deadheaded. Electrically driven pumps will consume a significant portion of the rated power when no demand flow exists.
- AODPs are suited for use in hazardous environments (no electric power required).
- Power is used in proportion to the pumping rate.
- They can be used in confined areas without heat buildup.
- AODPs can pump abrasive slurries and solids in suspension.
- The pumps can handle viscous liquids up to 50,000 SSU (11,000 cSt).
- A minimal degradation of the viscosity of shear-sensitive materials occurs.
- They can pump dry powders in air suspension.
- No close-fit, sliding, or rotating parts come in contact with liquid.
- No bypass is required as in other displacement pumps.
- When properly maintained, there is zero leakage.
- They are simple to maintain and to repair.
- There is no bedplate and no coupling to align.
- AODPs can be used in handling aggressive chemical solutions.
- They can handle a wider range of materials than any other type of pump.

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 96 of 153

SECTION 3.7
SCREW PUMPS

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Screw pumps are a special type of rotary positive displacement pump in which the flow through the pumping elements is truly axial. The liquid is carried between screw threads on one or more rotors and is displaced axially as the screws rotate and mesh (see Figure 1). In all other rotary pumps, the liquid is forced to travel circumferentially, thus giving the screw pump with its unique axial flow pattern and low internal velocities a number of advantages in many applications where liquid agitation or churning is objectionable.

The applications of screw pumps cover a diversified range of markets including navy, marine, and utilities fuel oil services; marine cargo; industrial oil burners; lubricating oil services; chemical processes; petroleum and crude oil industries; power hydraulics for navy and machine tools; and many others. The screw pump can handle liquids in a range of viscosities, from molasses to gasoline, as well as synthetic liquids in a pressure range from 50 to 5000 lb/in^2 (3.5 to 350 bar) and flows up to 8000 gal/min (1820 m^3/h).

Because of the relatively low inertia of their rotating parts, screw pumps are capable of operating at higher speeds than other rotary or reciprocating pumps of comparable displacement. Some turbine-attached lubricating oil pumps operate at 10,000 rpm and even higher. Screw pumps, like other rotary positive displacement pumps, are self-priming and have a delivery flow characteristic, which is essentially independent of pressure, provided there is sufficient viscosity in the liquid being pumped.

Screw pumps are generally classified into single- or multiple-rotor types. The latter is further divided into timed and untimed categories.

The single-screw or progressive cavity pump (see Figure 2) has a rotor thread that is eccentric to the axis of rotation and meshes with internal threads of the stator (rotor housing or body). Alternatively, the stator is made to wobble along the pump centerline.

Multiple-screw pumps are available in a variety of configurations and designs. All employ one driven rotor in a mesh and one or more sealing rotors. Several manufacturers have two basic configurations available: single-end (in Figure 3) and double-end (in Figure 4) construction, of which the latter is the better known.

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 97 of 153

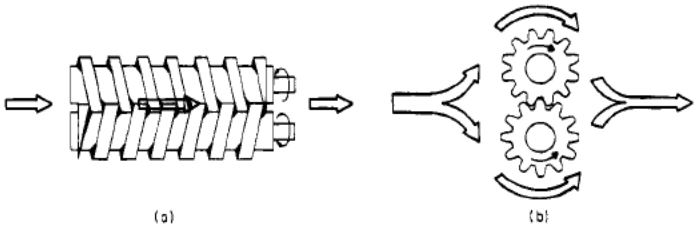


FIGURE 1 Diagrams of screw and gear elements, showing (a) axial and (b) circumferential flow

As with every pump type, certain advantages and disadvantages can be found in a screw pump design. These should be recognized when selecting the best pump for a particular application. The ***advantages*** of a screw pump design are as follows:

- A wide range of flows and pressures
- A wide range of liquids and viscosities
- High speed capability, allowing the freedom of driver selection
- Low internal velocities
- Self-priming, with good suction characteristics
- A high tolerance for entrained air and other gases
- Low velocities for minimum churning or foaming
- Low mechanical vibration, pulsation-free flow, and quiet operation
- A rugged, compact design that is easy to install and maintain
- High tolerance to contamination in comparison with other rotary pumps

The ***disadvantages*** are as follows:

- A relatively high cost because of close tolerances and running clearances
- Performance characteristics sensitive to viscosity changes
- High pressure capability may require long pumping elements
- Four sealing points between rotor and casing for timed two-screw design

THEORY

In screw pumps, it is the intermeshing of the threads on the rotors and the close fit of the surrounding housing that creates one or more sets of moving seals in a series between the pump inlet and outlet. These sets of seals or locks, as they are sometimes referred to, act

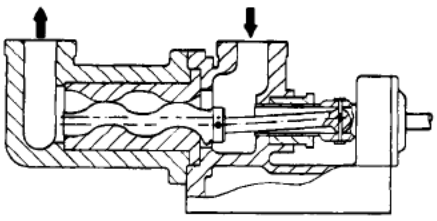


FIGURE 2 The single-screw or progressive cavity pump

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 98 of 153

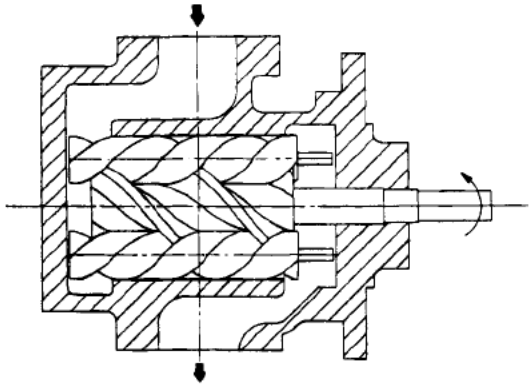


FIGURE 3 Multiple-screw single-end arrangement

as a labyrinth and provide the screw pump with its positive pressure capability. The successive sets of seals form fully enclosed cavities (see Figure 5) that move continuously from inlet to outlet. These cavities trap liquid at the inlet and carry it along to the outlet, providing a smooth flow.

Delivery Because the screw pump is a positive displacement device, it will deliver a definite quantity of liquid with every revolution of the rotors. This delivery can be defined in terms of displacement volume V_D, which is the theoretical volume displaced per revolution of the rotors and is dependent only upon the physical dimensions of the rotors. It is generally measured in cubic inches (cubic millimeters) per revolution. This delivery can also be defined in terms of theoretical capacity or flow rate Q_t measured in U.S. gallons per minute (cubic meters per hour), which is a function of displacement and speed N in rpm:

In USC units: $$Q_t = \frac{V_D N}{231}$$

In SI units: $$Q_t = 6 \times 10^{-8} V_D N$$

If no internal clearances existed, the pump's actual delivered or net flow rate Q would equal the theoretical flow rate. Clearances, however, do exist with the result that whenever a pressure differential occurs, there will always be internal leakage from outlet to inlet. This leakage, commonly called *slip* S, varies depending upon the pump type or model, the amount of clearance, the liquid viscosity at the pumping conditions, and the

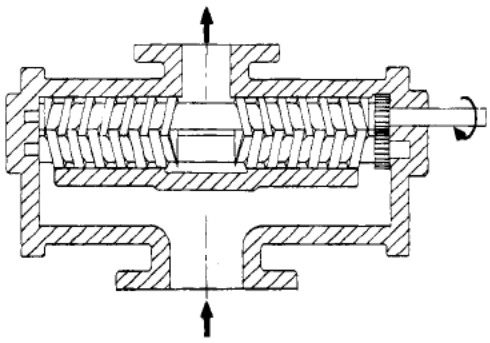


FIGURE 4 Mutliple-screw double-end arrangement

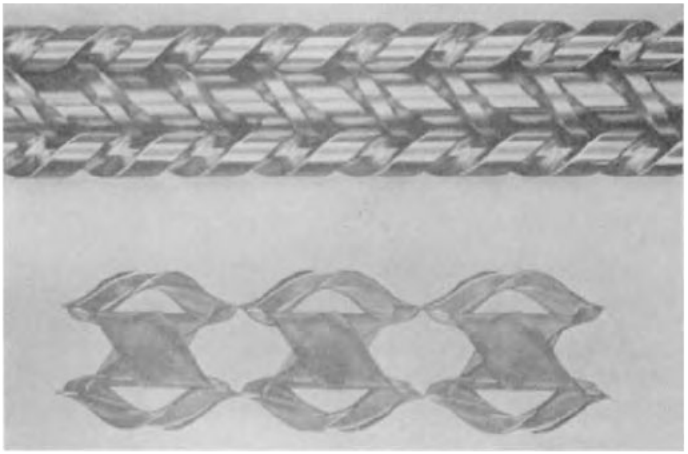


FIGURE 5 Axially moving seals and cavities. Alternate cavities filled with oil shown below (Imo Pump)

differential pressure. For any given set of these conditions, the slip for all practical purposes is unaffected by speed. The delivered flow rate or net flow rate therefore is the theoretical flow rate less the slip: $Q = Q_t - S$. If the differential pressure is almost zero, the slip may be neglected and $Q = Q_t$.

The theoretical flow rate of any pump can readily be calculated if all essential dimensions are known. For any particular thread configuration, assuming geometric similarity, the size of each cavity mentioned earlier is proportional to its length and cross-sectional area. The thread pitch measured in terms of the same nominal diameter, which is used in calculating the cross-sectional area (see Figure 6), defines the length. Therefore, the volume of each cavity is proportional to the cube of this nominal diameter, and the pump's theoretical flow rate is also proportional to the cube of this nominal diameter and the speed of rotation N (rpm):

$$Q_t = kD^3N$$

or, bringing the pitch into evidence,

$$Q_t = k_1 \times \text{pitch} \times D^2N.$$

From Figure 6, where pitch = KD, it follows that $k = k_1K$.

Thus, for a given geometry, it can be seen that a relatively small increase in pump size can provide a large increase in flow rate.

Slip can also be calculated,[1,2] but usually it depends upon empirical values developed by extensive testing. These test data are the basis of the design parameters used by every

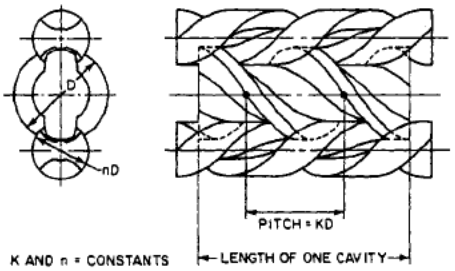


FIGURE 6 Screw-thread proportions, showing lead, diameters, and cavity length

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 100 of 153

pump manufacturer. Slip generally varies approximately as the square of the nominal diameter. The net flow rate therefore is

$$Q_t = kD^3N - S$$

Pressure Capability As mentioned earlier, screw pumps can be applied over a wide range of pressures, up to 5000 lb/in² (345 bar), provided the proper design is selected. Internal leakage must be restricted for high-pressure applications. Close-running clearances and high accuracy of the conjugate rotor threads are requirements. In addition, an increased number of moving seals between the inlet and outlet are employed, as in classic labyrinth-seal theory. The additional moving seals are obtained by a significant increase in the length of the pumping elements for a given size of rotor and pitch. Here the minimum pump length is sacrificed in order to gain pressure capability.

The internal leakage in the pumping elements resulting from the differential pressure between the outlet and inlet causes a pressure gradient across the moving cavities. The gradient is approximately linear (see Figure 7) when measured at any instant. Actually, the pressure in each moving cavity builds up gradually and uniformly from inlet to outlet pressure as the cavities move toward the outlet. In effect, the pressure capability of a screw pump is limited by the allowable pressure rise across any one set of moving seals. This pressure rise is sometimes referred to as *pressure per closure* or *pressure per lock* and generally is of the order of 125 to 150 lb/in² (8.6 10 10.3 bar) with normal running clearances, but it can go as high as 500 lb/in² (35 bar) when minimum clearances are employed.

The above analysis is used in the design of pumps for handling liquid products where there are sufficient lubricating properties in the liquid to allow the screws to contact the bores of the casing. However, when handling products containing significant amounts of dissolved and/or entrained gas, it is preferable to have no rubbing contact between the rotating elements and the casing bores. For these pumps, the rotors are relatively short and stiff to ensure there is no rotor to bore contact with the lower-viscosity, gas-laden multiphase fluids.

Design Concepts The pressure gradient in the pump elements of all the types of screw pumps produces various hydraulic reaction forces. The mechanical and hydraulic techniques employed for absorbing these reaction forces are among the fundamental differences in the types of screw pumps produced by various manufacturers. Another fundamental difference lies in the method of engaging, or meshing, the rotors and maintaining the running clearances between them. Two basic design approaches are used:

- The *timed rotors* approach relies upon an external means for phasing the mesh of the threads and for supporting the forces acting on the rotors. In this concept, theoretically, the threads do not come into contact with each other or with the housing bores in which they rotate (refer to Figure 4).
- The *untimed rotors* approach relies on the precision and accuracy of the screw forms for the proper mesh and transmission of rotation. They utilize the housing bores as journal bearings supporting the pumping reactions along the entire length of the rotors (refer to Figure 3).

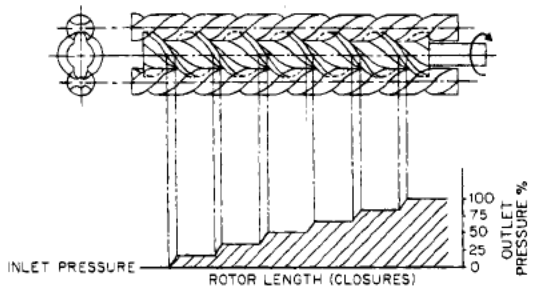


FIGURE 7 Pressure gradient along a screw set

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 101 of 153

Timed screw pumps require separate timing gears between the rotors and separate support bearings at each end to absorb the reaction forces and maintain the proper clearances. Untimed screw pumps do not require gears or external bearings and thus are considerably simpler in design.

CONSTRUCTION

Basic Types As indicated in the introduction, three major types of screw pumps exist:

- Single-rotor
- Multiple-rotor timed
- Multiple-rotor untimed

The second and third types are available in two basic arrangements, single-end and double-end. The double-end construction (see Figure 8) is probably the best-known version, as it has been by far the most widely used for many years because of its relative simplicity and compactness of design.

Double-End Screw Pumps The double-end arrangement is basically two opposed, single-end pumps or pump elements of the same size with a common driving rotor that has an opposed, double-helix design with one casing. As can be seen from Figure 8, the fluid enters a common inlet with a split flow going to the outboard ends of the two pumping elements and is discharged from the middle or center of the pump elements. The two pump elements are, in effect, pumps connected in parallel. The design can also be provided with a reversed flow for low-pressure applications. In either of these arrangements, all axial loads on the rotors are balanced, as the pressure gradients in each end are equal and opposite.

The double-end screw pump construction is usually limited to low- and medium-pressure applications, with 400 lb/in^2 (28 bar) being a good practical limit to be used for planning purposes. However, with special design features, applications up to 1400 lb/in^2 (97 bar) can be handled. Double-end pumps are generally employed where large flows are required or where very viscous liquids are handled.

Single-End Screw Pumps All three types of screw pumps are offered in the single-end construction. As pressure requirements in many applications have been raised, the single-end design has come into much wider use because it provides the only practical means

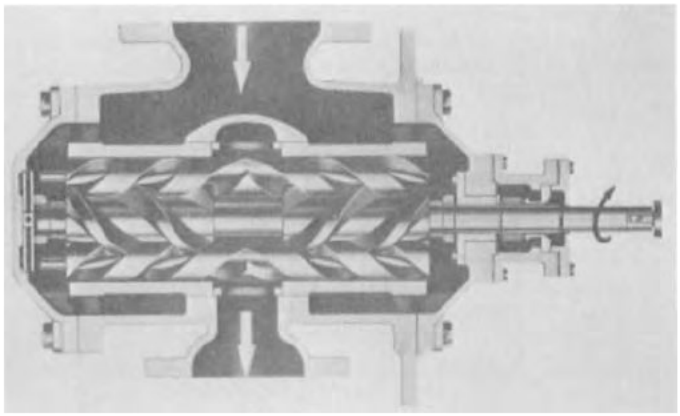


FIGURE 8 Double-end pump. Flow path provides axial balance (Imo Pump)

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 102 of 153

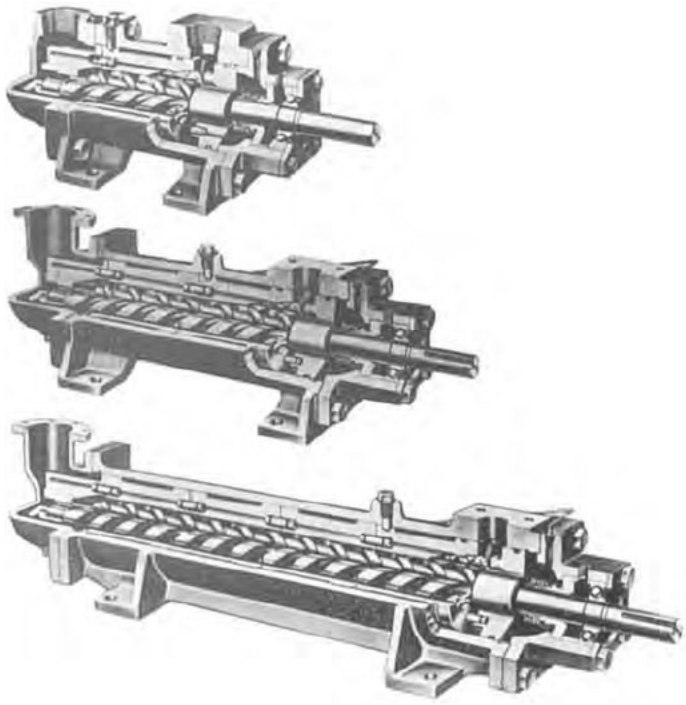


FIGURE 9 Increasing pump pressure capability by modular design (Imo Pump)

for obtaining the greatest number of moving seals necessary for high-pressure capabilities. The main penalties of the single-end pump are the requirement and complexity of balancing the axial loads.

The single-end construction is most often employed for handling low-viscosity fluids at medium-to-high pressures or hydraulic fluids at very high pressures. The single-end design for high pressures is developed by literally stacking a number of medium-pressure, single-end pumping elements in a series within one pump casing (see Figure 9). The single-end construction also offers the best design arrangement for quantity manufacturing.

Special mention must be made of the single-end, single-rotor design (see Figure 10). The pump elements of this design consist of only a stator and one rotor. The stator has a double-helix internal thread and is constructed of hard chrome stainless or tool steel material. One version of this design uses internally developed pressure in the pump to compress the elastomeric stator on the rotor, thus maintaining minimum running clearances.

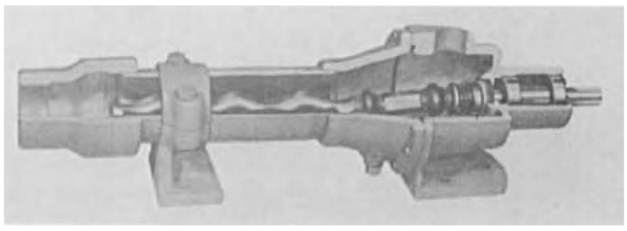


FIGURE 10 Single-rotor pump (Robbins-Meyers)

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 103 of 153

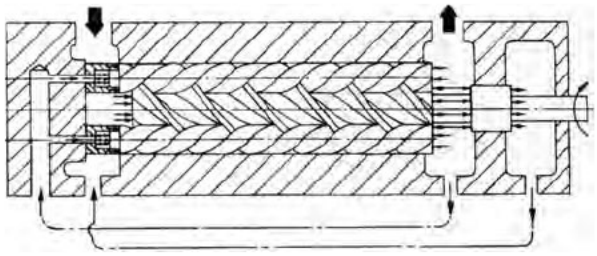


FIGURE 11 Axial balancing of power and idler rotors

In most single-end designs, special axial balancing arrangements must be used for each of the rotors, and, in this respect, the design is more complicated than the double-end construction. For smaller pumps, strictly mechanical thrust bearings can be used for differential pressures up to 150 lb/in^2 (10.3 bar), while hydraulic balance arrangements are used for higher pressures. For large pumps, hydraulic balance becomes essential at pressures above 50 lb/in^2 (3.5 bar).

Hydraulic balance is provided through the balance piston (see Figure 11) mounted on the rotors between the outlet and seal or bearing chambers, which are at inlet pressure. This piston is exposed to discharge pressure in the outlet chamber and is equal in area to the exposed area of the driven rotor threads; thus, the hydraulic forces on the rotor are canceled out.

Timed Design Timed screw pumps having timing gears and rotor support bearings are furnished in two general arrangements: internal and external. The internal version has both the gears and the bearings located in the pumping chamber and the design is relatively simple and compact (see Figure 12). This version is generally restricted to the handling of clean lubricating fluids, which serve as the only lubrication for the timing gears and bearings.

The external timing arrangement is the most popular and is extensively used. It has both the timing gears and the rotor support bearings located outside the pumping chamber (see Figure 13). This type can handle a complete range of fluids, both lubricating and nonlubricating, and, with proper materials, has good abrasion resistance. The timing gears and bearings are oil-bath-lubricated from an external source. This arrangement requires the use of four stuffing boxes or mechanical seals, as opposed to the internal type, which employs only one shaft seal.

The main advantage of the timed screw pump is that the timing gears transmit power to the rotors with no contact between the screw threads, thus promoting long pump life. The gears and rotors are timed at the factory to maintain the proper clearance between the screws. With certain designs and loading characteristics, the bearings at each end of

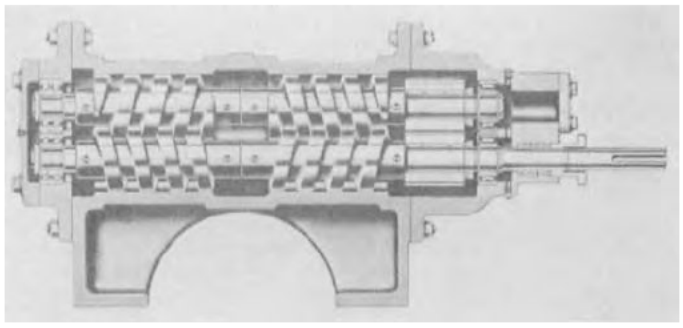


FIGURE 12 Double-end internal gear design (Flowserve Corporation)

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 104 of 153

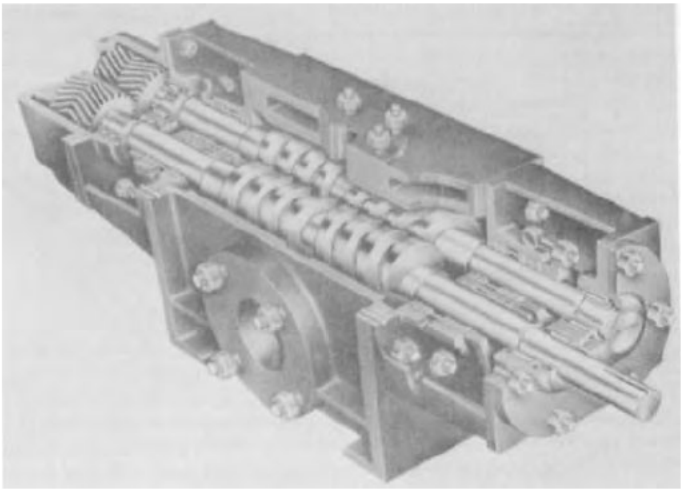


FIGURE 13 Double-end external gear design (Warren)

the rotating elements can support the rotors so that they do not come in contact with the housing; hence, no liner is required. One set of these bearings also positions and supports the timing gears. Under more heavily loaded conditions, the body bores act as sleeve bearings and provide additional support for the rotors.

The timing gears can be either spur or helical, herringbone, hardened-steel gears with tooth profiles designed for efficient, quiet, positive drive of the rotors. Antifriction radial bearings are usually of the heavy-duty roller type, while the thrust bearings, which position the rotors axially, are either double-row, ball-thrust or spherical-roller types.

The housing can be supplied in a variety of materials, including cast iron, ductile iron, cast steel, stainless steel, and bronze. In addition, the rotor bores of the housing can be lined with industrial hard chrome for abrasion resistance.

Since the rotors are not generally in metallic contact with the housing or with one another, they can also be supplied in a variety of materials, including cast iron, heat-treated alloy steel, stainless steel, Monel, and nitralloy. The outside of the rotors can also be furnished with a variety of hard coating materials such as nickel-based alloys, tungsten carbide, chrome oxide, or ceramic.

Untimed Design The untimed type of screw pump has rotors that have generated mating-thread forms that enable any necessary driving force to be transmitted smoothly and continuously between the rotors without the use of timing gears. The rotors can be compared directly with precision-made helical gears with a high helix angle. This design usually employs three rotor screws with a center or driven rotor that meshes with two close-fitting sealing or idler rotors symmetrically positioned about the central axis (see Figure 14). A close-fitting housing provides the only transverse bearing support for both driven and idler rotors.

The use of the rotor housing as the only means for supporting idler rotors is a unique feature of the untimed screw pump. No outboard support bearings are required on these rotors. The idler rotors in their related housing bores are, in effect, partial journal bearings that generate a hydrodynamic film. The key parameters of rotor size, clearance, surface finish, speed, fluid viscosity, and bearing pressure are related as in a journal bearing. Because, in this design, the screws transmit the driving torque to the idlers and because the casing bores support the rotors, the pumped fluid must have some lubricating properties. This prevents this pump design from being used in applications with any solids, abrasives or gas in the pumpage.

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 105 of 153

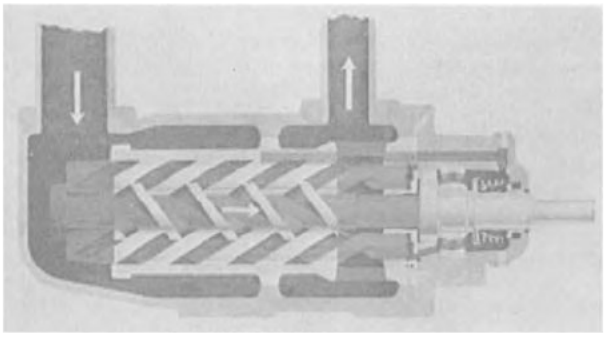


FIGURE 14 Single-end design (Roper)

Since the idler rotors are supported by the bores along their entire length, no significant bending loads are applied to them. The central driven rotor is also not subjected to any significant bending loads because of the symmetrical positioning of the idler rotors and the use of two threads on all the rotors (see Figure 15). This is quite different from the two-rotor design common to the timed type where the hydraulic forces generated in the pump create bending loads on the intermeshing pairs of screws.

In contrast to timed pumps, the untimed design, with its absence of timing gears and bearings, appears very simple, but its success depends entirely upon the accuracy and finish of the rotor threads and rotor housing bores. Special techniques and machine tools have been developed to manufacture these parts. The combination of design simplicity and manufacturing techniques has enabled this design to be used for very long rotor lengths, with a multiplicity of sealing closures, for applications up to 5,000 lb/in^2 (350 bar).

In special applications handling highly aerated oils, a four-rotor design is sometimes employed in the untimed type. Three idler rotors are equally spaced radially at 120° around the driven rotor. This design is not truly a positive displacement pump, and it falls outside the scope of this section.

The rotors in untimed pumps are generally made of gray or ductile iron or carbon steel. The thread surfaces are often hardened for high pressure and abrasive resistance. Flame hardening, induction hardening, and nitriding are currently used. Through-hardened tool steel or stainless steel can be used in some critical applications.

Rotor housings, or liners, are made of gray pearlitic iron, bronze, or aluminum alloy. In many instances, the bores as well as the rotors can be treated by the application of dry lubricant or toughening coatings. The pump casings are made of gray iron or ductile iron or cast steel where shock or other safety requirements demand it.

In many untimed designs, an antifriction bearing is employed on the shaft end of the driven rotor (refer to Figure 9) to provide precise shaft positioning for mechanical seal and

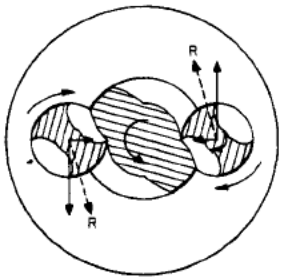


FIGURE 15 Force diagram on rotor set

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 106 of 153

coupling alignment. This bearing can be either an external grease-sealed bearing or an internal type with the pumped fluid providing the lubrication. The bearing also supports overhung loads with belt or gear drives.

Seals As with any rotary pump, the sealing arrangement for the shafts is important and is often critical. Every type of rotary seal has been used in screw pumps at one time or another. Except for canned or sealless arrangements, all types of pumps require at least one rotary seal on the drive shaft. The timed screw pumps with external timing and bearings require additional seals at each rotor end to separate the pumped fluid from the lubricating oil necessary for the gears and bearings.

For drive shafts, rotary mechanical seals as well as stuffing boxes or packings are used, depending on the manufacturer and/or customer preference. Double back-to-back arrangements or tandem mounted seals with a flushing liquid are sometimes used for very viscous or corrosive substances.

Many different sealing face materials such as carbon, bronze, cast iron, Ni-resist, carbides, or ceramics are used, along with different secondary elastomer seals of various materials chosen to suit the application. Balanced and nonbalanced seals can also be used to handle significant pressures. In some pump designs, the seals are subjected to only the suction pressure, while in other designs the seals must seal against the full discharge pressure. In spite of these advantages, the stuffing box is still preferred by some users. The stuffing box requires regular maintenance and tightening, which many users find objectionable, but a mechanical seal failure usually results in a major shutdown.

PERFORMANCE

Performance considerations of screw pumps are closely related to applications, and so any discussion must cover both. In the application of screw pumps, certain basic factors must be considered to ensure a successful installation. These are fundamentally the same regardless of the liquids to be handled or the pumping conditions.

In most cases, the pump selection for a specific application is not difficult if all the operating parameters are known. It is often quite difficult, however, to obtain this information, particularly inlet conditions and fluid viscosity. It is a common feeling that, inasmuch as the screw pump is a positive displacement device, these items are unimportant.

In any screw pump application, regardless of design, suction lift, viscosity, and speed are inter related. Generally, low operating speeds are required for high-viscosity fluids while higher speeds are possible with lower-viscosity fluids.

Inlet Conditions The key to obtaining good performance from a screw pump, as with all other positive displacement pumps, lies in a complete understanding and control of inlet conditions and the closely related parameters of speed and viscosity. To ensure quiet, efficient operation, it is necessary to completely fill the moving cavities between the rotor threads with liquid as they open to the inlet, and this becomes more difficult as the viscosity, speed, or suction lift increases. It can be said that, if the liquid can be properly introduced into the rotor elements, the pump will perform satisfactorily.

It must be remembered that a pump does not pull or lift liquid into itself. Some external force must be present to push the liquid into the rotor threads initially. Normally, atmospheric pressure is the only force present, but in some applications a positive inlet pressure is available.

As with centrifugal pumps, the concept of NPSH required by the pump is used to ensure that there is sufficient pressure available at the inlet to fill the rotor cavities and suppress cavitation at the inlet. With rotary pumps this is sometimes referred to as *Net Inlet Pressure Required*.

Since the purpose of the NPSH available is to fill the moving inlet cavities with liquid, it is apparent that the faster the pump is operated *the greater is the amount of* NPSH that is required.

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 107 of 153

Fluids and Vapor Pressure In many cases, screw pumps handle a mixture of liquids and gases, and therefore the general term *fluid* is more descriptive. Most of these fluids, especially petroleum products because of their complex nature, contain certain amounts of entrained and dissolved air or some other gas, which is released as a vapor when the fluid is subject to reduced pressures. If the pressure drop required to overcome entrance losses is sufficient to reduce the static pressure significantly, vapors are released in the rotor cavities and cavitation results.

In all screw pump applications, the absolute static pressure must never be allowed to drop below the vapor pressure of the fluid. This will prevent vaporization or cavitation. Cavitation, as mentioned previously, results when fluid vaporizes in the pump inlet because of incomplete filling of the pump elements and a reduction of pressure. Under these conditions, vapor bubbles, or voids, pass through the pump and collapse as each moving cavity moves into a domain of higher pressure. The result is noisy vibrations, the severity depending on the extent of vaporization or the incomplete filling and the magnitude of the discharge pressure. Since the cavitation bubbles occupy space normally taken by liquid, the output flow rate is reduced by this amount. It is therefore important to be fully aware of the characteristics of entrained and dissolved air as well as of the vapor pressure of the fluid to be handled. This is particularly true when a suction lift exists.

Net Positive Suction Head The suction conditions of screw pumps are normally defined by the *Net Positive Suction Head* (NPSH) available at the pump inlet. In some cases with systems open to the atmosphere, the inlet conditions can also be defined as suction lift. Suction lift occurs when the total available pressure at the pump inlet is below the atmospheric pressure and is normally the result of a change in elevation and pipe friction. Since atmospheric pressure at sea level corresponds to 14.7 lb/in^2 (1 bar) absolute, or 30 in Hg (762 mm Hg), in a system with the suction open to the atmosphere, this is the maximum amount of pressure available for moving the fluid, and suction lift cannot exceed these figures. In practice, the available pressure is lower because some of it is used up in overcoming friction in the inlet lines, valves, and fittings. It is considered the best practice to keep the suction lift as low as possible to ensure that an adequate NPSH is available to fill the suction cavities of the pumping screw. In all cases, the system NPSHA (available) should be greater than the pump NPSHR (required).

The NPSHA for a screw pump is calculated in the same way as for a centrifugal pump and the same basic application guidelines apply. The pump manufacturers test their pumps and measure the NPSHR (required) under various operating conditions. The end user must understand the suction pressure required and ensure that there is sufficient NPSHA (available) to suppress the cavitation. Generally the NPSHA should be at least 3 ft (1 m) greater than the NPSHR to provide a reliable application.

Many screw pumps operate with suction lifts of approximately 5 to 15 in Hg (127 to 381 mm Hg). Lifts corresponding to 24 to 25 in Hg (610 to 635 mm Hg) are not uncommon, and installations can operate satisfactorily when the absolute suction pressure is much lower. In the latter cases, however, the pumps usually take the fluid from tanks under a vacuum, and no entrained or dissolved air or gases are present. Great care must be taken when selecting pumps for these applications since the inlet losses can easily exceed the net suction head available for moving the fluid into the pumping elements.

The defining of suction requirements by the user and the stating of pump suction capabilities by the manufacturer have always been complex problems. In many cases, the NPSHA is difficult to predict due to changes in the fluid characteristics and the operating conditions. In addition, the NPSH required by the pump is a function of many variables, such as pump design, fluid characteristics, and operating conditions. If the operating conditions can be accurately defined, the pump manufacturers can predict the NPSHR and in many cases can provide pump modifications that can minimize the NPSHR.

To enable the pump manufacturer to offer the most economical selection and also assure a quiet installation, accurate suction conditions should be clearly stated. Specifying a lower NPSHA than actually exists may result in selection of a pump that operates at a lower speed than necessary. This means not only a larger and more expensive pump, but also a costlier driver. If the NPSHA is lower that stated, the outcome could be a noisy pump installation.

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 108 of 153

Many known instances of successful installations exist where screw pumps were properly selected for low NPSHA conditions. Unfortunately, many other installations with equally low NPSHAs exist, which are not so satisfactory. This is because proper consideration was not given at the time the pump was specified and selected to the actual suction conditions at the pump inlet. Frequently, suction conditions are given as "flooded" simply because the source feeding the pump is above the inlet. In many cases, no consideration is given to outlet losses from the tank or to pipe friction in the inlet lines, and these can be exceptionally high in the case of viscous liquids.

When it is desired to pump extremely viscous products, care should be taken to use the largest feasible size of suction piping to eliminate all unnecessary fittings and valves, and to place the pump as close as possible to the source of the supply. In addition, it may be necessary to supply the liquid to the pump under some pressure, which can be supplied by elevation, air pressure, or mechanical means. These actions will provide the maximum NPSH possible to the pump inlet.

Entrained and Dissolved Air Although screw pumps can handle significant quantities of gas as discussed in the later multiphase section, entrained or dissolved gas in a pump system designed only for liquids can create a problem and must be considered in the application. This is particularly true of installations where recirculation occurs and the fluid is exposed to air through either mechanical agitation, leaks, or improperly located drain lines.

Most liquids will dissolve air or other gases retaining them in the solution, the amount being dependent upon the liquid itself and the pressure to which it is subjected. It is known, for instance, that lubricating oils at atmospheric temperatures and pressures will dissolve up to 10% air by volume and that gasoline will dissolve up to 20%. When pressures below the atmosphere exist at the pump inlet, dissolved air will come out of the solution. Both this and the entrained air will expand in proportion to the existing partial pressure of the air (= absolute pressure minus the vapor pressure of the liquid). This expanded air will accordingly take up a proportionate part of the available volume of the moving cavities, with a consequent reduction in delivered flow rate.

One of the apparent effects of handling liquids containing entrained or dissolved gas is noisy pump operation. When such a condition occurs, it is usually dismissed as cavitation. This should not be the case, particularly with screw pumps. With properly designed systems and pumps, quiet, vibration-free operation can be produced and should be expected. Noisy operation is inefficient; steps should be taken to make corrections until the objectionable conditions are overcome. Correct system inlet designs and optimized pump designs with a proper speed selection can go a long way toward overcoming the problem.

In some applications, the amount of gas can be significant and can make up the majority of the fluid volume. See the later section on handling special multiphase applications.

Viscosity It is not often that a screw pump is called upon to handle liquids at a constant viscosity. Normally, because of temperature variations, a wide range of viscosities will be encountered. For example, a pump may be required to handle a viscosity range from 150 to 20,000 SSU, the higher viscosity usually resulting from cold-starting conditions. This is a perfectly satisfactory range for a screw pump, but a better and a more economical selection may be possible if additional information can be obtained. This information includes such things as the amount of time the pump is required to operate at the higher viscosity, whether the motor can be overloaded temporarily, whether a multi-speed motor can be used, and if the discharge pressure will be reduced during the period of high viscosity.

Quite often, only the type of liquid is specified, not its viscosity, and assumptions must be made for the operating range. For instance, Bunker C or No. 6 fuel oil is known to have a wide range of viscosity values and usually must be handled over a considerable temperature range. The normal procedure in a case of this type is to assume an operating viscosity range of 20 to 700 SSF. The maximum viscosity, however, might easily exceed the higher value if extra-heavy oil is used or if exceptionally low temperatures are encountered. If either should occur, the result may be improper filling of the pumping elements, noisy operation, vibration, and overloading of the motor.

Although it is the maximum viscosity and the expected NPSHA that are used to determine the size of the pump and to set the speed, it is the minimum viscosity that affects the

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 109 of 153

capacity. Screw pumps must always be selected to give the specified capacity when handling the expected minimum viscosity since this is the point at which the maximum slip, and hence minimum flow rate, occurs (see Figure 16). It should also be noted that the minimum viscosity often determines the selection of the pump model because most manufacturers have special lower-pressure ratings for handling liquids having a viscosity of less than 100 SSU.

Non-Newtonian Liquids The viscosity of most liquids is unaffected by any agitation or shear to which they may be subjected as long as the temperature remains constant. These liquids are accordingly known as *true* or *Newtonian* liquids, for which the viscosity is constant. Another class of liquids, however, such as cellulose compounds, glues, greases, paints, starches, slurries, and candy compounds, displays changes in viscosity as agitation is varied at constant temperature. The viscosity of these substances depends upon the shear rate at which it is measured, and these fluids are termed *non-Newtonian.*

If a substance is known to be non-Newtonian, the expected viscosity under the actual pumping conditions should be determined, because it can vary quite widely from the viscosity under static conditions. Since a non-Newtonian substance can have an unlimited number of viscosity values (as the shear rate is varied), the term *apparent viscosity* is used to describe its viscous properties. The apparent viscosity is expressed in absolute units and is a measure of the resistance to the flow at a given shear rate. It has meaning only if the shear rate used in the measurement is also given.

The grease-manufacturing industry is very familiar with the non-Newtonian properties of its products, as evidenced by the numerous curves that have been published of the apparent viscosity plotted against the rate of shear. The occasion is rare, however, when one can obtain accurate viscosity information when it is necessary to select a pump for handling these products.

It is practically impossible in most instances to give the viscosity of grease in the terms most familiar to the pump manufacturer, such as Saybolt Seconds Universal or Saybolt Seconds Furol, but only a rough approximation would be of great help. For applications of this type, data taken from similar installations are most helpful. Such information should consist of the type, size, flow rate, and speed of the installed pumps; the suction pressure; the temperature at the pump inlet flange; the total working suction head; and, above all, the pressure drop in a specified length of piping. From the latter, a satisfactory approximation of the effective viscosity under the operating conditions can be obtained.

If accurate shear rate-viscosity data are available, they can be used to more accurately predict pump performance. The shear rates in various areas of the pump can be calculated to determine the viscosity changes of the liquid as it passes through the pump. In this way, the effect on suction loss, slip, and friction loss can be analyzed to help predict the NPSH, flow rate, and power.

Speed It was previously stated that viscosity and speed are closely tied together and that it is impossible to consider one without the other. Although rotative speed is the

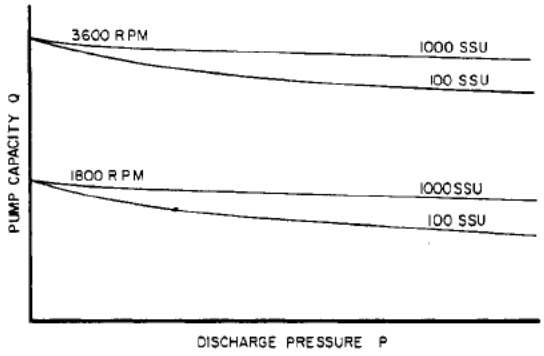


FIGURE 16 Flow rate (capacity)-vs-pressure rise

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 110 of 153

ultimate outcome, the basic speed that the manufacturer must consider is the internal axial velocity of the liquid going through the rotors. This is a function of pump type, design, and size.

Rotative speed should be reduced when handling liquids of high viscosity. The reasons for this are not only the difficulty of filling the pumping elements, but also the mechanical losses that result from the shearing action of the rotors on the substance handled. The reduction of these losses is frequently of more importance than relatively high speeds, even though the latter might be possible because of positive inlet conditions.

Capacity The delivered flow rate (capacity) of any screw pump, as stated earlier, is the theoretical capacity less the internal leakage, or the slip, when handling vapor-free liquids. For a particular speed, $Q = Q_t - S$, where the standard unit of Q and S is the U.S. gallon per minute (cubic meter per minute).

The delivered capacity of any specific rotary pump is reduced by

- Decreasing speed
- Decreased viscosity
- Increased differential pressure

Because of the internal clearances between rotors and their housing, lower viscosities and higher pressures increase the slip, which results in a reduced flow rate for a given speed. The impact of these characteristics can vary widely for the various types of pumps. The slip, however, is not measurably affected by changes in speed and thus becomes a smaller percentage of the total flow at higher speeds. This is a significant factor in the handling of low-viscosity fluids at higher pressures, particularly in the case of untimed screw pumps that favor high speeds for the best results and best volumetric efficiency. This will not generally be the case with pumps having support-bearing speed limits.

Pump volumetric efficiency E_v is calculated as

$$E_v = \frac{Q}{Q_t} = \frac{Q_t - S}{Q_t}$$

with Q_t varying directly with speed. As stated previously, the theoretical capacity of a screw pump varies directly as the cube of the nominal diameter. Slip, however, varies approximately with the square of the nominal diameter. Therefore, for a constant speed and geometry, doubling the rotor size will result in an eightfold increase in theoretical flow rate and only a fourfold increase in slip. It follows therefore that the volumetric efficiency improves rapidly with increases in the rotor size, assuming the internal clearances can be maintained.

On the other hand, viscosity changes affect the slip inversely in a complex manner that depends on pump size, speed, and the level of the viscosity itself. Testing or advanced analysis is required to establish this relationship.[1] However, for a given viscosity in the range of 100 to 10,000 SSU an acceptable approximation is that slip varies approximately with the differential pressure. Thus, in this case, a change from 100 to 200 lb/in^2 (7 to 14 bar) would double the slip.

Figure 17 shows typical results for flow rate and volumetric efficiencies as functions of pump size.

Pressure Screw pumps do not in themselves create pressure; they simply transfer a quantity of fluid from the inlet to the outlet side. The pressure developed on the outlet side is solely the result of resistance to the flow in the discharge line. The slip characteristic of a particular pump type and model is one of the key factors that determine the acceptable operating range, and it is generally well defined by the pump manufacturer.

Power The brake horsepower (bhp), or, in SI units, the brake kilowatts, required to drive a screw pump is the sum of the theoretical liquid horsepower (kilowatts) and the internal power losses. The theoretical liquid power *twhp* (*tkW*) is the actual work done in moving the fluid from its inlet pressure condition to the outlet at the discharge pressure.

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 111 of 153

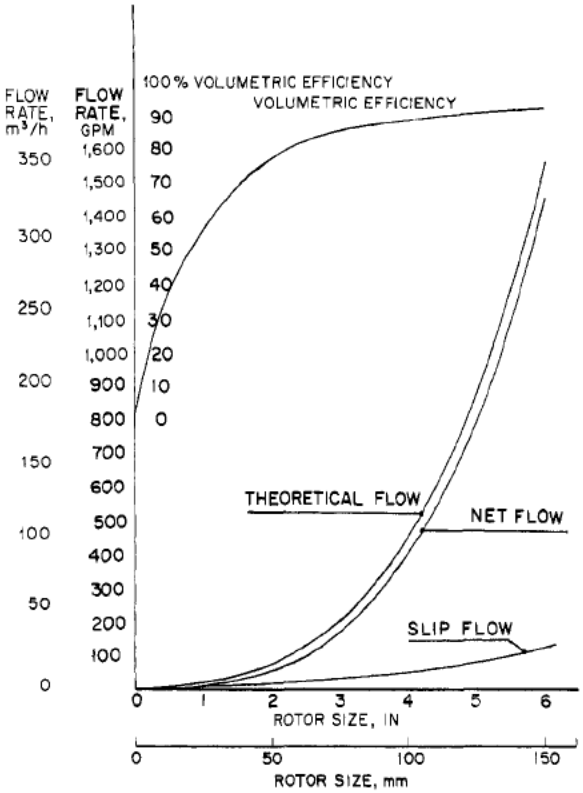


FIGURE 17 Typical variations of flow rate and volumetric efficiency as functions of pump size

Note that this work is done on all the fluid of the theoretical capacity, not just the delivered capacity, as the slip does not exist until a pressure differential D*P* occurs. Screw pump power ratings are expressed in terms of horsepower (550 ft-lbf/sec) in USCS units and in terms of kilowatts in SI units. The theoretical liquid horsepower (kilowatts) can be calculated as follows:

$$twhp = \frac{Q_t \Delta P}{1714} \left(tkW = \frac{Q_t \Delta P}{36} \right)$$

It should be noted that the theoretical liquid horsepower (kilowatts) is independent of the viscosity and is a function only of the physical dimensions of the pumping elements, the rotative speed, and the differential pressure.

The internal power losses are of two types: mechanical and viscous. The mechanical losses include all the power necessary to overcome the frictional drag of all the moving parts in the pump, including rotors, bearings, gears, and mechanical seals. The viscous losses include all the power lost from the fluid drag effects against all the parts in the pump as well as from the shearing action of the fluid itself. It is probable that the mechanical loss is dominant when operating at low viscosities and high speeds, and the viscous loss is the larger of these two losses at high-viscosity and slow-speed conditions.

In general, the losses for a given type and size of pump vary with the viscosity and the rotative speed and may or may not be affected by pressure, depending upon the type and

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 112 of 153

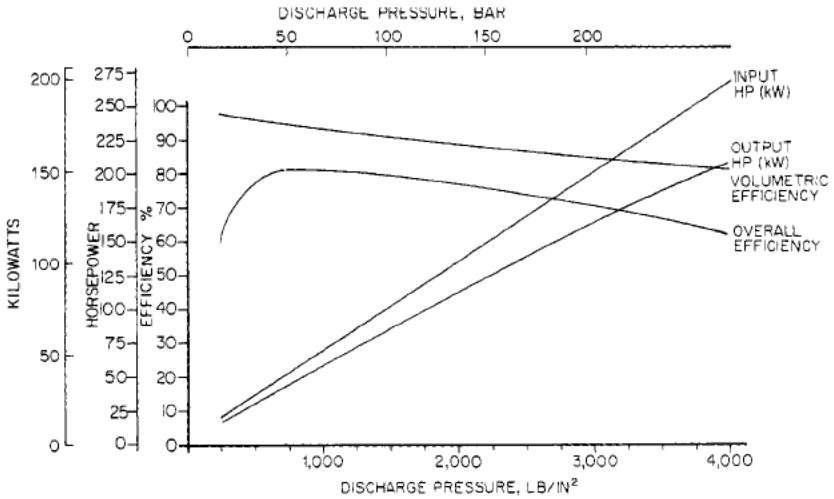


FIGURE 18 Typical overall efficiency curves

model of pump under consideration. These losses, however, must always be based upon the maximum viscosity to be handled since they will be highest at this point.

The actual pump power output, *whp* (*wkW*), or the delivered liquid horsepower (kilowatts), is the power imparted to the liquid by the pump at the outlet. It is computed similarly to theoretical liquid horsepower (kilowatts) using Q in place of Q_t. Hence, the value will always be less. The pump efficiency E_p is the ratio of the pump power output to the brake horsepower (see Figure 18).

In many cases the efficiency of the pump is not that significant, since there may be only a few pumps capable of handling high viscosity and difficult fluids. The main requirement is to provide a reliable equipment package that is capable of meeting the specified requirements.

SPECIAL MULTIPHASE APPLICATIONS

Screw pumps have been used with gas-entrained applications for many years, but recent process changes in oil field technologies have created requirements for pumping multiphase fluids, containing more than just nominal amounts of gases. In many oil well applications, the liquid oil flow eventually degenerates into all sorts of difficult multiphase mixtures of oil, gas, water, and sand. In the past, it was common for the gas to be separated and flared off at the well head with only the liquid product to be retained for further processing. If the gas is to be processed as well, separators, compressors, and dual pipelines are required to handle the gas phase. Therefore, a pump, which can handle these difficult liquids with high gas content, can save significant equipment costs as well as operating costs. Under various conditions, the well output can vary from 100% liquid to 100% gas and all possible combinations. The applications also require the pumping equipment to be able to switch rapidly between the extremes or to handle slugs of liquid or gas, while maintaining the full discharge pressure. Since these applications involve solid contaminants, internal bearing and non timed pump designs are not suitable. The single rotor design with elastomer stator has proven suitable for lower pressure applications while the timed two-screw type of pump has been proven capable of pumping these multiphase products at higher pressues.[2]

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 113 of 153

Oil well applications can include traditional on-shore sites as well as off-shore platform installations. Subsea installations are also being used to reduce the high costs of equipment and operation of traditional oil platforms. In these applications, the pumping equipment is mounted on the sea bed with piping running to floating production, storage and off-loading facilities or to onshore gathering facilities. Although the actual pumping conditions are similar to surface installations, the installation and operating environments are far more challenging.

When pumping multiphase products with high *gas void fractions* (GVF), the pump must be designed with a small pitch to provide a sufficient number of locks. The key to pumping multiphase products is to ensure that some liquid is always available to seal the screw clearances and reduce the slip. Even a small amount of recirculated liquid is sufficient to provide this seal and enable the screw pump to operate with GVF-values approaching 100%. Depending on a number of factors, the volume of liquid required to seal and cool the screws can be two to five percent of the total inlet volume flow rate. In order to ensure that sufficient liquid is available at conditions of high GVF, a separate liquid flush can be provided or a trapping type of pump casing can be used. This type of casing has the suction and discharge openings located on the upper side of the casing to trap the liquid inside the screw chambers to seal the screw clearances. In some cases, some of the liquid portion is separated from the multiphase mixture being pumped. This separated liquid can then be recirculated back to the screws and mechanical seals to provide sealing and cooling liquid at times when the product is almost all gas. Figure 19 shows a special screw pump designed for multiphase applications with a casing designed to trap the liquid inside the casing bores to seal the screw clearances.[3]

When pumping liquids, the slip through the internal clearances increases with the differential pressure and decreases with increasing viscosity. However, in multiphase applications, as the GVF increases, the slip decreases until the inlet volume flow rate is equal to the pump displacement. This results in an almost constant inlet volume flow rate, regardless of the differential pressure. This performance can be explained by examining the pressure drop of the multiphase product across the finite clearances. This theoretical analysis confirms that a small amount of liquid in the clearances will effectively seal these clearances and reduce the slip to near zero. Figure 20 shows the typical performance characteristics for

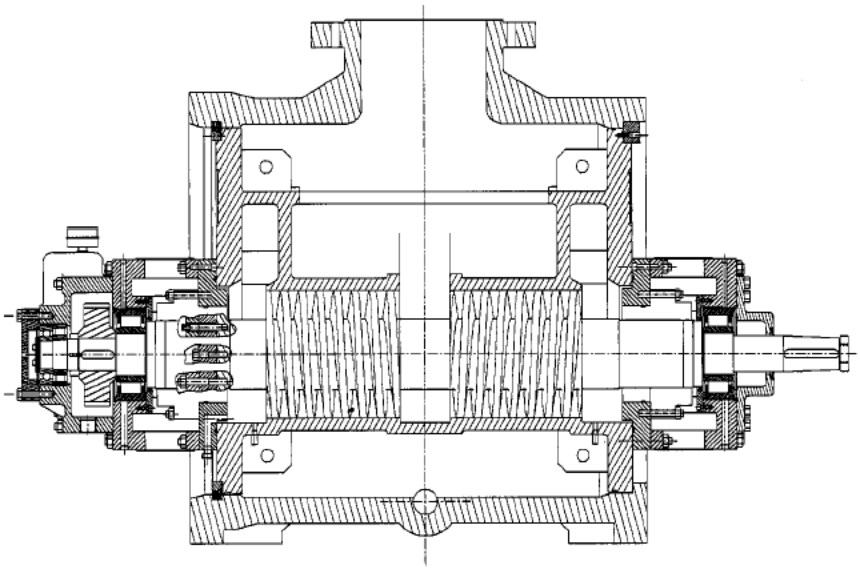


FIGURE 19 Special screw pump for multiphase applications (Flowserve Corporation)

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 114 of 153

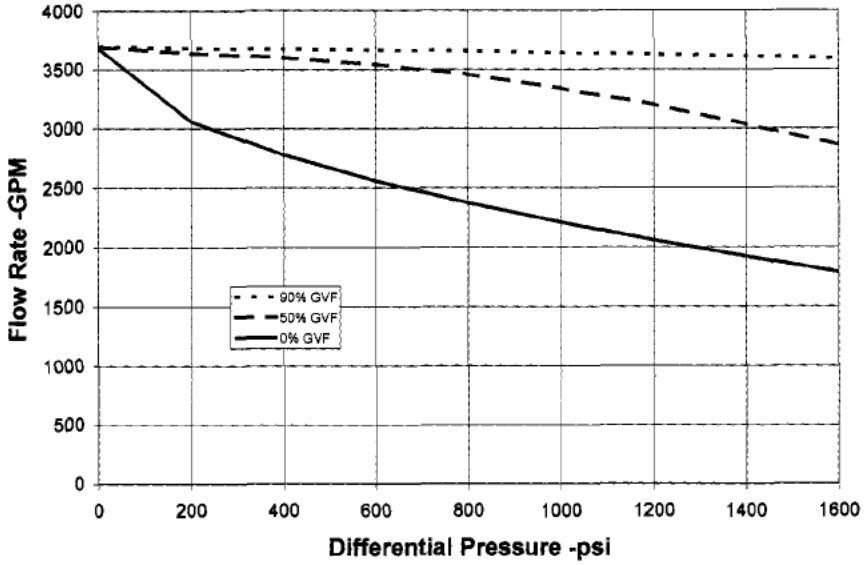


FIGURE 20 Typical performance of a screw pump on multiphase fluid (m^3/h = gpm/4.403; MPa = psi/145.04)

a screw pump in a multiphase application when pumping mixtures of air and water at various GVF values.[1,4]

It should be emphasized that screw pumps must be sized for the inlet volume conditions. Since the gas portion of the multiphase product is compressible, the inlet pressure and temperature conditions must be known in order to calculate the gas volume. The pump will ingest a fixed volume of product and the amount of liquid being pumped will depend on how much of this volume is being displaced by gas at the inlet to the screws.

When pumping higher GVF multiphase products some significant design changes must be made to the screw pumps to improve reliability and ensure proper pumping action. The normal practice used with viscous liquid pumping of using the casing bores to help support the rotor loads is not used with multiphase liquids due to the lower liquid viscosity, the high percentage of gas and the entrained sand and other contaminants present in oil field applications. As a result, pumps for multiphase applications are generally designed with stiffer rotors so that the rotors do not contact the casing bores. This design change is accomplished by using larger diameter shafts with shorter bearing spans. The integral shaft and rotor design used by some manufacturers significantly increases the rotor stiffness and minimizes the rotor deflection under load.

Another design feature used for multiphase applications involves limiting the pitch of the screws to ensure there are sufficient screw locks available to generate the pressure rise and produce a more gradual pressures rise across the screw length. Since the volume of the gas is compressed as the multiphase fluid moves through the screw passages, any large volume changes will result in pressure pulsations and resulting damaging vibration. A general guideline for higher pressure multiphase applications requires a minimum of three screw locks in order to minimize the pressure pulsations. In multiphase applications the maximum pressure rise occurs in the last lock when it opens to the full discharge pressure of the pump. Special porting of the screw profiles can also be used to help minimize the pressure pulsation and resulting vibration.

Again the mechanical seals are the most critical items in the pump required to ensure pump reliability. The pump should be designed to provide the best environment for the mechanical seals to operate properly. This includes limiting rotor deflection at the seal area and providing suitable flush or barrier fluid. Since these pumps can be designed to operate with the seals in either the suction or the discharge it is normally preferable to have

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 115 of 153

the seals work against the lower suction pressure as opposed to the higher discharge pressure. When sealing against variable high suction or discharge pressures, the barrier fluid pressure must be controlled to ensure that it is always greater than the product pressure.

In many multiphase applications, the drive motors are equipped with variable speed controls to allow the pump to operate over a wide speed range to match specific start-up and operating well conditions. In these cases the speed is controlled to maintain a constant pressure at the pump suction. If the well flow rate increases, the suction pressure increases, causing the speed of the pump to also increased to maintain a constant suction pressure and thereby match the flow rate from the well. Multiphase pumps are often used as booster pumps to draw down the well head pressure and thereby increase the well flow rate and boost production. These pumps are often sized to handle high flow rates and can then be installed on a cluster of wells to boost the pressure into the pipeline.

Subsea A special area of application for multiphase pumps is subsea. With special modifications, the rotary screw multiphase pumps can be coupled to submersible motors and mounted on the sea bed, instead of on surface platforms. The idea of pumping multiphase products directly from a subsea well head to onshore or floating production and storage facilities by means of submersible multiphase pumps has significant potential savings in separating equipment and platforms.[5] These special subsea pumps often require much higher pressure capabilities than surface mounted pumps, to handle both the high ambient subsea pressures and also the high discharge pressures required. Again the mechanical seals are a critical design area with special considerations required to ensure a reliable design. The sealing environment can be significantly improved by maintaining a constant differential pressure across the seal faces by means of a pressure compensator. This pressure compensator senses the product pressure on the seals and then provides a slight pressure boost to the barrier fluid to maintain this constant differential pressure. The pump and system must be designed in conjunction with the submersible motor, pressure compensator and lube oil cooler. Obviously, the motor and valve control functions as well as the lube oil top-up capabilities must be integrated into the complete pump module to provide a reliable unit to operate in the subsea environment. Subsea multiphase pumps are being investigated for deeper water applications and higher boost pressures. Initial subsea pumping installations involved water depths up to 1500 m, with boost pressures of 1250 psi (86 bar), applications up to 3000 m water depth with pressure boosts to 3000 psi (207 bar) are now being pursued. Figure 21 shows a special two-screw multiphase

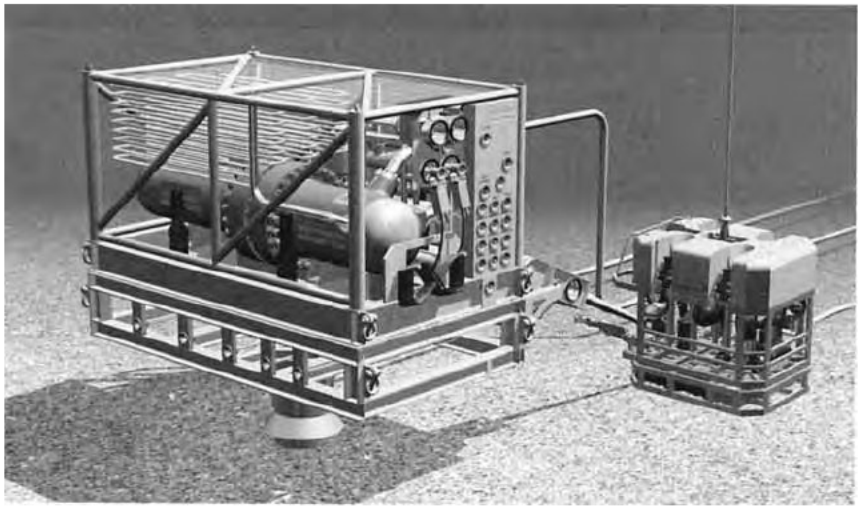


FIGURE 21 Multiphase screw pump module for subsea deployment (Flowserve Corporation)

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 116 of 153

pump module, configured for subsea applications with high-pressure, shell-type casing design, integrated submersible motor, pressure-compensator and oil cooler.

Wet Gas Compression Another application for multiphase screw pumps is for "wet gas compression." This application involves pumping multiphase products that are essentially gas but can also have occasional high liquid content. These applications generally involve fluids with gas content in the 98% GVF range. There are two areas of concern with this type of application. As previously indicated, there must always be some liquid available in the pump to seal the clearances between the screws and the casing bores. Also with this high gas content, special considerations must be made for the thermodynamic heating characteristics of the compressed gas. Since the gas provides much lower heat absorption than the normal liquid, the product temperature can increase significantly as the pressure is increased. The pump must also be able to handle the full range of GVF values from slugs of liquid to 100% gas. For these applications special screw designs, liquid injection and recirculating liquid separators can be used to ensure there is sufficient liquid available to seal and cool the rotors.

INSTALLATION AND OPERATION

Rotary pump performance and life can be improved by following the recommendations on installation and operation outlined in this section.

Pipe Size Resistance to the flow usually consists of differences in elevation, fixed resistances or restrictions such as orifices, and pipe friction. Nothing can be done about the first, since this is the basic reason for using a pump. Something can be done, however, about restrictions and pipe friction. Significant amounts of money can be wasted because of piping that is too small for the job. Certainly, not all pipe friction can be eliminated as long as fluids must be handled in this manner, but every effort should be made to use the largest pipe that is economically feasible. Numerous tables are available for calculating frictional losses in any combination of piping. Among the most recent are the tables in the *Hydraulic Institute Engineering Data Book*[6] along with other similar hydraulic engineering references.

Before any new installation is made, the cost of larger-size piping, which will result in lower pump pressures, should be carefully balanced against the cost of a less expensive pump, a smaller motor, and a savings in power over the expected life of the system. The larger piping may cost a little more in the beginning, but the ultimate savings in power will often substantially offset the original cost. These facts are particularly true for extremely viscous fluids.

Foundation and Alignment The pump should be mounted on a smooth, solid foundation readily accessible for inspection and repair. It is essential that the driver shaft and the pump shaft are in proper alignment. The manufacturer's recommendation of concentricity and parallelism should always be followed and checked occasionally. For high temperature applications the alignments should be checked with the equipment at operating temperature.

The suction pipe should be as short and straight as possible with all joints airtight. It should not contain places where air or other entrapped gases may collect. If it is not possible to have the fluid flow to the pump under gravity, a foot or check valve should be installed at the end of the suction line or as far from the pump as possible. All piping should be independently supported to avoid strains on the pump casing.

Start-Up A priming connection should be provided on the suction side, and a relief valve should be set from five to ten percent above the maximum working pressure on the discharge side. Under normal operating conditions with completely tight inlet lines and wetted pumping elements, a screw pump is self-priming. Starting the unit may involve simply

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 117 of 153

opening the pump suction and discharge valves and starting the motor. It is always advisable to prime the unit before the initial startup to wet the screws. In new installations, the system may be full of air, which must be removed. If this air is not removed, the performance of the unit will be erratic, and, in certain cases, air in the system can prevent the unit from pumping. Priming the pump should preferably consist of filling not only the pump with fluid but as much of the suction line as possible.

The discharge side of the pump should be vented at startup. Venting is especially essential when the suction line is long or when the pump is initially discharging against the system pressure.

If the pump does not show a discharge of liquid after being started, the unit should be shut down immediately. The pump should then be primed and tried again. If it still does not pick up fluid promptly, there may be a leak in the suction pipe, or the trouble may be traceable to an excessive suction lift from an obstruction, throttled valve, or another cause. Attaching a gage to the suction pipe at the pump will help locate the trouble.

Once the screw pump is in service, it should continue to operate satisfactorily with practically no attention other than an occasional inspection of the mechanical seal or packing for excessive leakage and a periodic check to be certain that the alignment is maintained within reasonable limits.

Care should be taken to ensure that the suction pipe is free of any solid contaminants, left over from the construction phase. Flushing of the suction pipe before connecting to the pump is recommended. In some applications, suction strainers are installed to prevent solid particles from reaching the pump. If required, the strainer can be removed after the system start-up phase is completed.

Noisy Operation Should the pump develop noise after satisfactory operation, this is usually indicative of an excessive suction lift resulting from cold liquid, air in the liquid, misalignment of the coupling, or, in the case of an old pump, excessive wear.

Shutdown Whenever the unit is shut down, if the operation of the system permits, both the suction and discharge valves should be closed. This is particularly important if the shutdown is for an extended period because leakage in the foot valve, if the main supply is below the pump elevation, could drain the oil from the unit and necessitate repriming as in the initial starting of the system.

Abrasives One other point has not yet been discussed, and this is the handling of liquids containing abrasives. Since screw pumps depend upon close clearances for proper pumping action, the handling of abrasive fluids usually causes rapid wear. Much progress has been made in the use of harder and more abrasive-resistant materials for the pumping elements so that a good job can be done in some instances. It cannot be said, however, that performance is always satisfactory when handling liquids laden excessively with abrasive materials. On the whole, screw pumps should not be used for handling fluids of this character unless a shortened pump life and an increased frequency of replacements are acceptable. In these cases, reducing the operating speed can maximize the operating life of the pump.

CONCLUDING REMARKS

As indicated, screw pumps are manufactured in a number of different configurations and designs to suit a variety of different applications. Generally, screw pumps should be considered if there is a requirement for high pressure, high viscosity, or low hydraulic pulsation levels. Screw pumps can be used with lubricating and non-lubricating products and on many special and difficult applications. The keys to the success of screw pump applications include accurate knowledge of the fluid characteristics, proper pump selection and sizing, correct system design, suitable installation conditions, and the proper operating and maintenance procedures.

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 118 of 153

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GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 119 of 153

SECTION 3.8
VANE, GEAR, AND LOBE PUMPS

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DEFINITIONS AND NOMENCLATURE

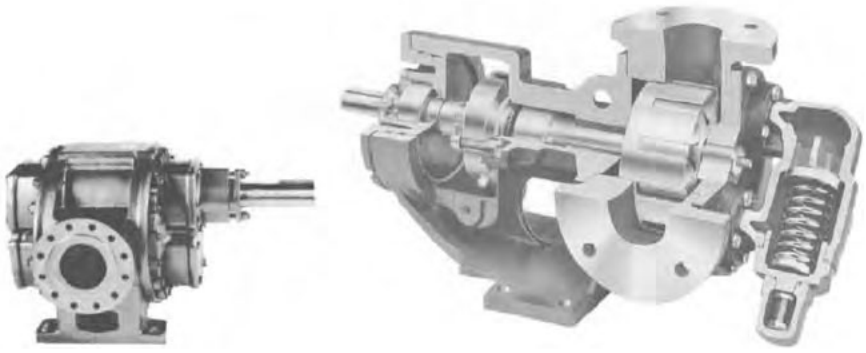
Vane, gear, and lobe pumps are positive displacement rotary pumps. The Hydraulic Institute defines them as mechanisms consisting of a casing with closely fitted vanes, gears, cams, or lobes that provide a means for conveying a fluid. Their principle motion is rotating, rather than reciprocating, and they displace a finite volume of fluid with each shaft revolution. When describing them, the general term *fluid* is used, rather than the more restrictive *liquid*. Fluid, in this case, is understood to include not only true liquids, but mixtures of liquids, gases, vapors, slurries, and solids in suspension as well.

How They Work Pumping in a vane, gear, or lobe pump begins with the rotating and stationary parts of the pump defining a given volume or cavity of fluid enclosure. This enclosure is initially open to the pump inlet but sealed from the pump outlet and expands as the pump rotates. As rotation continues, the volume progresses through the pump to a point where it is no longer open to the pump inlet but not yet open to the pump outlet. It is in this intermediate stage where the pumping volume or cavity is completely formed.

Depending on the particular pump, there can be more than one cavity in existence at any one time. As this happens, fluid also fills the clearances between the pumping elements and pump body, forming a seal and lubricating the pumping elements as they in turn pump the fluid. Rotation continues and the cavities progress, moving fluid along the way. Soon a point is reached where the seal between the captured fluid volume and outlet part of the pump is breached. At this point the vanes, gears, or lobes force the volume of captured fluid out of the pump. While this is happening, other cavities are simultaneously opening at the inlet port to receive more fluid in a continual progression from suction to discharge ports.

For optimum pumping action, the *open-to-inlet* (OTI) volume should expand slowly and continuously with pump rotation. The *closed-to-inlet-and-outlet* (CTIO) pumping cavity volume should remain constant once it is formed, and the *open-to-outlet* (OTO) volume

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 120 of 153



(a) (b)

FIGURE 1a, 1b Typical rotary gear pumps (1b: Viking Pump)

should expand slowly and continuously with pump rotation. At no time should any fluid in the pumping chambers be simultaneously open to both the inlet and outlet if the pump is truly a positive displacement pump. When these conditions are met, the result is a very smooth continuum of flow with minimal pulsations or pressure spikes.

With rotary pumps (see Figure 1), a driver turns one shaft and rotor assembly, which in turn physically meshes with another to form the cavities that move the fluid. This is known as an *untimed arrangement*. For some applications, however, there would be problems with the gears meshing this way. For instance, stainless steel gears will gall and seize if rubbed against each other.

To circumvent this, the timed pump was developed. It uses timing gears physically located outside the pumping chamber to transmit torque between the pump shafts and synchronize the pumping elements relative to each other. By preventing them from contacting each other, they eliminate many of the problems of material compatibility, and dry running. Lobe pumps (Figure 2) are built this way.

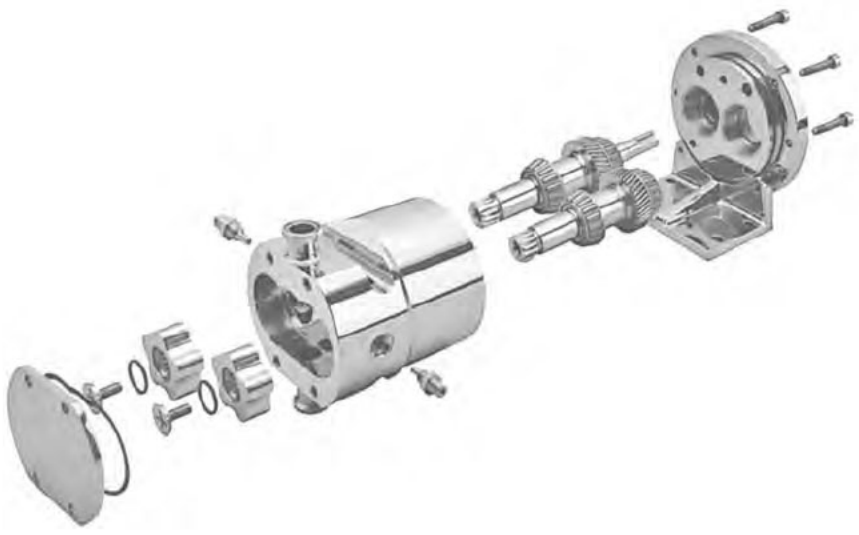


FIGURE 2 Lobe pump (Viking Pump)

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 121 of 153

In some special cases, pressure-balancing designs, relief valve arrangements, or other such considerations have led to some designs where the pump must only operate in one direction. However, the principle of operation of most rotary pumps permits them to operate equally well in either direction.

Main Components The pumping chamber of a rotary pump is the area containing the pumped fluid while the pump is operating. Fluid enters the pumping chamber through the inlet port and leaves through the outlet port. The body is that part of the pump that surrounds the boundaries of the pumping chamber and is also referred to as a *casing, housing,* or *stator*. *End plates* can either be part of the body or separate parts, and they serve to close off the ends of the body to form the pumping chamber. End plates can also be referred to as *pump covers*.

The rotating assembly generally refers to all those parts that rotate when the pump is operating. *Rotors* are usually given more descriptive names, depending on the specific pump type. All rotary pumps employ a drive shaft to accept driving torque from the pump driver. The majority of rotary pumps are mechanically coupled to their driver with various types of couplings, but sealless magnetically driven pumps have become more common in recent years.

The cavity through which the drive shaft protrudes is called the *seal chamber*, and leakage through it is controlled by a *mechanical seal* or *packing*. In a mechanical seal, two faces with opposing axial loads are maintained in close contact with each other. When compressible packing and a stuffing box are used in place of a mechanical seal and seal chamber, the packing is compressed in the stuffing box by a gland that keeps it in intimate contact with the stationary and rotating elements. A *lantern ring* or *seal cage* is often placed between two of the packing rings to enable cooling and lubrication from an external source.

A number of other auxiliary devices and arrangements can be found in vane, gear, and lobe pumps, but two are especially characteristic of these pump types. Given the positive displacing nature of these pumps, and the potentially high pressures that can result at the outlet of the pump if there is an obstruction or blockage, *safety relief valves* must be used with all positive displacement pumps. They limit the pressure by opening an auxiliary passage at a predetermined set pressure and relieving flow back to the inlet side of the pump or, better, to the fluid's original source. They can be installed externally or serve as an integral part of the pump. One exception is flexible member pumps, which, by the nature of their design, usually do not require one, due to the resiliency and expandability of their elastomeric components.

Nonetheless, most rotary pumps are available with integral relief valves. Although these devices are a necessary investment for safety, they are not a substitute for an external system relief valve. They are a secondary safety device at best and are not intended for continuous duty, flow control, or system pressure modulation.

To reduce fluid viscosity in the pump body to facilitate a successful startup and maintain event-free operation, *heating jackets* are used. They can be either integral to the pump (either welded on or part of the body casting) or a separate bolt-on type. They are common with asphalt, gelatins, paraffin, molasses, greases, and similar fluids where, without heating, the power and torque required to drive the pump could easily overload the driver. If viscous enough, a cold startup could actually destroy the mechanical seal, shear a shaft coupling, break a drive shaft, or cause damage to the equipment some other way. Heating jackets are not intended to be the main source of heating in the system. If the system requires the fluid to be heated, some other means, such as heat tracing the system piping, must be used.

Materials of Construction When selecting materials for rotary pumps, consideration must be given to the following material properties:

- The modulus of elasticity (for deformation purposes)
- The coefficient of thermal expansion (for varying temperatures)
- The coefficient of friction (for resistance to galling when in sliding contact)

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 122 of 153

For rotary pumps with flexible members, further consideration must be given to the materials' bulk modulus for recovery from deformation.

The close running clearances of rotary pumps require that their materials resist deformation and deflection by the various forces present when the pump is operating. If they do not, then such deformation or deflection could open the clearances and lower the operating efficiency dramatically or close the clearances and cause high mechanical loading and seizing between the moving and stationary parts.

The materials must also have compatible coefficients of thermal expansion. With the potential for deflection of the rotating parts always present, the materials selected must also have good bearing characteristics or, in particular, a good so-called "PV-value" (defined below) to resist galling and wear. This is especially important when pumping low-lubricity fluids.

Furthermore, materials used for corrosion resistance in noncontacting surfaces of centrifugal pumps may be unusable in rotary pumps where the continuous sliding contact between parts can wear away their passivating or protective layers. In general, rotary material restrictions become more severe when handling low-viscosity fluids at higher pressures and/or low lubricity fluids with abrasives. Most rotary pumps do not handle solids well, except for small trace amounts. Progressing cavity pumps, however, (see Section 3.7) are an exception and can handle very high solids content. In addition, even where there is no load-bearing contact between the rotating and stationary parts under normal conditions, the high transient forces generated at startup, shutdown, or any other unusual operating conditions (such as cavitation) must be considered when selecting pump materials.

The performance of flexible member pumps heavily depends on the material of the flexible member. Its bulk modulus must be high enough to keep distortion under pressure within functional limits and it must be resilient enough to spring back to its original shape after flexing or compressing. For instance, if, once deflected, the vanes in a flexible vane pump stayed that way, the pump could no longer operate. That is, these materials must be chosen not only to satisfy the desired hydraulic conditions, but also for resistance to deterioration from fatigue, chemicals, and the temperatures to which they may be exposed.

Vane Pumps Two basic types of vane pumps exist. The most common is the rigid sliding metal vane type, and the other is the flexible or elastomeric vane used for low viscosity (e.g., gasoline) or chemically aggressive fluids. Both are based around external sliding vanes rotating about a nonconcentric cam.

All rigid vane pumps have moveable sealing elements in the form of nonflexing blades, rollers, buckets, scoops, and so on. These elements move radially inward and outward by cam surfaces to maintain a fluid seal between the OTI and OTO sectors during pump operation. When the cam surface is internal to the pump body and the vanes are mounted in or on the rotor, the pump is called an *internal vane pump*. The OTI volume is defined by the body walls, the rotor walls, the fluid seal contact between the vanes, and the body. The body wall surface, the rotor surfaces, and the vane-to-rotor and vane-to-body fluid seal points define the CTIO volume. The body surface, the rotor surface, the vane-to-body fluid seal points, and the vane-to-rotor fluid seal points define the OTO volume.

In internal vane pumps, the volume behind the vanes must always be either a composite constant volume or else be vented, because of the piston-like pumping action of the vanes on the fluids trapped there. However, no such venting is required when the vanes are in the form of rocking slippers.

When the cam surface is external to the radial surface of the rotor and the vane, or the vanes are mounted in the body or stator, the pump is called an *external vane pump* and is illustrated in Figure 3. The OTI, CTIO, and OTO volumes are defined the same as for internal vane pumps when multiple external vanes are used. In this case, the rotor surface, the body surface, and the fluid seal points between them define the CTIO volume.

In addition to rigid, sliding metal vane pumps, flexible or elastomeric vane type pumps also exist. This kind of pump, illustrated in Figure 4, has a pumping action similar to that of an internal vane pump with the OTI, CTIO, and OTO volumes defined by the rotor surfaces, the body surfaces, the fluid seal contacts between the rotor flexible vanes, and the body surfaces.

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 123 of 153

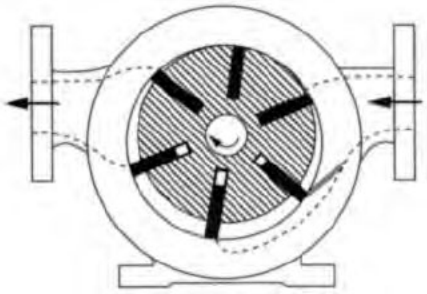


FIGURE 3 A typical external vane pump

The flexible liner pump in Figure 5 is similar in pumping action to the external vane pump, and all three chamber volumes of it are defined by the inner surface of the body, the outer surface of the liner, and the liquid seal contact between the liner and body bore. Most flexible liner pumps, unlike other rotary pumps, have at least one position of the rotor in which no fluid seal exists between the OTI and OTO volumes. The pump depends only on fluid velocity and inertia to limit backflow during this phase of rotation.

Vane pumps offer flows at up to 1,000 gpm (3,785 l/min) and pressures at up to 125 lb/in^2 (8.6 bar). They are commonly used for low-pressure transfers of gasoline, kerosene, and similar light hydrocarbons.

Gear Pumps The gear pump is the most common of all rotary pumps due to the wide variety of applications it can be used in.

Gear pumps have two gears that mesh to provide its pumping action, with one gear driving the other. The physical contact between the gears forms a part of the moving fluid seal between the inlet and outlet ports. The outer radial tips of the gears and the sides of the gears form part of the moving fluid seal between the inlet and outlet ports. The gear contact locus moves along the tooth surfaces and jumps discontinuously from tooth to tooth as the gears mesh and unmesh during rotation. These two characteristics distinguish gear pumps from lobe pumps where the rotors (lobes) are incapable of driving each other, (timing gears doing that instead,) and the fluid seal contact locus between lobes moves continuously across all the radial surfaces of the lobes.

Gear pumps are classified as *external* or *internal* (see Figure 6). External gear pumps have their gear teeth cut on their external or outside diameter and mesh about their outside diameters. Bearings (which are often product-lubricated bushings) support the shafts at both ends with the gears located between the bearings. This resists shaft deflection and contact between the gears and casing wall.

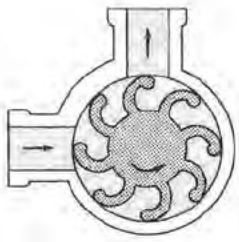
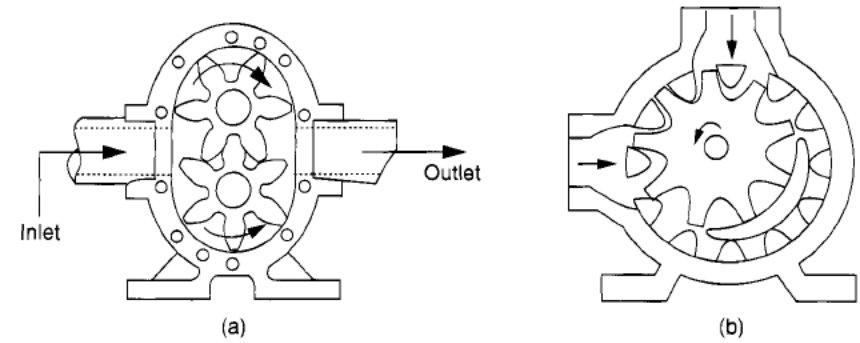


FIGURE 4 Flexible vane pump

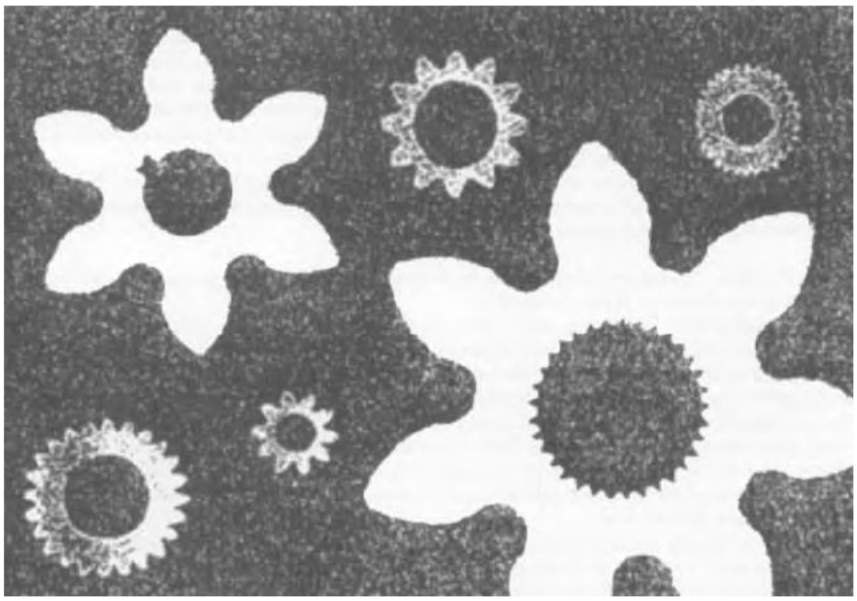


FIGURE 5 Flexible liner pump

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 124 of 153



(a) (b)



(c)

FIGURE 6 (a) external gear pump (Roper), (b) internal gear pump (Viking), and (c) various types of gears used (Roper)

Internal gear pumps, on the other hand, have one larger gear (rotor) with gear teeth cut internally on the major diameter meshing with and driving a smaller externally cut gear (idler). Pumps of this type can be with or without a crescent-shaped partition to define the OIT, CTIO, and OTO zones.

The OTI volume of the pump chamber in gear pumps is defined by the body walls and by where each tooth tip meets and seals with the body walls as it leaves the OTI volume. The fluid trapped between the gear teeth and the body walls is sealed from both inlet and outlet chambers and is the CTIO volume. The OTO volume is defined by the body walls and the gear tooth surfaces between the fluid seal points where each tooth tip leaves the body wall and enters the OTO volume and fluid seal points where the gears mesh.

A part or all of the side (or axial) surfaces of the gears run in small-clearance contact with the axial end faces of the pumping chamber. The gear teeth run in small-clearance

contact with each other where they mesh. The tips run in small-clearance contact with the radial surfaces of the pumping chambers in their travel from the OTI to OTO volume. Load-bearing contact between the rotors or between the rotors and the stator may exist in all three of these zones, and the apertures defined by the running clearances in these zones determine the amount of slip between the OTO and OTI volumes for any given pressure difference and viscosity between them.

Both gears share pumping torque, and the proportional amount of the total torque experienced by each gear at any instant is determined by the locus of the fluid seal point between the gear teeth. As this fluid seal point moves toward the center of gear rotation, the pumping torque on that gear increases, and as the seal point moves away from the center, the torque decreases. When external timing gears are used, they transfer torque from one rotating assembly to another to safeguard against accelerated wear when dry running or handling low-lubricity or somewhat more abrasive fluids (e.g., foodstuffs). Here, bearing (bushing) life depends on projected loading and journal velocity. A product of projected loading (P, psi) times journal velocity (V, ft/min) is used to determine the ability of internally (product) lubricated bearings to operate without significant wear. In the example given with Figure 7, the idler shaft of the gear pump is shown supporting its gear and is subject to the differential pressure, resulting in the projected bearing loading as illustrated in the figure. Data exist for various material pairs running against each other. Bronze, iron, carbon, and silicon carbide are some examples of popular materials used for the internally lubricated bushings of rotary pumps. For bronze, the PV-value

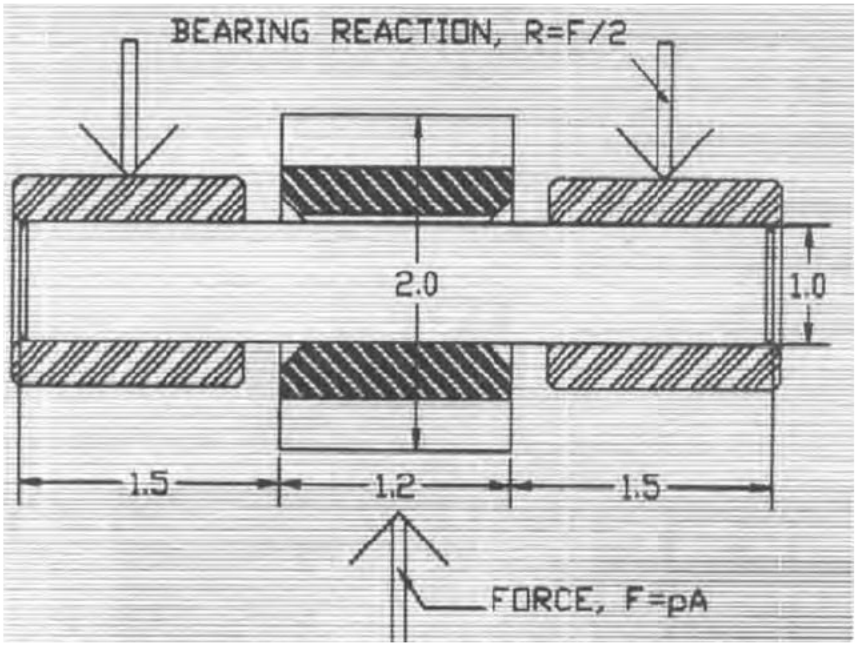


FIGURE 7 Example of dry-running PV-evaluation for a gear pump bushing. Given: rpm = 1800; suction pressure = 6 psia (0.414 bara); discharge pressure = 128 psig (8.825 barg); and dimensions are shown in inches (mm = in × 25.4). Thus, Δp = (128 + 14.7) − 6 = 136.7 psi (9.425 bar); and force on gear = 136.7 × (2 in × 1.2 in) = 328 lbf (1459 N). Half of this force is shared by each bearing, namely 164 lbf (730 N). The unit load on each bearing P = 164/(1.5 in × 1 in) = 109 psi (7.54 bar). Since the journal velocity V (ft/min) = 2π × rpm × d (ft)/2 = 2π × 1800 × [(1 in/12)/2] = 471 ft/min (2.39 m/s), PV = 109 × 471 = 51,400 psi-ft/min (18 bar-m/s). (Although the result is given here also in SI units, the PV-loading of a bearing is generally correlated in terms of psi-ft/min.)

is 60,000; for iron it is under 30,000; and carbon is rated at approximately 120,000. Some grades of silicon carbide reach PV-values exceeding 200,000. On the other hand, plastics have very low values of PV, typically under 3000.

External gear pumps are capable of flows up to 1,500 gpm (5,680 l/min), pressures up to 500 lb/in^2 (34.5 bar), and viscosities up to 1,000,000 SSU (216,000 centistokes). They are found in both clean and dirty services serving the *original equipment manufacturer* (OEM), refinery, tank farm, marine, and API-related industries. Internal gear pumps are capable of flows up to 1,100 gpm (4,165 l/min), pressures up to 225 lb/in^2 (15.5 bar), and viscosities up to 1,000,000 SSU (216,000 centistokes). They are typically used for lower pressure transfers of fuel oils, paints, and various chemicals in the chemical processing and OEM industries.

Lobe Pumps The lobe pump (Figure 2) receives its name from the rounded shape of the rotor radial surfaces that permits the rotors to continuously overlap each other as they rotate. Lobe pumps can be either single- or multiple-lobe pumps and carry fluid between their rotor lobes much in the same way a gear pump does.

Unlike gear pumps, however, neither the number of lobes nor their shape permits one rotor to drive the other, and so all true lobe pumps require timing gears. The body surfaces, rotor surfaces, the contact between rotors, and the contact between rotor lobe ends and the pump body define the OTI volume of a pump. The contact between the lobe ends and the body wall and the adjoining body wall and lobe surfaces define the CTIO volume. The body walls, rotor surfaces, lobe-to-body wall contacts, and the lobe-to-lobe contacts define the OTO volume.

In the two-rotor lobe pumps shown in Figure 8, the torque is shared by both rotors with the proportional amount of torque dependent on the position of the rotor-to-rotor contact point on the rotor contact locus. When the contact point is at the major locus radius (maximum lobe radius of one rotor in contact with the minimum lobe radius of an adjoining rotor), one rotor sees the full pumping torque, while the other rotor feels a balanced torque. The transfer of a full pumping torque from one rotor to the other takes place as many times in each complete revolution of a rotor as there are lobes on the rotor.

Lobe pumps are capable of flows up to about 1,000 gpm (3,785 l/min) and pressures up to 225 lb/in^2 (15.5 bar). They are commonly used to pump sludge in wastewater treatment plants and in stainless steel systems for handling foodstuffs in the food, beverage, dairy, and pharmaceutical industries.

A summary of the main application parameters of vane, gear, and lobe pumps is shown in Table 1 together with similar data for other rotary pumps; e.g., screw pumps (Section 3.7).

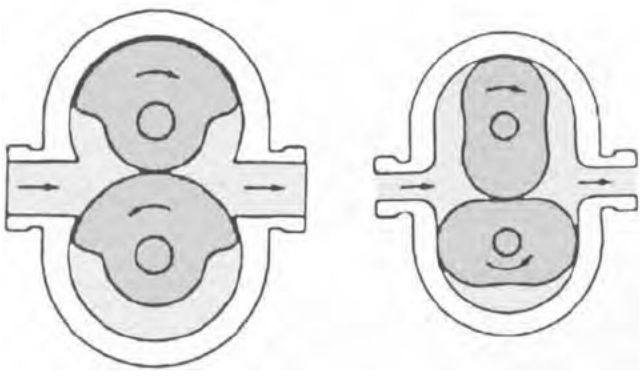


FIGURE 8 Typical single-lobe (left) and multiple-lobe (right) pumps

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 127 of 153

TABLE 1 Application Matrix for Rotary Pumps

Pump Type	Flow, gpm	Pressure, psi	Temperature, °F	Viscocity, cP	Solids, %	Dry-run	Corrosion resistance	Pulsations	Lift, ft	Price index
External gear	1 to 1000	3000	400	20 to 10,000	<1%	N	G	G	15′	110%
Internal gear	1 to 1000	3000	400	10 to 10,000	<1%	N	G	G	15′	110%
Lobe	50 to 1000	400	400	100 to 500,000	<20%	Y	G	G	10′	150%
Vane	50 to 1000	200	250	0.3 to 1,000	<1%	N	M	M	10′	100%
Diaphragm	50 to 1000	100	200	1,000 to 100,000	<20%	Y	E	B	20′	130%
Progressing cavity	100 to 1000	500	200	1,000 to 100,000	<80%	N	B	E	15′	200
Two-screw	100 to 8000	1500	650	1,000,000	<1%	N	M	G	25′	400%
Three-screw	10 to 3500	4500	550	1,000,000	<1%	N	M	E	25′	300%
Metering (diaphragm)	<1	200	400	1 to 1,000	<1%	N	E	B	5′	100%

Note: m^3/h = gpm/4.40; bar = psi/14.5; deg C = (deg F − 32)/1.8; Pa-s = cP/1000; m = 0.3048 × ft.) Values are approximate, for rough guidance only. *Code Letters: Excellent; Good; Marginal; Bad*

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 128 of 153

COMPARISONS TO CENTRIFUGAL PUMPS

A rotary pump uses mechanical and hydraulic forces to create a flow against a system backpressure. Centrifugal pumps, on the other hand, create pressure by imparting a velocity to the fluid and converting the velocity energy to a pressure energy as the fluid flows around the casing and out the discharge nozzle. A comparison of the resulting performance of these two different pump types is shown in Figure 9.

The conditions of service will usually determine the best pump for an application. For instance, for varying flow rates, a centrifugal pump would be a good choice. An example of this is a municipal water system where consistent head is maintained. By contrast, for a constant flow in the presence of varying back pressures, a rotary pump would be better. An example of this is an oil pipeline, where system economics dictate constant flow rates, regardless of any system pressure variations from changes in viscosity or the pipe diameter or condition.

Other differences exist between centrifugal and rotary pumps as well. The performance curves, affinity laws, and terminology used to describe rotary pumps are all different. And since rotary pumps are primarily for viscous fluids, the applications and markets served by these two pumps are also different. Other factors, such as vertical versus horizontal mounting, metal versus nonmetallic materials, sealless and magnetically driven versus dual-containment mechanical seals and conventional drivers, are similar whether considering a centrifugal or rotary pump.

Rotary Pump Curves Centrifugal pump curves plot the flow on the X-axis with the discharge head on the Y-axis. However, rotary pumps develop the flow against a system backpressure, rather than developing head with a corresponding flow rate. Their performance curves therefore show the flow on the Y-axis with differential pressure along the X-axis, as shown in Figure 10.

The influence of differential pressure on the flow is greatest with lower viscosity. Conversely, the maximum viscosity represents the worst (most) case from a power standpoint and is therefore the point around which the driver is sized.

Instead, a rotary pump is selected for a given differential pressure and viscosity at the nearest commercially acceptable speed (with a tendency toward the lower speeds). For viscosities under 3,000 SSU (650 centistokes), this means at synchronous motor speeds with the flow rate falling at the design point or (ideally) slightly above it. For higher viscosities,

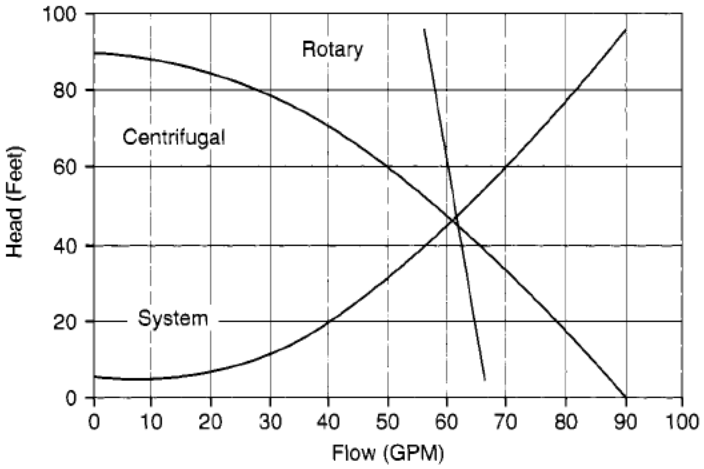


FIGURE 9 Typical performance curves (m = ft × 0.3048; m^3/h = gpm/4.403)

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 129 of 153

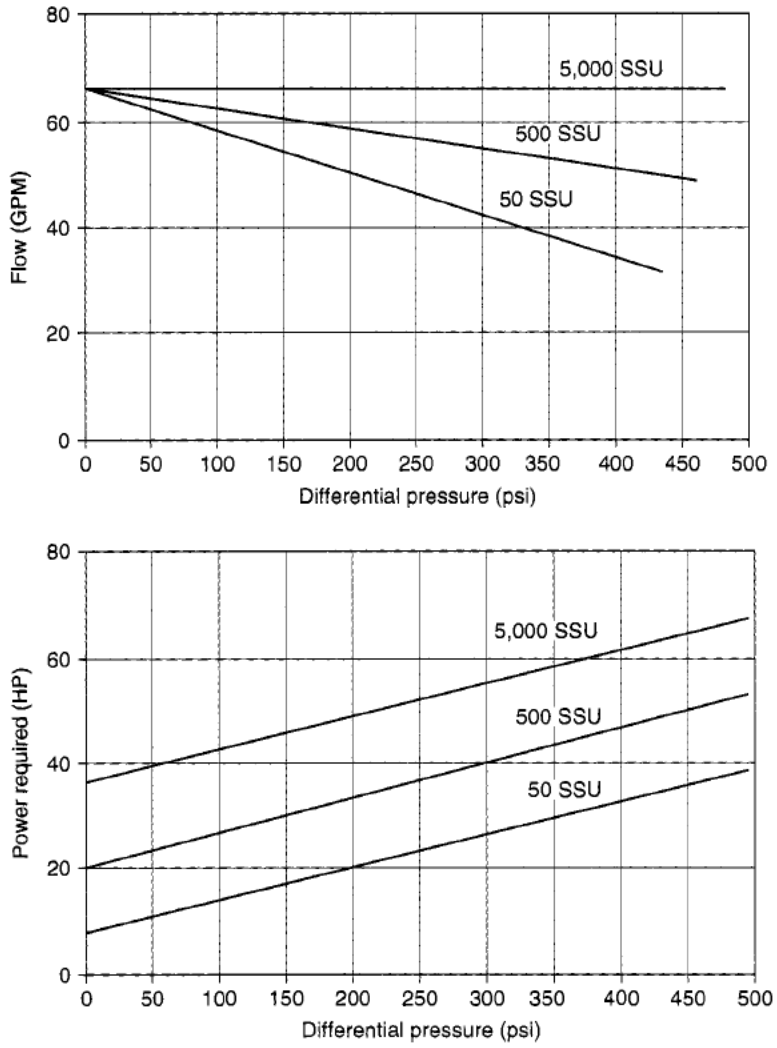


FIGURE 10 Typical rotary pump flow and power curves (bar = psi/14.5; kW = hp × 0.746; 50, 500, 5000 SSU = 7, 108, and 1080 centistokes respectively.)

rotary pumps will be run at reduced speeds, which in some cases can go well below 100 revolutions per minute (rpm).

Performance with Viscous Fluids Viscos drag can be considerable with higher viscosities. With an increasing viscosity, an increasing amount of energy must be expended to overcome these forces and produce the same amount of hydraulic work. At some point, viscosity will simply overtake the centrifugal pump, and it can no longer overcome the inertia and viscous drag losses of the fluid. The effect of this can be seen in Figure 11.

Since rotary pumps do not work this way, they are better suited for high-viscosity fluids. Certain rotary pumps, such as multiple screw pumps, can even go as high as 1,000,000 SSU

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 130 of 153

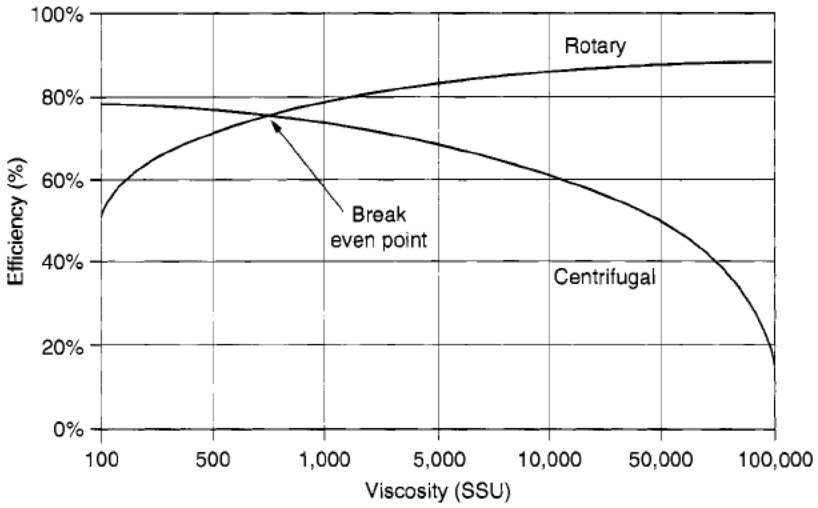


FIGURE 11 Typical efficiences as a function of viscosity. (In this range, centistokes ≈ SSU/5.)

(216,000 centistokes) without a significant deterioration of performance or efficiency. Compared to centrifugal pumps, rotary pumps are less efficient at lower viscosities but more efficient at higher viscosities. From an applications standpoint, this crossover point will vary from pump to pump and depends on many factors, but as a rule of thumb, it usually falls between 500 and 1,000 SSU (108 and 216 centistokes).

Mechanical Installation All rotary pumps, particularly rigid rotor pumps, must be installed so no mechanical forces other than those imposed by pump-generated pressures can act to warp or distort the pump chamber or rotating assembly. This is important because relatively small distortions of a few thousandths of an inch can cause interference between rotating and stationary parts and generate high wear rates or pump damage.

To avoid such distortions, the pump must not be installed with overly long, rigid fittings or in such a way that the pump body supports the weight of the piping system. Large mechanical forces that distort the pump body may also cause distortions of the mechanical seals and accelerate wear of the bearings. A cardinal rule, then, is that a rotary pump must not be rigidly coupled to the piping system in a way that causes it to support the weight of the piping or otherwise exposes it to any forces from thermal expansion. Poor mechanical installation is the cause of many field problems.

Dry Running Most rotary pumps will be damaged if allowed to run dry. As long as the system provides a positive static pressure on fluid present at the inlet, the pump should prime and not run dry. If a negative inlet gage pressure, or suction lift, is present at the pump inlet, the installation should be checked to ensure airtight seals throughout the entire upstream piping.

The sealing arrangement on the pump must also be checked for any leakage of air or gas into the pump through the seal. If these conditions are satisfied, fluid will enter the pump soon after pump operation starts because of the vacuum generated by pump operation. If these conditions are not met, air or other gases will flow through leaks in the inlet system or seal to satisfy the pump flow rate requirements and the pump will run dry or partially dry.

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 131 of 153

Suction Strainers Like most other pumps, rotary pumps last longer when handling clean fluids. Nonetheless, this is an ideal scenario, and the pump will more realistically encounter dirty or abrasive-laden fluids of varying degrees. Fine particles and abrasives will cause wear in the close clearances of the pump, which eventually reduces pump flow rate by increasing the slip through the increased clearances.

As such, all rotary pumps should have a suction strainer to exclude larger materials such as welding slag, scale, rust, chips, rags, bolts, nuts, and so on. Since a suction strainer contributes to suction line losses, this reduces the net inlet pressure available. The finer the filtration, the greater the restriction and the more frequently it must be maintained. This leads to a trade-off between the cost of the added maintenance versus the cost of replacing the worn pump parts earlier than they would be otherwise.

When pumping fluids over 5,000 SSU (1080 centistokes), the finest strainer screen practical is a $\frac{1}{16}$-inch (1.5 mm) perforation. Strainers and filters not only require periodic maintenance, but should also be instrumented accordingly. It is important for the user to provide some means of monitoring, such as a differential pressure gauge or switch, since a clogged strainer will cause the pump to cavitate or even run dry.

Entrained Air and Dissolved Gases An important consideration with rotary pumps is the amount of entrained air or gas in the fluid. It is generally neglected, since a rotary pump cannot become vapor-bound the way a centrifugal pump can. Nonetheless, with entrained air present, there can be a loss in the outlet flow. If the entrained air is a large enough percentage, there may be unacceptable noise and vibration levels as well.

For example, if a fluid contains 5% entrained gas by volume and the suction pressure is atmospheric, the mixture is 95% liquid and 5% gas. This mixture fills up the moving voids on the inlet side, with 5% of the space filled with gas and the remainder with liquid. Therefore, in terms of the amount of liquid handled, the output is reduced directly by the amount of gas present, or 5%. Unless this is understood up front, it could lead to a less than satisfactory output flow rate through no direct fault of the pump itself.

Entrained air is common in systems where the liquid is cycled frequently. In many cases, the foaming or air entrainment cannot be avoided, such as with the lubrication system on a large reduction gearbox. Instead, the condition must be known and well understood before selecting a pump for the application.

If dissolved gases (gases different than the fluid's own vapor) are present in the fluid, the effect on the output flow is the same as with entrained gases. This is because the dissolved gases will come out of solution when the pressure is lowered, just as the fluid's own vapor will. This will have the same net effect as the entrained gas and will occupy the available displacement capacity. Although the fluid mass transfer rate will not be affected, this is likely to be small comfort since the measured liquid displacement will be reduced.

Noise Pumps are often the most offensive noise sources in hydraulic machinery. High-pressure pulsations and heavily loaded sliding elements within the pump produce broadband, high-energy airborne noise. Vane, gear, and lobe pumps (as well as the screw pumps of Section 3.7), however, are among the lowest noise producers of any fixed displacement-type pump. Flow is delivered continuously without the variations that produce noise in conventional hydraulic pumps. Pumping elements utilize a fluid film, reducing the sliding contact, and the visco-elastic properties of the fluids they pump help dampen whatever fluid-borne pressure pulsations are present. These design features are responsible for the wide use of these pumps wherever noise is critical. For instance, they are widely installed on die-casting machines, plastics equipment, presses, and an enormous variety of machine drives and machine tools.

Inlet Pressure The absolute pressure above the vapor pressure *available* at the pump inlet must always exceed the absolute pressure above the vapor pressure *required* by the pump.

Rotary pumps are not effective at stopping the flow through them. In applications where the flow cannot be permitted through an idle pump, or where inlet or outlet static

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 132 of 153

pressure heads exist, check valves in the system must be used to stop the flow. For example, in intermittent deliveries where the pump is "lifting" liquid from a source below its inlet without any valving in place to address this when the pump is stopped, the fluid will gradually drain backward through the pump and back to its source.

Cavitation The creation of vapors and the subsequent collapse of the vapor bubbles upon reaching the higher-pressure discharge side of the pump is known as *cavitation*. Cavitation forces the liquid into the vapor voids at high velocities and produces local pressure surges of high intensities impinging on the pump surfaces. These forces can exceed the tensile strength of the metal, eroding it in the process. Left uncorrected, cavitation can cause pitting of the vanes, gears, or lobes and interior casing walls; bearing failures; and even shaft breakage. In addition, cavitation causes noise, vibration, and a loss of output flow. The bigger the pump, the greater the noise and vibration can be.

Lowering the static suction lift of a system, increasing the suction pipe diameter, and simplifying the suction piping layout can reduce cavitation by raising the net inlet pressure or NPSH available. In reality though, the net inlet pressure available is not something that can easily be altered. Instead, the pump vendor will usually be asked to select a pump with a lower net inlet pressure required. The best way to do this is to select a larger pump running slower. This will give the fluid more residence time to fill the void on the suction side, and with larger internal passages and ports, pump entry losses will be reduced as well.

FLUID CONSIDERATIONS

Very few commercially handled fluids are homogenous liquids. In actual systems, most fluids have air or other gases dissolved or entrained in them. In other cases, solids (abrasive or nonabrasive) may be in the fluid. The variety of fluids handled by rotary pumps requires that each pump application be uniquely considered in terms of the effects of the pumped fluid on pump performance. General summaries of the effects of various fluid characteristics on pump performance are covered in the following paragraphs.

Temperature The temperature of the pumped fluid affects pump performance in three main ways. If the pump is to handle fluid at temperatures considerably different from ambient ones, the materials of construction, both in the pump and in the seals, and the operating clearances of the pump must be selected to provide the desired operating characteristics at the temperature of operation. The selection process becomes even more stringent when the pump is operated over a wide range of temperatures. Such a use may preclude the use of pump construction materials having high thermal coefficients of expansion. Furthermore, in the actual application, it may be necessary to preheat or precool the pump to the operating temperature before the pump is started in order to avoid thermal shocking the internal components when the fluid enters the pump. If this is not done, the resulting rapid heating or cooling of the pump members from the inside out can damage the pump. Preheating or pre-cooling can usually be accomplished with a secondary medium such as a heating jacket.

Viscosity Another effect of fluid temperature is on the viscosity of the fluid. For the majority of the commercial liquids, the viscosity increases with decreasing temperature and decreases with increasing temperature. If the pump application requires usage over a wide temperature range, the highest viscosity at this temperature range must be known to determine whether the pump is operating below the upper speed limit imposed by this fluid viscosity. The effects of viscosity on speed and the net inlet pressure (above the vapor pressure of the liquid) required is shown in Figure 12.

Lubricity Still another important fluid characteristic is lubricity. Many rotary pumps are not designed to operate on liquids with no lubricity. Either the wear rate or the pump mechanical friction may be inefficiently high if these pumps are used on nonlubricating

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 133 of 153

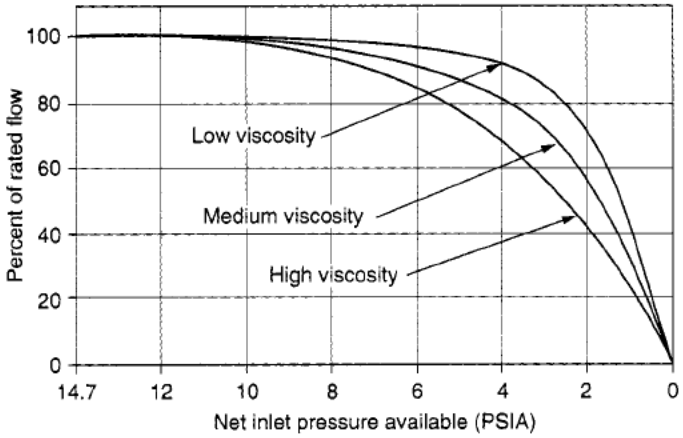


FIGURE 12 How viscosity and net inlet pressure available influence flow (bar = psi/14.5. Expressed as head of pumped liquid, the net inlet pressure is called NPSH.)

fluids. Vane pumps are the exception here; for they can handle even liquid CO_2, a very poor lubricating substance.

Non-Newtonian Fluids The viscosity of most fluids is unaffected by agitation or shearing as long as the temperature remains constant. They are known as *true* or *Newtonian fluids*. Certain fluids, however, change viscosity as a shear is applied at a constant temperature. The viscosity of these fluids will depend upon the shear rate at which it is measured, and these fluids are termed *non-Newtonian*. Some examples are cellulose compounds, glues, greases, paints, starches, slurries, and candy compounds.

If a fluid is non-Newtonian, the viscosity under actual pumping conditions must be determined and can vary quite a bit from the viscosity under static conditions. An example is grease, where the static viscosity is around 20,000 SSU (4300 centistokes). Under actual pumping conditions, however, the viscosity is closer to 500 SSU (108 centistokes). Without realizing this, a larger, slower pump may have been offered when a smaller, less expensive pump running at a higher speed would have been acceptable.

Tests or computations should determine the effective fluid viscosity under actual operating conditions. If the pump is to operate over a range of speed and pressures, the maximum and minimum effective viscosities over this range should be known to allow for the additional slip or horsepower that may be required and to ensure the correct operating speed and pump driver power selections. If these data are unavailable, as is often the case, then data from a similar installation can be helpful. Ideally, the information will include the pump size, flow rate, and speed as well as the NPSH available and the pressure drop over a specific length of piping. With this information, assumptions can be made regarding the actual viscosity.

Corrosiveness Knowledge of the corrosiveness of the pumped fluid on the pump materials in contact with the fluid is important to satisfactory pump application. The clearances in rotary pumps are small, and corrosion rates of only a few thousandths of an inch per year may seriously affect the efficiency of the pump, particularly when it is handling low-viscosity fluids. In addition, general compatibility of the fluid with the materials of pump construction must be considered. For example, a certain solvent to be pumped may soften, dissolve, or destroy the elasticity of flexible members in the pump, including those flexible members used in the sealing arrangements. Hence, the effect of the pumped fluid on the physical and chemical state of materials used in pump construction should be considered when choosing a particular application.

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 134 of 153

Abrasiveness Abrasives present a problem for every pump type, and rotary pumps are particularly vulnerable. With small operating clearances, the tendency of the abrasive is to wear down and open the tight clearances of the pump. Most rotary pumps' internal clearances are long, narrow, rectangular cross sections that can be modeled as two parallel flat plates, with one plate stationary and the other moving. These clearances range from essentially zero to a few thousandths of an inch. Thus, even minor variations in manufacturing tolerances or the effects of wear over time can cause considerable variations in the percentage change of the aperture volume.

Because rotary pumps depend upon close clearances for their pumping action, whenever abrasive or dirty fluids are encountered, accelerated wear can be expected. Wear is difficult to predict because it depends on many variables that are difficult to measure (such as particle size, size distribution, or whether smooth or jagged shaped).

Classic theory tells us that laminar slip flow Q_s between two plates follows the general equation:

$$Q_s = k\left\{\frac{P_d \times w \times d^3}{v \times l}\right\} \tag{1}$$

where Q_s = slip through the clearance, gpm (m^3/hr)
k = a constant
P_d = differential pressure, lb/in^2 (bar)
w = width of clearance, in (mm)
d = clearance gap or depth, in (mm)
v = absolute viscosity, lb-sec/ft^2 (centipoises) [Note: (absolute viscosity in centipoises) = (kinemetic viscosity in centistokes) × sp. gr.]
l = length of clearance, in (mm)

Rotary pumps rely on close running clearances to seal between the suction and discharge pressures. This presents certain manufacturing trade-offs. For instance, the clearances must be tight enough to yield an acceptable volumetric efficiency with lower viscosities, yet large enough to facilitate internal lubrication with higher viscosities. For a given pump with a fixed viscosity and differential pressure, the slip flow is a function of the cube of the clearances only.

Unlike a centrifugal pump impeller, which can be trimmed to about 85% of its maximum diameter, rotary pump gears, screws, or lobes cannot be trimmed. As Figure 13 shows, up to a certain critical point, wear has little effect on the flow. Beyond this point, which varies from case to case, performance starts to deteriorate rapidly.

Much progress has been made in the use of harder and more abrasive-resistant materials for the pumping elements. However, rotary pumps should not be used for abrasive fluids, unless a decreased pump life and/or increased maintenance intervals are acceptable.

Other fluid characteristics could be important in the selection and application of vane, gear, and lobe pumps. Some liquids are sensitive to shear and must be handled in pumps with low shear rates. Others cannot tolerate exposure to the atmosphere, either because they can explode, crystallize, and damage the seal arrangement or otherwise interfere with the pump operation. Rotary pumps for these applications should have only static seals or multiple-section rotary seals with a protective liquid or gas in the seal zones between the fluid and outside atmosphere. In aseptic pumps, for example, stationary and moving seals can be designed with multiple seals to enable the use of steam under pressure as a sterile barrier between the pumping chamber and the atmosphere. Pumps in industrial applications where it could be dangerous if the fluid were exposed to the atmosphere may use multiple sealing arrangements with an inert liquid under pressure acting as a barrier between the pumping chamber and the outside atmosphere.

For the effective application of rotary pumps, the temperature, viscosity, lubricity, corrosiveness, and nonliquid content of the pumped fluid must be known, along with any special characteristics of the liquid, such as shear or atmospheric sensitivity. With these known, the proper pump type, seal arrangements, horsepower requirements, and speed can be determined.

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 135 of 153

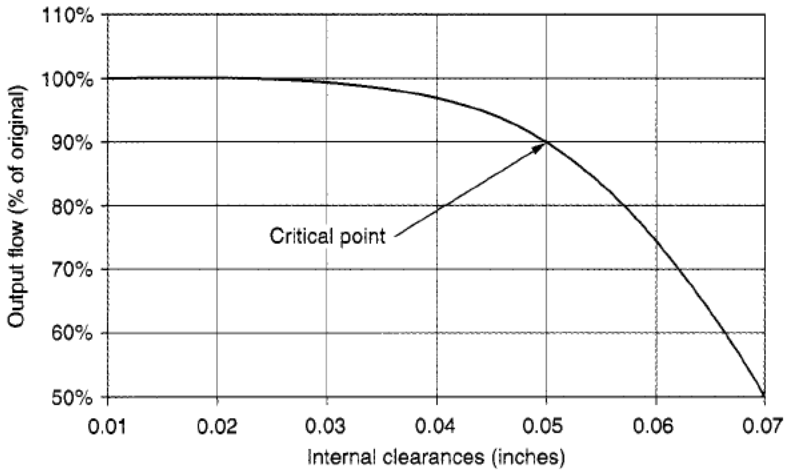


FIGURE 13 Output flow rate as a function of clearance size and wear

Vapor Pressure A second effect of fluid temperature is on the vapor pressure of the fluid. Vapor pressure is the pressure that must be maintained at a given temperature to prevent the liquid from partially vaporizing into a gaseous state. Vapor pressure increases with temperature, and if the local pressure falls below the vapor pressure, vaporization and cavitation will occur.

Vapor pressure is particularly important when handling hydrocarbons and petrochemicals, which can have very high vapor pressures. For instance, low-sulfur crude oil vapor pressures can be as high as 100 lb/in^2 absolute (6.89 absolute) under summer ambient temperatures. Unless this is known when selecting a pump, the results could be a pump that works in cold weather but cavitates in warm weather.

OPERATING CHARACTERISTICS

The operating characteristics covered in this section assume the fluid is a true incompressible liquid with a viscosity independent of the rate of shear (shear strain). Common fluids used in testing are either a light lubricating oil or cool water with small amounts of soluble oil added for lubricity.

Displacement The *theoretical* or *geometric displacement* q_0 of a rotary pump is the fluid volume transferred during each revolution. For any given pump, the displacement depends only upon the physical dimensions of the pump elements and the pump geometry and is independent of other operating conditions. In those pumps designed for variable displacement, the pump usually is rated at its maximum displacement. Thus, neglecting slip, the *theoretical displacement volume flow rate* of the pump Q_d is the *theoretical displacement per revolution* q_0 multiplied by the rotative speed or rpm; namely, $Q_d = q_0 \times rpm$.

Slip Flow slip, Q_s, is an important aspect of rotary pump performance. It is defined as that quantity of fluid that slips from the OTO volume to the OTI volume per the unit of time. Slip is a function of the clearances between the rotating and stationary members, the differential pressure between the OTO volume, the OTI volume, and the fluid viscosity. Hydraulically, it is equivalent to a bypass line from the pump outlet back to the inlet. The most common unit of measuring slip is U.S. gallons per minute (cubic meters per hour). Slip is essentially independent of *RPM*.

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 136 of 153

The major slip paths through the pump in the presence of a positive differential pressure are the clearances between the end faces of the rotors and the endplates of the pump chamber, and those between the outer radial surfaces of the rotors and the inner radial surfaces of the chamber. If the differential pressure across the pump remains constant during a revolution, then the instantaneous slip rate usually varies throughout the revolution. This variation in the slip is caused by the same effect that would be produced if the physical dimensions of the equivalent bypass around the pump were varied as a function of the angular rotation of the drive shaft. This is also one of the common causes of flow pulsation in rotary pumps. It is particularly dominant when pumping low-viscosity fluids at high pressures.

The average slip for any set of operating conditions can be found by measuring the flow rate from the outlet port (assuming an incompressible liquid) and subtracting that flow rate from the theoretical displacement flow rate Q_d that would otherwise be expected at those operating conditions. The paths can also vary considerably in length with the changing positions of the rotary and body-sealing surfaces during rotation.

The effect of pressure on the slip is complex. The primary effect is direct in that slip increases in direct proportion to pressure. However, several secondary effects should be considered as well. The first is the effect of pressure differences across the pump on the dimensions of the slip path. This occurs because of the deflection of pump elements as a function of pressure. This is relatively small in rigid element pumps but can be significant in flexible vane pumps where the pressure can cause the vanes to flatten out and move away from the body walls. In addition, while the slip may increase in flexible member pumps at high pressures, in rigid rotor pumps it can actually decrease as clearances close, due to the high-pressure deflection of the rotors.

Another secondary consideration is the indirect effect of pressure on the fluid velocity through the slip paths. At any given viscosity, the flow through these paths can have the characteristics of turbulent, laminar, or slug flows. The majority of practical applications would require that the slip be a minor percentage of the pump displacement. To remain so, the velocity of fluid flow through slip paths would normally be in the laminar flow region, and the slip would then be directly proportional to the pressure difference. A pressure increase could cause a change to turbulent flow and a corresponding change in the slip as a function of pressure.

Also, an indirect effect of pressure exists on the effective compression ratio of compressible fluids. The compression ratio reduces the amount of net volume flow through the outlet port relative to the displacement of the pump. Although not a true slip in the sense discussed up to this point, the type of slip caused by this effect reduces the net volume delivered through the outlet port and consequently affects the volumetric efficiency. This effect is a secondary effect in most liquids but can become a large component of the slip in aerated or compressible liquids. An increase in the compression ratio caused by an increase in the pressure difference causes an increase in slip from this effect.

Flow Rate and Displacement The flow rate or *capacity* Q_c of a rotary pump is the net quantity of fluid delivered by the pump per the unit of time through its outlet port or ports under any given operating condition. When the fluid is incompressible, the flow rate is numerically equal to the total volume of liquid displaced by the pump per the unit of time minus the slip, all expressed in the same units. When a rotary pump is operating with zero slip, the theoretical or geometrical displacement Q_d of the pump becomes the flow rate Q_c. A common unit of flow rate is U.S. gallons per minute (cubic meters per hour):

$$Q_c = Q_d - Q_s \tag{2}$$

The theoretical displacement q_0 per revolution represents the fluid confined within the gear spaces and can be found by integrating the differential rate of a net volume transfer over one shaft revolution with respect to the angular displacement of the drive shaft through any complete planar segment taken through the pump chamber between the inlet and outlet ports. This integration depends on the gear shape, which is defined by the involute; and so the exact calculation of this volume is mathematically involved. Therefore, in most practical applications, an approximation can be made with reasonable accuracy, as the following example illustrates.

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 137 of 153

Gear pump displacement calculation (approximate method). Assume an external gear pump with two gears measuring 3 in (7.62 cm) OD, and 2 in (5.08 cm) ID (root), and 4 in (10.16 cm) gear width W. Assuming that the liquid occupies half of the volume between the OD and ID cylinders (the other half being the solid material of the teeth), the volume of fluid between OD and ID for the two gears is

$$2 \times \frac{1}{2} \times \frac{\pi}{4} \times (\text{OD}^2 - \text{ID}^2) \times W = 15.71 \text{ in}^3 (257.4 \text{ cm}^3) \text{ per revolution.}$$

Therefore,

$$q_0 = \frac{15.71}{231 \text{ in}^3/\text{gal}} = 0.068 \text{ gal/rev.} = \frac{257.4}{1{,}000{,}000 \text{ cm}^3/\text{m}^3} = 0.0002574 \text{ m}^3/\text{rev.}$$

For a rotative speed of 1000 *rpm*, the *theoretical displacement flow rate* Q_d (which does not account for slip) thus becomes

$$Q_d = q_0 \times \text{rpm} = 0.068 \times 1000 = 68 \text{ gpm} = 0.0002574 \times 1000 \times 60 \text{ min/h} = 15.44 \text{ m}^3/\text{h}$$

From a practical standpoint, pump displacement often cannot be computed precisely from the geometry of the pump and is instead established by testing. In either case, the manufacturer will usually state this because it is needed to determine the efficiency of the pump under various operating conditions.

Speed Considerations Centrifugal pumps typically run at synchronous motor speeds. The flow and head are fine-tuned by such things as trimming the impeller or underfiling the impeller vanes. Rotary pumps, on the other hand, operate with their internal dimensions fixed. Performance is usually fine-tuned externally by speed adjustments where the operating speed N is the number of revolutions of the driving or main rotor per unit time. When no gear reduction or increase exists between the drive shaft and the main rotor, the speed is measured or set at the drive shaft.

The most common unit of speed is rpm, and its direction is usually described as either clockwise or counterclockwise when viewed from the front or drive end of the pump looking from the driver.

Usually, no minimum speed exists for a rotary pump, but certain vane pumps that depend on a centrifugal force to draw the vanes out of their slots do have minimum speed requirements. Another exception is with low-viscosity fluids at high pressures where the slow speed enables slip to equal the theoretical displacement, resulting in no net output flow. To avoid this situation and assure the pump is running at a suitable volumetric efficiency, the pump should always run within its permitted speed range. Figure 14 shows a typical speed versus viscosity curve for a rotary pump.

Since flow is directly proportional to speed, it can be tempting to increase the pump speed above its maximum speed to obtain more flow. However, besides all the aforementioned concerns, one other item must be considered. With any rotary pump, as the cavities rotate, they present a void for the incoming fluid to fill. This void is available for a fixed amount of time, and the fluid in the suction chamber must accelerate to fill this void in the available time. The higher the fluid viscosity, the more energy is required to accelerate the fluid to fill the void. If the fluid cannot fill the void in the time available, a partial void and cavitation will occur. Recognizing a need to limit pump speed, most manufacturers provide charts showing the maximum speed, which decreases with increasing viscosity of the liquid. This speed is related to the required excess of the inlet total pressure P_1 above the vapor pressure p_v of the liquid (called NPSH when expressed in terms of the equivalent head of the liquid) increases with both the pitch line velocity V_p of the gears and the viscosity. Thus, if the available level of P_1 is limited, the pump speed is also limited accordingly. For external gear pumps, this relationship can be approximated by the formula:

$$P_1 - p_v = V_p^{0.8} \times \nu^{0.1}/61 \text{ psi (Multiply by 6.895 to obtain the answer in kPa)}$$

where ν is the kinematic viscosity in SSU, and

$$V_p = D_p \times \text{rpm} \times 3.1/n_t$$

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 138 of 153

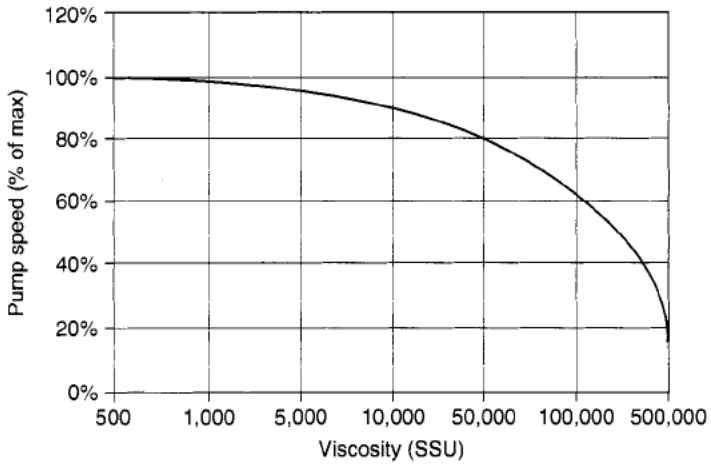


FIGURE 14 Typical speed reduction as a function of viscosity (In this range, centistokes = SSU/4.6)

and D_p is the pitch diameter of the gears in inches (= mm/25.4) and n_t is the number of teeth on a gear.[1]

Pressure The absolute pressure of the fluid at any location in the pump, expressed in lb/in^2 (bar), is the total pressure there and the basis for all other pressure definitions associated with pump operation. Helping to simplify matters, the velocity pressure component of the fluid is usually small enough relative to the total pressure to be neglected.

The *inlet* or *suction pressure* p_{in} is the total pressure at the inlet to the pump. In common practice, the inlet pressure can be variously expressed as absolute pressure (lb/in^2 absolute or bar), as positive or negative gage pressure (lb/in^2 gage or bar), or as vacuum (inches or millimeters of mercury).

The total *differential pressure* Δp (or P_d) is the algebraic difference between the outlet pressure and the inlet pressure, with both expressed in the same units. This differential pressure is used in the determination of power input and in evaluating the slip characteristics of the pump:

$$\Delta p = p_{out} - p_{in} \tag{3}$$

The *net inlet pressure* $p_{net\,in}$ of a rotary pump is the difference between the inlet pressure expressed in absolute units and the vapor pressure of the fluid expressed in absolute units:

$$p_{net\,in} = p_{in} - p_{vapor} \tag{4}$$

Expressed as head of the pumped liquid, this is the net positive suction head, NPSH. The required NPSH or *net inlet pressure required* p_{nipr} is the minimum net inlet pressure that can exist without the creation of enough vapor in the inlet to interfere with proper operation of the pump. The inlet pressure usually is not the true minimum pressure in the OTI volume. The flow of fluid in its passage through the OTI volume causes a fluid friction pressure loss, which causes the pressure to drop below the inlet pressure at some point in the OTI volume or inlet chamber. This loss increases with fluid velocity and hence with pump speed and fluid viscosity. It is a function of the pump geometry, which determines the fluid path lengths and local velocities where the fluid changes direction in flowing around curves or corners in the pump OTI volume boundary.

The required net inlet pressure is established by the manufacturer for particular speeds, pressures, and fluid characteristics. In practice, the pump user is warned that the

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 139 of 153

net inlet pressure is near or at the required net inlet pressure by noisy and rough operation of the pump caused by the incomplete filling of the CTIO volume with an accompanying reduction in pump flow rate.

The two other pressure ratings used are the maximum allowable working pressure, which is the maximum gage pressure at the outlet port permitted for safe operation. It is usually determined by the stiffness and strength of the pump body and the type of seals used. The maximum differential pressure is the maximum allowable difference between the outlet pressure and the inlet pressure, measured in the same units, and is determined by the capability of the rotating assembly and its fluid seal contact zones to withstand the pressure difference between the OTO and OTI volumes.

Efficiency and Power Overall efficiency (E_{tot}) for any type of pump is the ratio of the output fluid power P_{out} to the input shaft power P_s. In a development similar to that for centrifugal pumps in Section 2.1.1, E_{tot} can then be broken into three main component efficiencies; namely, volumetric, hydraulic, and mechanical. Volumetric efficiency (E_{vol}) is the ratio of ingested flow rate Q_{net} to the theoretical or displacement flow rate Q_d, where $Q_{net} = Q_d - S$, where S is the internal leakage flow rate (slip) through clearances back to the inlet. Thus the volumetric efficiency is defined in the same way as it is for screw pumps (Section 3.7). Hydraulic efficiency (E_{hyd}) is the ratio of the total fluid power P_{tf} of both the net flow and the slip flow to the actual power P_{fl} delivered to the fluid, where $P_{tf} = P_{out}/E_{vol}$. P_{fl} in turn is the sum of P_{tf} and the additional power required to overcome internal hydrodynamic friction P_{fricl}. Mechanical efficiency (E_{mech}) is the ratio of the P_{fl} to P_s, where the shaft power $P_s = P_{fl}$ plus the mechanical losses P_{mech}, namely the power due to bearing friction, seal drag, internal rubbing, and so on. For incompressible flow, $P_{out} = [\Delta p\ (\text{psi}) \times Q_{net}\ (\text{gpm})/1714]$ hp or $[\Delta p\ (\text{bar}) \times Q_{net}\ (\text{m}^3/\text{h})/36]$ kW, where Δp is the pressure rise of the pump, and Q_{net} is also the flow rate leaving the pump.

As in the case of centrifugal pumps, the above definitions yield the following relationship of the component efficiencies with the overall efficiency:

$$E_{tot} = E_{vol} \times E_{hyd} \times E_{mech} \tag{5}$$

For rotary pumps, efficiency evaluation requires a significantly different perspective in comparison to centrifugal pumps. There are several reasons for that. Centrifugal pumps handle primarily low-viscosity liquids, typically under 100 cSt, while rotary pumps operate at much wider range of viscosities, often above 10,000 cSt and at times reaching even to 1,000,000 cSt. The Hydraulic Institute developed a chart (Figure 11 in Section 2.1.1) to correct centrifugal pump flow, head and power, when viscosities exceed normally very low, water-like values, but such cases are very rare, and thus a need to apply those corrections is rare. Also, centrifugal pumps can reach much higher power levels, as compared to rotary pumps, and thus overall efficiency (E_{tot}) evaluation, as it impacts energy consumption, is more important for centrifugal pumps. In contrast, rotary pumps are typically smaller (lower power), and their ability to deliver a constant (or nearly constant) flow rate, with little dependence on system restriction (relative insensitivity of flow to system pressure), is a consideration in the selection and application of rotary pumps. For these reasons, volumetric efficiency (E_{vol}) has historically played an important role in the world of rotary pumps, while overall (E_{tot}) efficiency is of primary importance in the world of centrifugal pumps.

The above efficiency definitions for rotary pumps are stated as follows in equation form. Volumetric efficiency is given by

$$E_{vol} = Q_{net}/Q_d = (Q_d - S)/Q_d \tag{6}$$

where

$$Q_d = \text{theoretical (ideal) flow, in the absence of slip, } = q_0 \times \text{rpm} \tag{7}$$

and where Q_{net} = net flow rate leaving a rotary pump
S = slip (internal recirculation back to inlet)
q_0 = unit flow rate (constant for a given pump) per revolution

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 140 of 153

Overall efficiency is formally stated as

$$E_{tot} = P_{out}/P_s \tag{8}$$

where, assuming incompressible flow, useful fluid power is given by

$$P_{out} = [Q_{net}\,(\text{gpm}) \times \Delta p\,(\text{psi})/1714]\ \text{hp} = Q_{net}\,(\text{m}^3/\text{h}) \times \Delta p\,(\text{bar})/36\ \text{kW} \tag{9}$$

and total fluid power (useful plus that which drives the slip) is

$$P_{tf} = [Q_d \times \Delta p/1714]\ \text{hp} = [Q_d \times \Delta p/36]\ \text{kW} \tag{10}$$

The actual power P_{fl} delivered to the fluid is P_{tf} augmented by the internal hydrodynamic friction P_{fricl}:

$$P_{fl} = P_{tf} + P_{fricl} \tag{11}$$

and the input shaft power delivered to the pump is the total fluid power plus the mechanical losses:

$$P_s = P_{fl} + P_{mech} \tag{12}$$

Slip S and the viscous drag, P_{visc}, where,

$$P_{visc} = P_{fricl} + P_{mech} \tag{13}$$

are determined (usually by testing) by pump manufacturers. These are specific to each design type and size and are published in pump catalogues as functions of pump speed, differential pressure, and viscosity.

Volumetric efficiency has numerical significance only when comparing similar designs (their comparative/competitive ability to maintain a constant flow rate with minimum slip), operating at the same speed. The reason for this is that, for rotary pumps, the ideal, theoretical, or displacement flow rate Q_d is directly proportional to pump speed, but slip does not depend on pump speed, and only depends on differential pressure, viscosity, and internal clearances. Consider, for example, a rotary pump delivering a required flow rate of 90 gpm (20.4 m³/h) at 1000 rpm, with 10 gpm (2.3 m³/h) internal slip (the ideal flow rate thus being 100 gpm or 22.7 m³/h). The volumetric efficiency of this pump is 90/100 = 90%. If this pump speeds up to 2000 rpm, its ideal flow will double (200 gpm or 45.4 m³/h), but the slip remains the same 10 gpm (2.3 m³/h), with a net flow rate of 200 − 10 = 190 gpm (43.2 m³/h), thereby yielding higher volumetric efficiency 190/200 = 95%, but too great a flow rate as compared to the required amount.

If a competitive pump were presented for evaluation, operating at the same speed (1000 rpm), same unit flow (q_0) but with smaller clearances and a slip of only 5 gpm (1.1 m³/h), then its volumetric efficiency would be (100 − 5)/100 = 95%, which makes it a more efficient design with regard to volumetric capability.

However, if the width of the gears of the 2000-rpm pump were reduced by half, its ideal flow would be reduced to 100 gpm (22.7 m³/h) with no change in slip, because end and radial clearances remain the same. Volumetric efficiency would then become (100 − 10)/100 = 90%. This altered 2000-rpm pump now has lower volumetric efficiency (a greater proportion of slip), but it is also less expensive due to the halved gear length. These facts would need to be evaluated together by the end user.

NUMERICAL EXAMPLE

The trends of the above example and the distinctions discussed between rotary pumps (gear, vane, or lobe) and centrifugal pumps become clearer in the following series of performance computations pertaining to a single rotary pump operating at two greatly differing viscosities (Cases 1 and 2) and at two greatly differing speeds at each of these two viscosities (subcases of Cases 1 and 2). Most rotary pump manufacturers publish pump performance curves showing four main relationships that enable these computations to proceed. Specifically, Figures 15 through 18 illustrate these relationships for the pump of this example.

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 141 of 153

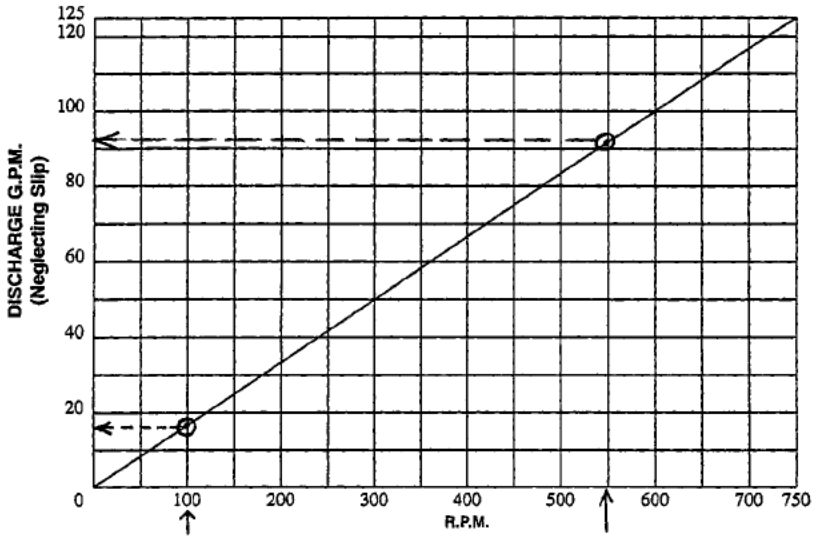


FIGURE 15 Ideal (theoretical) flow rate versus speed. This is typically a straight line, with a slope equal to pump unit flow q_0. ($m^3/h = 0.227 \times gpm$) (Ranger, Inc.)

Figures 15 and 17 are plotted for convenience, because they can be alternatively calculated from Eqs. 7 and 10; but Figures 16 and 18 can be obtained only from the manufacturer's curves (internally tested values).

Case 1: Viscosity = 30 SSU (~ 1 cSt)

Subcase 1.1: Speed = 100 rpm. Displacement flow rate: $Q_d = q_0 \times 100 = 0.17 \times 100 = 17.0$ gpm (3.86 m^3/h) (or read from Figure 15).

Results for $\Delta p = 25$ psi (1.72 bar)

Slip flow rate, $S = 4.2$ gpm (0.95 m^3/h) from Figure 16

Net flow rate, $Q_{net} = 17.0 - 4.2 = 12.8$ gpm (2.91 m^3/h)

Total fluid power $P_{tf} = \Delta p \times Q_d/1714 = 25 \times 17.0/1714 = 0.25$ hp (= 0.185 kW) (or read from Figure 17)

Useful output power, $P_{out} = \Delta p \times Q_{net}/1714 = 25 \times 12.8/1714 = 0.187$ hp (0.139 kW)

Viscous power, $P_{visc} \approx 0.1$ hp (0.075 kW) from Figure 18

Shaft power, $P_s = P_{tf} + P_{visc} = 0.25 + 0.1 = 0.35$ hp (0.26 kW)

Volumetric efficiency, $E_{vol} = Q_{net}/Q_d = 12.8/17.0 = 75.2\%$

Overall efficiency $E_{tot} = P_{out}/P_s = 0.187/0.35 = 53.4\%$

Results for $\Delta p = 50$ psi (3.45 bar)

$S = 8.4$ gpm (1.91 m^3/h)

$Q_{net} = 17.0 - 8.4 = 8.6$ gpm (1.95 m^3/h)

$P_{tf} = \Delta p \times Q_d/1714 = 50 \times 17.0/1714 = 0.496$ hp (0.370 kW)

$P_{out} = \Delta p \times Q_{net}/1714 = 50 \times 8.6/1714 = 0.251$ hp (0.187 kW)

$P_{visc} \approx 0.1$ hp (0.075 kW)

$P_s = P_{tf} + P_{visc} = 0.496 + 0.1 = 0.596$ hp (0.445 kW)

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 142 of 153

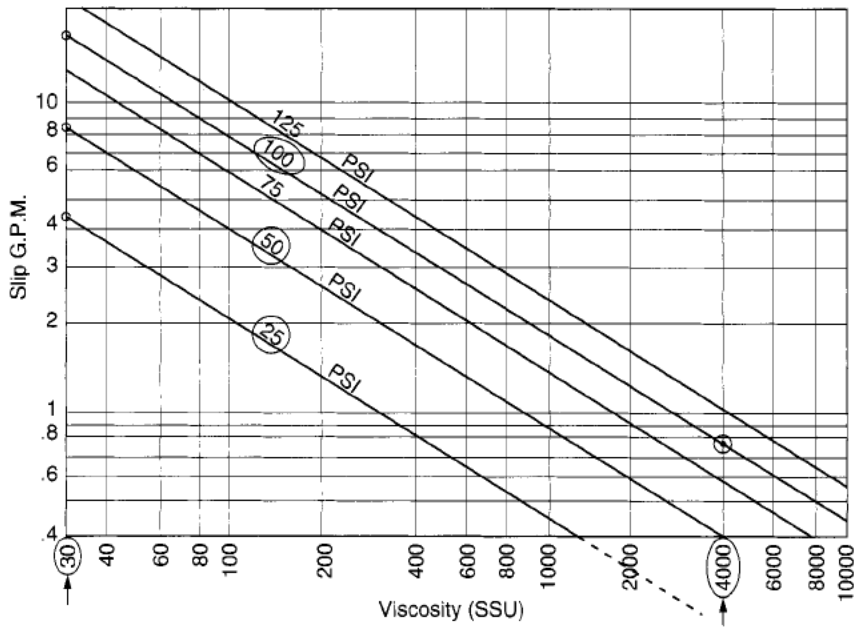


FIGURE 16 Relationship between internal slip and viscosity at several values of differential pressure. (m^3/h = 0.227 × gpm and bar = psi/14.5) (Ranger, Inc.)

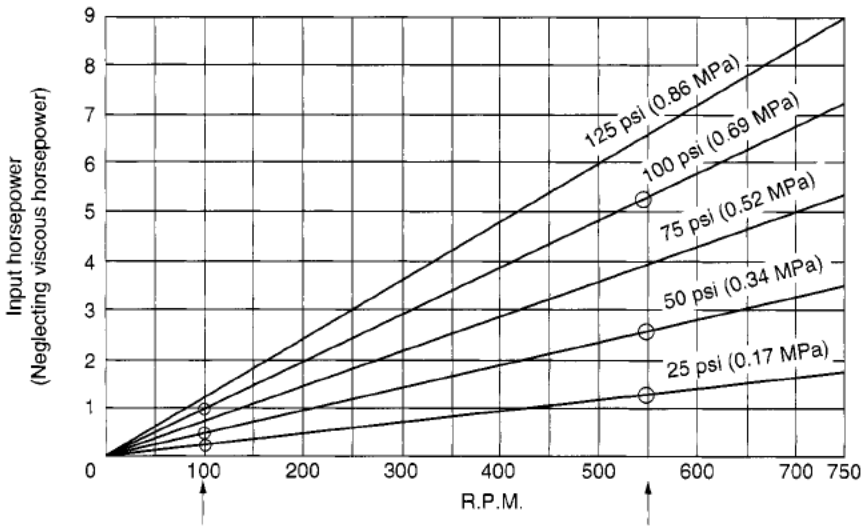


FIGURE 17 Total fluid power (minus power due to internal hydrodynamic friction or "viscous power") versus speed at several values of differential pressure (kW = hp/1.34, and bar = psi/14.5) (Ranger, Inc.)

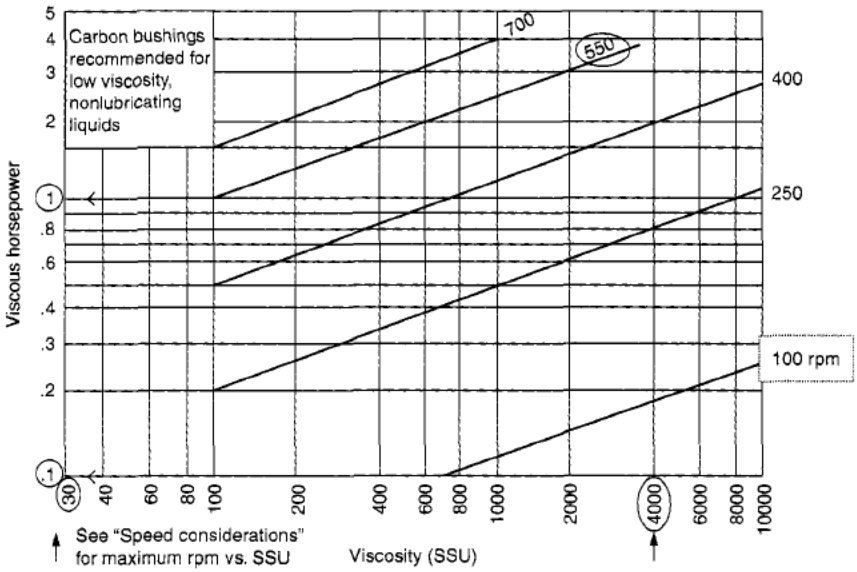


FIGURE 18 Viscous power loss versus viscosity, at several values of speed (kW = hp/1.34) (Ranger, Inc.)

$E_{vol} = Q_{net}/Q_d = 8.6/17.0 = 50.6\%$
$E_{tot} = P_{out}/P_s = 0.251/0.6 = 41.8\%$

Results for Δp = 100 psi (6.89 bar)
S = 16.8 gpm (3.82 m³/h)
$Q_{net} = 17.0 - 16.8 = 0.2$ gpm (0.045 m³/h)
$P_{tf} = \Delta p \times Q_d/1714 = 100 \times 17.0/1714 = 0.992$ hp (0.740 kW)
$P_{out} = \Delta p \times Q_{net}/1714 = 100 \times 0.2/1714 = 0.012$ hp (0.009 kW)
$P_{visc} = 0.1$ hp (0.075 kW)
$P_s = P_{tf} + P_{frict} = 0.992 + 0.1 = 1.092$ hp (0.815 kW)
$E_{vol} = Q_{net}/Q_d = 0.2/17.0 = 1.2\%$
$E_{tot} = P_{out}/P_s = 0.012/1.092 = 1.1\%$

Observation: The slip S appears to vary linearly with differential pressure. This is not always the case for all gear pumps and is strictly coincidental in this particular example.

Subcase 1.2: Speed = 550 rpm. Displacement flow rate: $Q_d = q_0 \times 550 = 0.17 \times 550 = 93.5$ gpm (21.24 m³/h) (or read from Figure 15).

Results for Δp = 25 psi (1.72 bar)
S = 4.2 gpm (0.95 m³/h)
$Q_{net} = 93.5 - 4.2 = 89.3$ gpm (20.28 m³/h)
$P_{tf} = \Delta p \times Q_d/1714 = 25 \times 93.5/1714 = 1.364$ hp (1.017 kW)
$P_{out} = \Delta p \times Q_{net}/1714 = 25 \times 89.3/1714 = 1.303$ hp (0.972 kW)

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 144 of 153

P_{visc} = 1.0 hp (0.746 kW)
$P_s = P_{tf} + P_{frict}$ = 1.364 + 1.0 = 2.364 hp (1.764 kW)
$E_{vol} = Q_{net}/Q_d$ = 89.3/93.5 = 95.5%
$E_{tot} = P_{out}/P_s$ = 1.303/2.364 = 55.1%

Results for Δp = 50 psi (3.45 bar)
S = 8.4 gpm (1.91 m³/h)
Q_{net} = 93.5 − 8.4 = 85.1 gpm (19.33 m³/h)
$P_{tf} = \Delta p \times Q_d/1714$ = 50 × 93.5/1714 = 2.728 hp (2.035 kW)
$P_{out} = \Delta p \times Q_{net}/1714$ = 50 × 85.1/1714 = 2.482 hp (1.852 kW)
P_{visc} = 1.0 hp (0.746 kW)
$P_s = P_{tf} + P_{frict}$ = 2.728 + 1.0 = 3.728 hp (2.781 kW)
$E_{vol} = Q_{net}/Q_d$ = 85.1/93.5 = 91.0%
$E_{tot} = P_{out}/P_s$ = 2.482/3.728 = 66.6%

Results for Δp = 100 psi (6.89 bar)
S = 16.8 gpm (3.82 m³/h)
Q_{net} = 93.5 − 16.8 = 76.7 gpm (17.42 m³/h)
$P_{tf} = \Delta p \times Q_d/1714$ = 100 × 93.5/1714 = 5.455 hp (4.069 kW)
$P_{out} = \Delta p \times Q_{net}/1714$ = 100 × 76.7/1714 = 4.475 hp (3.338 kW)
P_{visc} = 1.0 hp (0.746 kW)
$P_s = P_{tf} + P_{frict}$ = 5.455 + 1.0 = 6.455 hp (4.815 kW)
$E_{vol} = Q_{net}/Q_d$ = 76.7/93.5 = 82.0%
$E_{tot} = P_{out}/P_s$ = 4.475/6.455 = 69.3%

Case 2: Viscosity = 4000 SSU (~ 900 cSt)

Subcase 2.1: Speed 100 rpm. Displacement flow rate: $Q_d = q_0 \times 100$ = 0.17 × 100 = 17.0 gpm (3.86 m³/h) (or read from Figure 15).

Results for Δp = 25 psi (1.72 bar)

Slip flow rate, S = 0.1 gpm (0.023 m³/h) from Figure 16

Net flow rate, Q_{net} = 17.0 − 0.1 = 16.9 gpm (3.84 m³/h)

Total fluid power $P_{tf} = \Delta p \times Q_d/1714$ = 25 × 17.0/1714 = 0.25 hp (= 0.185 kW) (or read from Figure 17)

Useful output power, $P_{out} = \Delta p \times Q_{net}/1714$ = 25 × 16.9/1714 = 0.247 hp (0.184 kW)

Viscous power, P_{visc} = 0.18 hp (0.134 kW) from Figure 18

Shaft power, $P_s = P_{tf} + P_{frict}$ = 0.25 + 0.18 = 0.43 hp (0.32 kW)

Volumetric efficiency, $E_{vol} = Q_{net}/Q_d$ = 16.9/17.0 = 99.4%

Overall efficiency $E_{tot} = P_{out}/P_s$ = 0.247/0.43 = 57.4%

Results for Δp = 50 psi (3.45 bar)
S = 0.4 gpm (0.091 m³/h)
Q_{net} = 17.0 − 0.4 = 16.6 gpm (3.77 m³/h)
$P_{tf} = \Delta p \times Q_d/1714$ = 50 × 17.0/1714 = 0.496 hp (0.370 kW)
$P_{out} = \Delta p \times Q_{net}/1714$ = 50 × 16.6/1714 = 0.484 hp (0.361 kW)
P_{visc} = 0.18 hp (0.134 kW)

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 145 of 153

$P_s = P_{tf} + P_{frict} = 0.496 + 0.18 = 0.676$ hp (0.504 kW)
$E_{vol} = Q_{net}/Q_d = 16.6/17.0 = 97.6\%$
$E_{tot} = P_{out}/P_s = 0.484/0.676 = 71.6\%$

Results for Δp = 100 psi (6.89 bar)
$S = 0.75$ gpm (0.170 m³/h)
$Q_{net} = 17.0 - 0.75 = 16.25$ gpm (3.69 m³/h)
$P_{tf} = \Delta p \times Q_d/1714 = 100 \times 17.0/1714 = 0.992$ hp (0.740 kW)
$P_{out} = \Delta p \times Q_{net}/1714 = 100 \times 16.2/1714 = 0.945$ hp (0.705 kW)
$P_{visc} = 0.18$ hp (0.134 kW)
$P_s = P_{tf} + P_{frict} = 0.992 + 0.18 = 1.172$ hp (0.874 kW)
$E_{vol} = Q_{net}/Q_d = 16.25/17.0 = 95.6\%$
$E_{tot} = P_{out}/P_s = 0.945/1.172 = 80.6\%$

Subcase 2.2: Speed = 550 rpm. Displacement flow rate: $Q_d = q_0 \times 550 = 0.17 \times 550 = 93.5$ gpm (21.24 m³/h) (or read from Figure 15).

Results for $\Delta p = 25$ psi (1.72 bar)
$S \approx 0.1$ gpm (0.023 m³/h)
$Q_{net} = 93.5 - 0.1 = 93.4$ gpm (21.21 m³/h)
$P_{tf} = \Delta p \times Q_d/1714 = 25 \times 93.5/1714 = 1.364$ hp (1.017 kW)
$P_{out} = \Delta p \times Q_{net}/1714 = 25 \times 93.4/1714 = 1.362$ hp (1.016 kW)
$P_{visc} = 3.9$ hp (2.909 kW)
$P_s = P_{tf} + P_{frict} = 1.364 + 3.9 = 5.264$ hp (3.927 kW)
$E_{vol} = Q_{net}/Q_d = 93.4/93.5 = 99.9\%$
$E_{tot} = P_{out}/P_s = 1.362/5.264 = 25.9\%$

Results for Δp = 50 psi (3.45 bar)
$S = 0.4$ gpm (0.091 m³/h)
$Q_{net} = 93.5 - 0.4 = 93.1$ gpm (21.14 m³/h)
$P_{tf} = \Delta p \times Q_d/1714 = 50 \times 93.5/1714 = 2.728$ hp (2.035 kW)
$P_{out} = \Delta p \times Q_{net}/1714 = 50 \times 93.1/1714 = 2.716$ hp (2.026 kW)
$P_{visc} = 3.9$ hp (2.909 kW)
$P_s = P_{tf} + P_{frict} = 2.728 + 3.9 = 6.628$ hp (4.944 kW)
$E_{vol} = Q_{net}/Q_d = 93.1/93.5 = 99.6\%$
$E_{tot} = P_{out}/P_s = 2.716/6.628 = 41.0\%$

Results for Δp = 100 psi (6.89 bar)
$S = 0.75$ gpm (0.170 m³/h)
$Q_{net} = 93.5 - 0.75 = 92.8$ gpm (21.08 m³/h)
$P_{tf} = \Delta p \times Q_d/1714 = 100 \times 93.5/1714 = 5.455$ hp (4.069 kW)
$P_{out} = \Delta p \times Q_{net}/1714 = 100 \times 92.8/1714 = 5.414$ hp (4.039 kW)
$P_{visc} = 3.9$ hp (2.909 kW)
$P_s = P_{tf} + P_{frict} = 5.455 + 3.9 = 9.355$ hp (6.979 kW)
$E_{vol} = Q_{net}/Q_d = 92.8/93.5 = 99.3\%$
$E_{tot} = P_{out}/P_s = 5.414/9.355 = 57.9\%$

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 146 of 153

SUMMARY

1. Results

Speed		100 rpm			550 rpm	
Δp	25 psi	50 psi	100 psi	25 psi	50 psi	100 psi
			Viscosity = 30 SSU			
Eff_{vol}	75.2%	50.6%	1.2%	95.5%	91.0%	82.0%
Eff_{tot}	53.4%	41.8%	1.1%	55.1%	66.6%	69.3%
			Viscosity = 4000 SSU			
Eff_{vol}	99.4%	97.6%	95.6%	99.9%	99.6%	99.3%
Eff_{tot}	57.4%	71.6%	80.6%	25.9%	41.0%	55.9%

2. Observations
   - *Volumetric* efficiency is very high at high viscosity due to low slip, regardless of pressure and speed.
   - At low viscosity and high pressure *total* efficiency tends to be very low.
   - Higher speed helps *total* efficiency (due to low drag) at low viscosity but reduces it at higher viscosity due to high viscous drag.

Pump Performance Figures 19, 20, and 21 show the change in displacement capacity Q_d, flow rate or capacity Q_c, and slip Q_s as the differential pressure across the pump P_d (= Δp), the viscosity v, and the pump speed N are varied. Several assumptions are made in these graphs. It is assumed that inlet conditions are satisfactory and that there is no inlet effect on the pump capacity over the charted range. It is assumed that the fluid is Newtonian and the liquid is incompressible. In Figures 19 and 20, it is assumed that viscosity is constant and relatively low, and that the speed is within the normal range of the pump. In Figure 20, it is assumed that both pressure and speed are within the normal ratings of the pump.

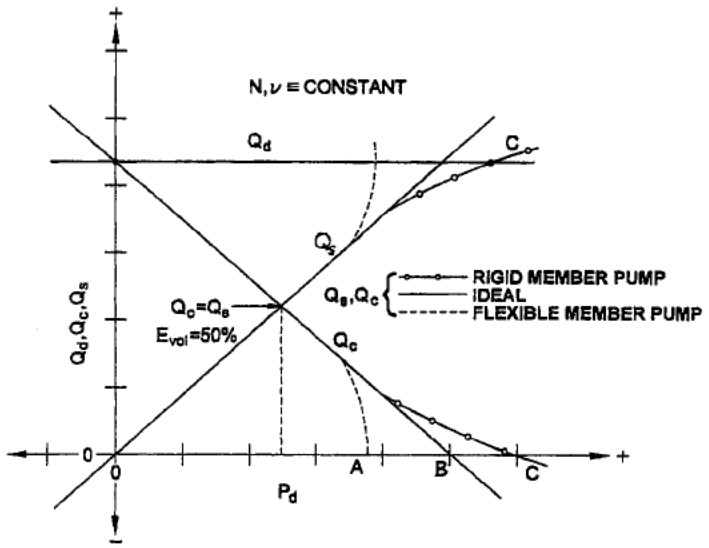


FIGURE 19 Varation of Q_d, Q_c, and Q_s with P_d, N and v constant

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 147 of 153

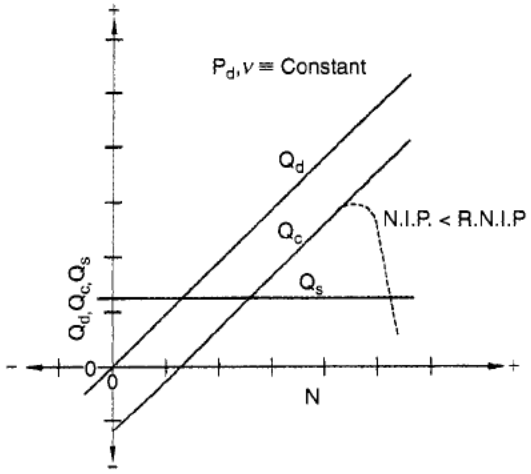


FIGURE 20 Varations of Q_d, Q_c, and Q_s with N, P_d, and v constant

The graph in Figure 19 is plotted with the flow rate, slip, and displacement flow rate on a linear scale as the ordinate (Y-axis) and pressure on a linear scale as the abscissa (X-axis). A further assumption is that the clearances and viscosity are such that the slip increases proportionately with pressure. The solid lines are the ideal characteristics when secondary effects are neglected. It may be noted that at zero pressure, slip Q_s is zero and Q_c equals Q_d. As the pressure increases, Q_s increases until it equals Q_d at pressure B. If the

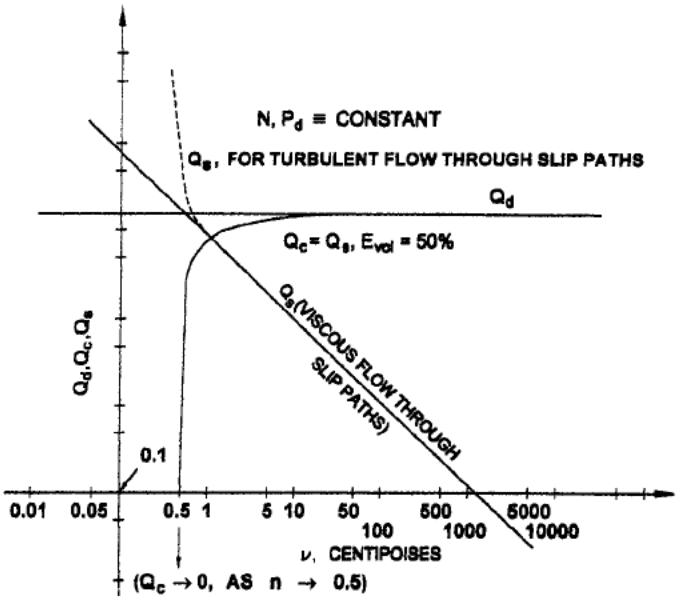


FIGURE 21 Varations of Q_d, Q_c, and Q_s with v, N and P_d constant

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 148 of 153

pressure imposed on the pump were to increase past this point, Q_s would exceed Q_d and the flow through the pump would be from outlet to inlet, causing a negative Q_c.

Although rotary pumps do not operate in this end of the pressure range, the condition of pressure B can be reached when a valve is closed, blocking the outlet of the pump. Pressure B, then, represents dead-ended pressure developed by a rotary pump when its outlet line is blocked. Should Q_d be numerically equal to 100 in whichever units are used, then the plot of Q_c is numerically equal at each point to the volumetric efficiency E_{vol} of the pump and Q_c becomes zero at pressure B.

Pressure B data is usually not available from pump manufacturers since these values are usually far beyond the normal safe pressure rating of the pump. However, they can be estimated by extrapolating the data given. Cautions should be observed, however. A flexible member can quickly reach a pressure at which flexible member deflection becomes excessive, and the pressure at which Q_c equals zero is very quickly reached. This is illustrated in Figure 19 by the dashed line breaking away from the solid Q line and intersecting the abscissa at pressure point A. This characteristic of flexible member pumps provides a self-limited maximum pressure should the pump be dead-ended. In rigid rotor pumps, as the pressure increases beyond the normal operating range, deflections of the shafts and rotors bring the rotors into heavy bearing contact with the body chamber walls, reducing the dimensions of the slip clearance path. The dashed line leaving the Q_c and Q_s curves and intersecting the abscissa and Q_d line at pressure point C illustrates this. Consequently, the zero flow pressure, which is ideally at point B, may be considerably different from this extrapolated value, and it should be measured if it is important to the application.

Under the same assumptions given for Figure 19, Figure 20 shows the relative independence of the slip with speed when differential pressure is constant. Here the chart of Q intersects the abscissa at the point where the speed is low enough for the displacement flow rate to be equal to the slip at the pressure of operation. The speed at which Q_d equals $2 \times Q_s$ is the speed at which Q_s equals Q_c and the volumetric efficiency E_{vol} becomes 50%.

As the speed increases, the pumping action of the shear stress in the clearances tends to reduce the slip below the ideal line. However, for most rotary pumps, the detrimental effects of increasing the speed above the normal operating range is usually caused by inlet losses in the pump. This effect is cavitation and is illustrated by the dashed line, for which the net inlet pressure (N. I. P.) or NPSH is less than that required.

Figure 21 shows the effect of viscosity on the slip and flow rate in a rotary pump. The graphs are on a log-log scale. In this chart, it is assumed that the pressure, speed, and viscosity combine to keep the flow through clearances of the pump in the laminar or "viscous" flow region.

As viscosity increases, slip Q_s becomes very small and the flow rate of the pump Q_c approaches the displacement flow rate Q_d. As the viscosity decreases, the slip very rapidly approaches the displacement flow rate and the flow rate of the pump drops rapidly to zero or to a negative quantity. For any given pump with speed N and differential pressure P_d, there is a viscosity below which the slip flow, through clearances of the pump, will change to a turbulent flow. It is unlikely that this change will occur simultaneously through all slip paths. However, once it begins to occur, the slip will increase much more rapidly with further reductions in viscosity because of the turbulent flow relationship of the slip, pressure, and viscosity expressed in Eq. 14:

$$Q_s = \frac{KP_d^{1/2}}{v^{1/x}} \tag{14}$$

where x usually is in the range of 4 to 10, and K is a constant.

Volumetric efficiency E_{vol} drops very rapidly with the viscosity if the viscosity is lower than that required for 50% volumetric efficiency (represented by the crossover point of the slip and capacity curves). This crossover point occurs for most rotary pumps operating at rated differential pressure P_d for viscosities between 0.1 and 10 centipoise. For the majority of commercially available models, this point usually falls in the viscosity range of 0.3 to 3.0 centipoise at the maximum rated differential pressure.

The effect of inlet pressure on the capacity can be seen clearly if all secondary effects are eliminated. This assumes the pump is operating within its normal pressure limits and speed

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 149 of 153

range and that viscosity is high enough to reduce the slip to a negligible value. A graph of the capacity as a function of inlet pressure is shown in Figure 22 with these assumptions.

There is no change in capacity or flow rate as inlet pressure is lowered until the pressure reaches pressure *A* on the graph. If the inlet pressure were lowered further, the flow rate would drop as shown. The cause of this drop is complex in detail but simple in concept. The liquid flow from the inlet port through the inlet chamber of the pump causes a pressure drop, which causes a minimum pressure point somewhere in the inlet chamber. When the pressure in the liquid at this minimum pressure point approaches the vapor pressure of the liquid, vapor begins to form there. When the amount and time persistence of this vapor cause the vapor to be swept into the CTIO volume of the pump, the amount of liquid in this volume is reduced, leaving a deficiency of liquid volume. For example, if half the fluid volume swept from the inlet chamber is vapor, then only half the normal-capacity liquid volume is available at the outlet chamber, and the flow rate is reduced accordingly.

An increase in speed would mean an increase in flow rate, all other things being equal. This would increase the pressure drop between the inlet port and the inlet chamber and correspondingly increase the absolute inlet pressure (which is measured at the inlet port) at which the flow rate would begin to drop (pressure *A*). Correspondingly, if the speed and flow rate were constant and the viscosity increased, the pressure drop between the inlet port and the minimum pressure point in the inlet chamber would increase with viscosity. This also would cause pressure *A* to move to higher inlet absolute pressures. Operation with absolute inlet pressures below pressure *A* for any given speed and viscosity is usually unsatisfactory both because of the drop in capacity (and hence in volumetric efficiency) and because of the noisy and rough operation caused by the formation and collapse of the vapor.

For lower viscosity liquids where the collapse of the vapor bubbles may be quite rapid, cavitation may cause a significant amount of damage to the body or rotor surfaces. It is important to understand that cavitation damage may occur even though the absolute inlet pressure is above pressure *A*. Locations may be found in the inlet chamber, particularly where the flow direction changes rapidly, as around sharp corners, where vapor formation in the form of very fine bubbles, and hence cavitation, may occur. However, these vapor cavities may be swept

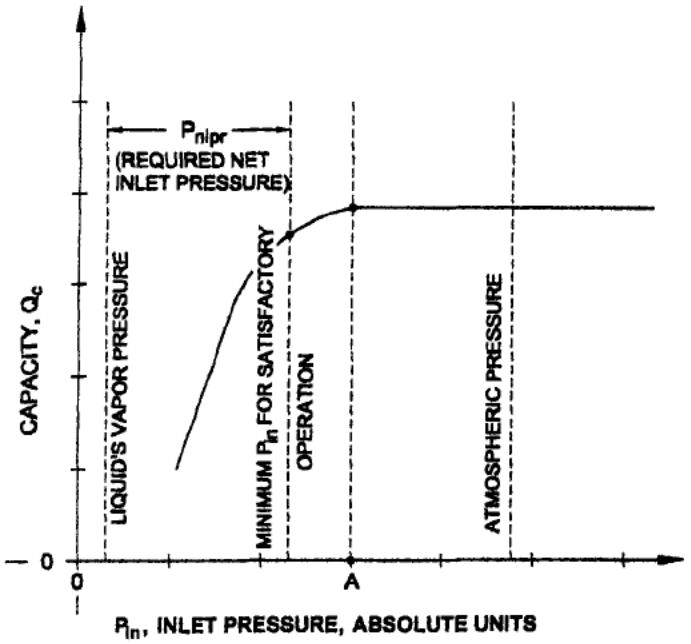


FIGURE 22 Varations of Q_c and P_{in} with other operating conditions constant

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 150 of 153

into higher pressure regions of the inlet chamber and collapse near body or rotor surfaces to cause cavitation damage, although the capacity of the pump is not affected. For any given viscosity then, there is an upper limit to the speed at which the pump can be operated.

The inlet pressure may be allowed to drop slightly below pressure *A* without a significant deterioration in pump performance. However, there is a point at which the pressure becomes too low for satisfactory pump operation. This pressure determines the required net inlet pressure P_{nipr} or NPSHR for the particular pump and the particular set of operation conditions. For any given set of operating conditions, satisfying the required net inlet pressure or NPSHR is a main limitation on the pump operating speed.

The other main limit on the pump operating speed is the pump outlet pressure. In every application, some fluid frictional losses occur in the outlet system of the pump. Even if the pump outlet is opened to the atmosphere, a pressure drop exists between some maximum pressure point in the pump outlet chamber and the pump outlet itself. However, by far, the most common situation is one in which a significant fluid friction pressure is developed in the system external to the outlet of the pump. This fluid pressure usually is a function of pump capacity and therefore of pump speed. If the inlet conditions are maintained to keep the inlet pressure above pressure *A* as the pump speed increases, a speed will be reached at which the pump outlet pressure equals the outlet pressure rating of the pump. Operating at speeds higher than this will cause the pump outlet pressure to exceed the rated pressure and this could result in damage to the pump.

In most applications, one of the two limits described determines the maximum permitted speed of pump operation. However, if neither of these conditions limits the speed and the speed is continuously increased, a speed will be reached at which the peripheral velocity of the rotors will exceed the cavitation velocity of the liquid. A further increase in speed beyond this point would be limited by the cavitation occurring at the rotor outer radial surfaces.

SPECIFICATIONS, INDUSTRIES, AND APPLICATIONS

In addition to the many application-oriented classifications for vane, gear, or lobe pumps, many technical classifications are also issued by various industrial, governmental, or even military organizations. It is also important for both the manufacturer and the user to know of any voluntary or regulatory specifications governing the construction and materials of the pumps destined for use in a specific industry and application.

Many industry-based specifications, such as the International Association of Milk, Food, and Environmental Sanitarians, the United States Public Health Service, and the Dairy Industry Committee, have jointly issued specifications called "The 3-A Sanitary Standards for Pumps for Milk and Milk Products." These standards govern the materials of construction of the pump, the surface finish and shape details of the contact surfaces, the finish and shape of external surfaces, the method of mounting, and restrictions on gaskets, seals, and other pump auxiliary features. Pumps constructed to meet 3-A Sanitary Standards are called *sanitary* pumps and carry a 3-A seal of approval mounted on the pump body. Similar specifications have been developed by the International Association of Milk, Food, and Environmental Sanitarians, the United States Public Health Service, the United States Department of Agriculture, the American Poultry Industries, and the Dairy and Food Industries Supply Association to cover design features used in the handling of cracked eggs.

Standards for pumps used in various processes in the petroleum industry are concerned with the materials and design features that are intended to prevent catastrophic failures when explosive, flammable, or toxic fluids are being handled. Others generated by governmental agencies and professional societies, such as the American Society of Mechanical Engineers and the American Petroleum Institute Standard 676, establish manufacturing procedures and design limitations on pumps for refineries, nuclear power stations, and so on. In these cases, the terms *sanitary*, *aseptic*, *explosion proof*, *API pump* or *N stamp* are not merely generic terms and cannot be casually applied. They must instead be carefully applied and used only when the manufacturer warrants that the pumps actually meet these specifications. In addition, as an unavoidable consequence, the cost of manufacturing pumps to meet any of these standards makes them more expensive than pumps built for general service applications.

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 151 of 153

TABLE 2 Markets and applications

Animal Renderings Industry

General transferring to burners of:
- animal feed
- beef by-products
- bone meal
- molten fats
- poultry by-products

Automobile Industry

Distribution and assembly line supply of:
- grease
- lubricating oil
- motor fuel
- paint, lacquers and thinners
- test stand systems

Chemical Industry

Any process or distribution system handling of:
- acids
- adhesives
- bleaches
- caustics
- cellulose acetate
- glycerin
- nitrates

Cosmetics and Soap Industry

Nonemulsifying supply to packaging machinery of:
- detergent
- lanolin
- liquefied soap
- shampoo

Food Products Industry

General process transferring of:
- coconut oil
- corn oil
- fish oils
- glucose
- glycerin
- lard
- milk and milk by products
- molasses
- palm oil
- sucrose solutions
- vegetable shortening

General Industry and OEMs

Various fluid handling requirements for:
- burners and heater sets
- centrifuge separator systems
- cooking fat filtration units
- filtering and metering units
- fluid reclamation systems
- hydraulic auto lifts and elevators
- lube oil supply units for rotating equipment
- machinery coolant and lubricant circulation
- paint spray booths
- roofing tar systems
- textile dyeing and printing machinery
- wood preservation machinery

Marine Industry

Unloading, tank stripping and engine room supply of:
- asphalt
- heating oil
- heavy fuel oil
- marine diesel oil
- lubricating oil
- molasses

Paint and Lacquer Indus

Various process handling applications for:
- dyes and pigments
- lacquer
- latex
- paint
- titanium dioxide slurry
- thinners and solvents

Petroleum Industry

Blending, packaging, and unloading of:
- asphalt
- bitumen
- creosote
- grease
- fuel oil
- heavy fuel oil
- lubricating oils

Pulp and Paper Industry

Various general mill and coating applications for:
- adhesives
- black liquor
- de-inking solvents
- fuel oil
- printing ink
- sulfate soap
- tall oil
- titanium dioxide slurry
- turpentine
- viscose
- waste pond recycling
- white liquor

Plastics and Petrochemical Industry

Process transfer of:
- cellophane
- epoxy hardener
- film dope
- insulation coatings
- isocyanate
- polyol
- polyester
- rayon

Public Utilities and Power Stations

General transfer and supply of:
- central lubrication systems
- fuel oil supply to burners
- oil pressure for damper controls
- transfer of gashouse tars, creosote, etc.

Steel Industry

Circulation and supply of:
- coolants
- quench oils
- lube oil to rolling mills
- hydraulic oil to hydraulic lifters

Rotary pumps are used for both metering and transfer applications; although, small diaphragm metering pumps are now more common. Metering pumps tend to be smaller and operate with varying flow rates. Transfer pumps typically have a fixed speed, although variable speed units are becoming more common as the cost for this equipment falls.

In general, the three most basic applications for these pumps are as follows:

1. **Liquid handling** In this class, well over 2,000 fluids are pumped and performance is judged on the pump's capability to handle the specific liquids, and the hydraulic power generated by the pump is a secondary consideration.
2. **Hydraulic fluid power** In this class, the hydraulic power generated by the pump is used to actuate valves, pistons, cylinders, rotary actuators, and similar devices. Pumps in this class are usually special vane, gear, piston, or screw types and are designed for pressures up to 9,000 lb/in^2 (620 bar) on selected hydraulic power fluids.

GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 152 of 153

3. **Commercial and retail** In this class are the pumps designed for a single specific use, application, or even fluid. Pumps in this miscellaneous class include everything from miniature pumps for home aquariums to well water pumps for agricultural use to fuel injection pumps for automobiles.

Among these three application areas, the most common is liquid handling tasks, and they make up the largest percentage of uses for rotary vane, gear, and lobe pumps.

Markets and Applications The markets and applications served by vane, gear, and lobe pumps are among the broadest of all rotary pumps. Although it would not be possible to list every application served by these pumps, a broad sampling of the more common ones is shown in Table 2.

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FURTHER READING

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GD ENERGY PRODUCTS LLC
Petitioner
EX1010
Page 153 of 153