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MARKS' STANDARD HANDBOOK FOR MECHANICAL ENGINEERS

11th EDITION

EUGENE A. AVALLONE
THEODORE BAUMEISTER III
ALI M. SADEGH

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Marks'

Standard Handbook for Mechanical Engineers

Revised by a staff of specialists

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One concludes that in **harmonic response,** time plays a secondary role to the frequency. In fact, the only significant information is extracted from the magnitude and phase angle plots of Fig. 3.4.4. They are referred to as **frequency-response plots.**

Since time plays no particular role, the harmonic response is called **steady-state response.** In general, for linear systems with constant parameters, such as the mass-damper-spring system under consideration, the response to the initial excitations is added to the response to the excitation forces. The response to initial excitations, however, represents **transient response.** This is due to the fact that every system possesses some amount of damping, so that the response to initial excitations disappears with time. In contrast, steady-state response persists with time. Hence, in the case of harmonic excitations, it is meaningless to add the response to initial excitations to the harmonic response.

Vibration Isolation A problem of great interest is the magnitude of the force transmitted to the base by a system of the type shown in Fig. 3.4.2 subjected to harmonic excitation. This force is a combination of the spring force kx and the dashpot force $c\dot{x}$. Recalling Eq. (3.4.23), write

$$\begin{aligned} kx &= kA|G|\cos(\omega t - \phi) \\ c\dot{x} &= -c\omega A|G|\sin(\omega t - \phi) \\ &= c\omega A|G|\cos\left(\omega t - \phi + \frac{\pi}{2}\right) \end{aligned} \tag{3.4.30}$$

so that the dashpot force is 90° out of phase with the spring force. Hence, the magnitude of the force is

$$\begin{aligned} F_{tr} &= \sqrt{(kA|G|)^2 + (c\omega A|G|)^2} = kA|G|\sqrt{1 + (c\omega/k)^2} \\ &= kA|G|\sqrt{1 + (2\zeta\omega/\omega_n)^2} \end{aligned} \tag{3.4.31}$$

Let the magnitude of the harmonic excitation be $F_0 = kA$; the measure of the force transmitted to the base is then

$$\begin{aligned} T = \frac{F_{tr}}{F_0} &= |G|\sqrt{1 + (2\zeta\omega/\omega_n)^2} \\ &= \sqrt{\frac{1 + (2\zeta\omega/\omega n)^2}{[1 - (\omega/\omega_n)^2]^2 + (2\zeta\omega/\omega_n)^2}} \end{aligned} \tag{3.4.32}$$

which represents a nondimensional ratio called **transmissibility.** Figure 3.4.5 plots F_{tr}/F_0 versus ω/ω_n for various values of ζ.

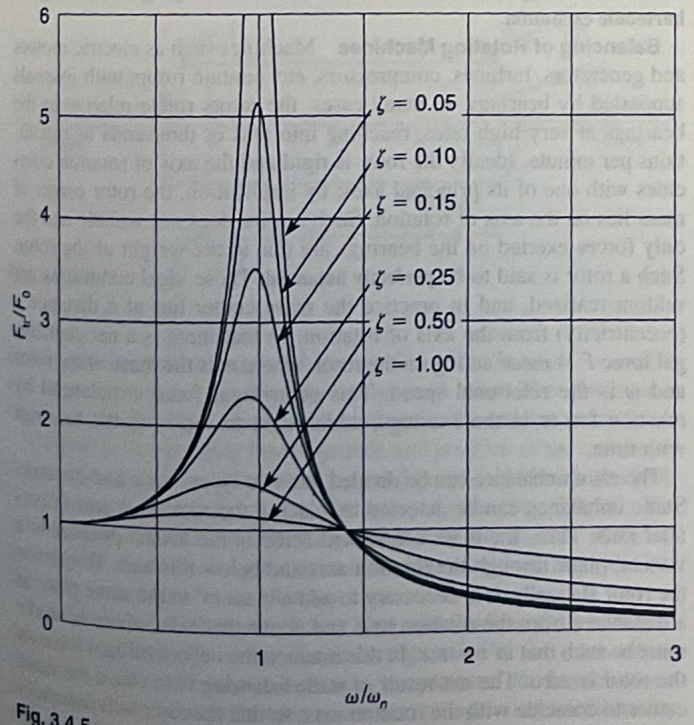


Fig. 3.4.5

The transmissibility is less than 1 for $\omega/\omega_n > \sqrt{2}$, and decreases as ω/ω_n increases. Hence, for **an isolator to perform well,** its natural frequency must be much smaller than the excitation frequency. However, for very low natural frequencies, difficulties can be encountered in isolator design. Indeed, the natural frequency is related to the static deflection δ_{st} by $\omega_n = \sqrt{k/m} = \sqrt{g/\delta_{st}}$, where g is the gravitational constant. For the natural frequency to be sufficiently small, the static deflection may have to be impractically large. The relation between the excitation frequency f measured in rotations per minute and the static deflection δ_{st} measured in inches is

$$f = 187.7\sqrt{\frac{2 - R}{\delta_{st}(1 - R)}} \quad \text{rpm} \tag{3.4.33}$$

where $R = 1 - T$ **represents the percent reduction** in vibration. Figure 3.4.6 shows a logarithmic plot of f versus δ_{st} with R as a parameter.

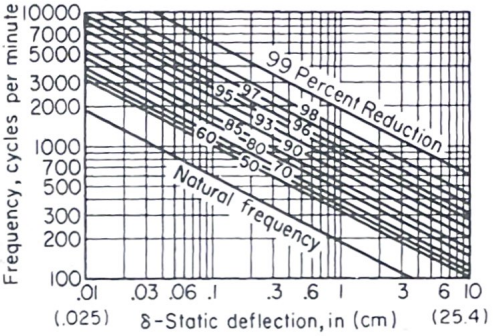


Fig. 3.4.6

Figure 3.4.7 depicts two types of isolators. In Fig. 3.4.7*a*, isolation is accomplished by means of springs and in Fig. 3.4.7*b* by rubber rings supporting the bearings. Isolators of all shapes and sizes are available commercially.

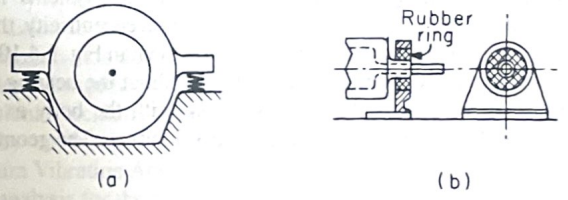


Fig. 3.4.7

Rotating Unbalanced Masses Many appliances, machines, etc., involve components spinning relative to a main body. A typical example is the clothes dryer. Under certain circumstances, the mass of the spinning component is not symmetric relative to the center of rotation, as when the clothes are not spread uniformly in the spinning drum, giving rise to harmonic excitation. The behavior of such systems can be simulated adequately by the single-degree-of-freedom model shown in Fig. 3.4.8, which consists of a main mass $M - m$, supported by two springs of combined stiffness k and a dashpot with coefficient of viscous damping c, and two eccentric masses $m/2$ rotating in opposite sense with the constant angular velocity ω. Although there are three masses, the motion of the eccentric masses relative to the main mass is prescribed, so that there is only one degree of freedom. The equation of motion for the system is

$$M\ddot{x} + c\dot{x} + kx = m/\omega^2 \sin \omega t \tag{3.4.34}$$

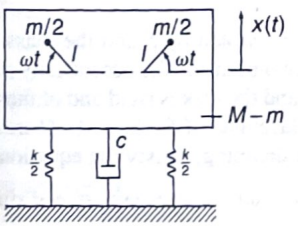


Fig. 3.4.8

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14.1 DISPLACEMENT PUMPS

by Terry L. Henshaw

REFERENCES: Buse, "Positive-Displacement Pumps," Marks' Handbook, 9th ed., McGraw-Hill. Henshaw, "Reciprocating Pumps," Van Nostrand Reinhold. "Hydraulic Institute Standards," 14 ed., Hydraulic Institute. Chesney, Water Injection—Pump Development, paper 68-Pet-11, ASME. Lees, Designing for High Pressures, paper 93-DE-9, ASME. "Positive Displacement Pumps—Reciprocating," API Standard 674, American Petroleum Institute.

NOTE: Much of the text and some of the illustrations are from the author's book: Henshaw, "Reciprocating Pumps," Van Nostrand Reinhold, 1987.

GENERAL

A **displacement pump** (also called **positive-displacement,** or just **p-d**) is a pump which imparts energy to the pumpage (the material pumped) by trapping a fixed volume at suction (inlet) conditions, compressing it to discharge pressure, then pushing it into the discharge (outlet) line. A displacement pump does not rely on velocity to achieve pumping action, as does a centrifugal pump or ejector.

Displacement pumps fall into two major classes: reciprocating and rotary, as illustrated in Fig. 14.1.1.

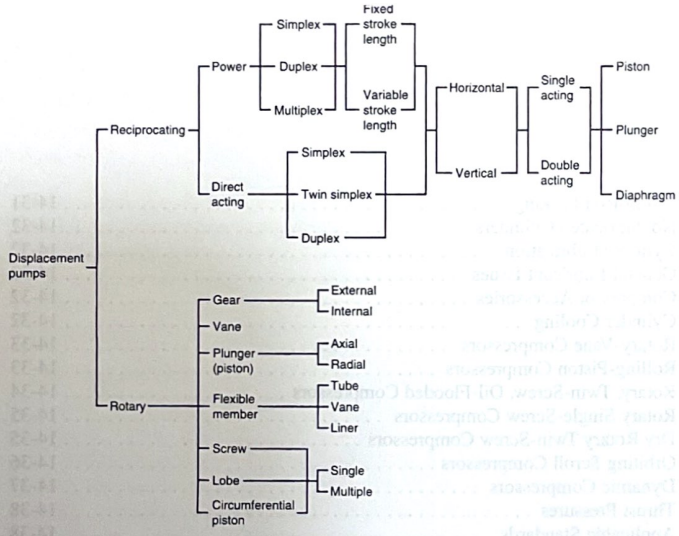


Fig. 14.1.1 Classification diagram of displacement pumps.

Uses and Applications

Displacement pumps serve primarily in applications of low capacity and high pressure, those mostly beyond the capabilities of centrifugal pumps. Some of these services could be performed by centrifugals, but not without an increase in power requirements and/or maintenance. Because displacement pumps achieve high pressures with low pumpage velocities, they are well-suited for abrasive-slurry and high-viscosity services. A reciprocating pump must have special fittings to be suitable for most slurries.

Net Positive Suction Head

Net positive suction head (NPSH), also called **net positive inlet pressure (NPIP)** and **net inlet pressure (NIP),** is the difference between suction pressure and vapor pressure, at the pump suction nozzle, when the pump is running. In a reciprocating pump, NPSH is required to push the suction valve from its seat and to overcome the friction losses and acceleration head within the pump liquid end. In a rotary pump, NPSH is required to push the pumpage into the cavities created by the pumping elements.

If sufficient NPSH is not provided by the system, the pumpage will begin to flash (boil) as it flows into the pump. The vapor will cause a deterioration of pump performance. As the vapor flows into regions of higher pressure in the pump, it condenses back to a liquid. This collapse of the vapor bubbles occurs with significant impact, impinging upon metal surfaces with enough energy to break out small pieces of the metal. This is called **cavitation** damage. The shock created by the bubble collapse may be severe enough to crack a fluid cylinder or break a crankshaft.

All pumps require the system to provide some NPSH. The NPSH provided by the system is called NPSHA (the A is for available). The NPSH required by the pump is called NPSHR. For displacement pumps, NPSHR is normally expressed in pressure units (lb/in^2 or kPa).

Since water usually contains dissolved air, the vapor pressure of the solution is higher than for deaerated water, but this is often overlooked when NPSH calculations are performed. The Hydraulic Institute recommends an NPSH margin of 3 lb/in^2 (20 kPa) for power pumps in systems where the pumpage has been exposed to a gas other than the liquid's own vapor. A liquid (such as propane) at its bubble point in the suction vessel requires no such margin.

RECIPROCATING PUMPS

A **reciprocating pump** is a displacement pump which reciprocates the pumping element (piston, plunger, or diaphragm). The capacity of a reciprocating pump is proportional to its speed, and is relatively independent of discharge pressure.

A **power pump** is one that reciprocates the pumping element with a crankshaft or camshaft (see Figs. 14.1.2 and 14.1.3). It requires a driver which has a rotating shaft, such as a motor, engine, or turbine. A constant-speed power pump will deliver essentially the same capacity at any pressure within the capability of the driver and the strength of the pump.

A **direct-acting** pump is a reciprocating pump driven by a fluid which has a differential pressure (see Fig. 14.1.4). The motive fluid pushes on

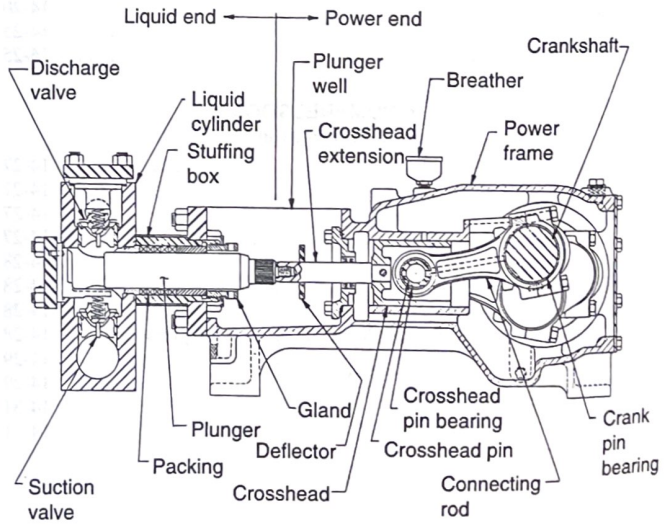


Fig. 14.1.2 Horizontal power pump.

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a piston (or diaphragm) which pushes the pumping element through a rod (or directly on the pumpage).

Reciprocating pumps are classified by the following features:

Drive end (power or direct-acting)
Orientation of centerline of the pumping element (horizontal or vertical)
Number of discharge strokes per cycle of each drive rod (single-acting or double-acting)
Pumping element (piston, plunger, or diaphragm)
Number of drive rods (simplex, duplex, triplex, quintuplex, etc.)

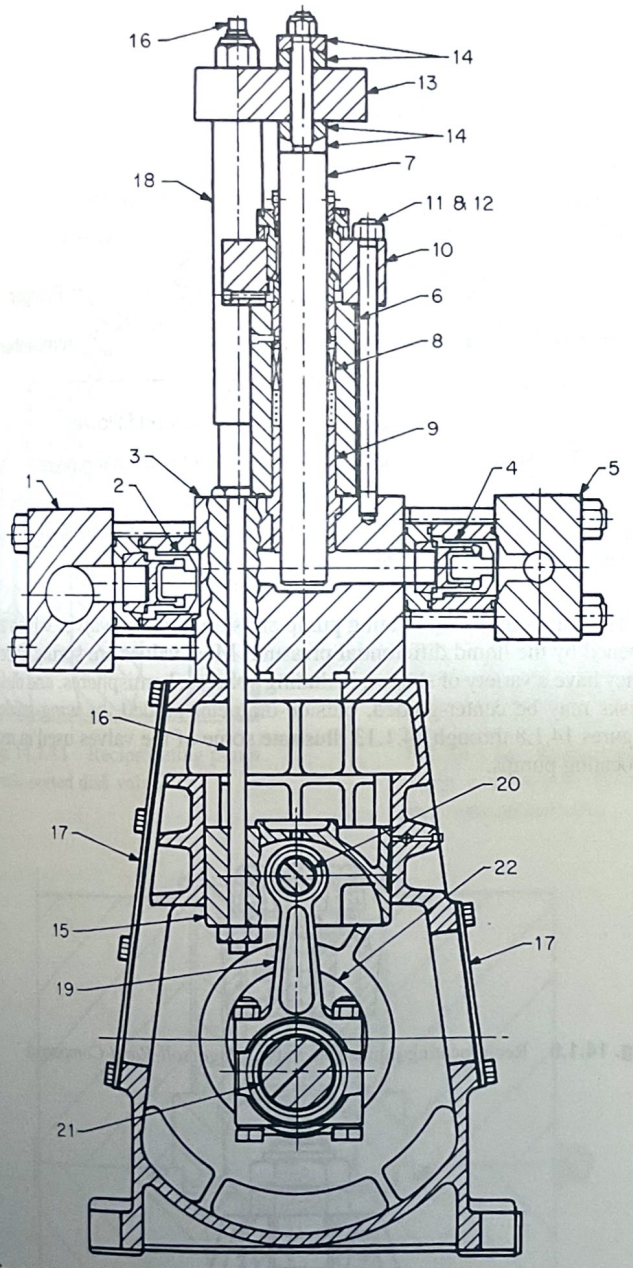


Fig. 14.1.3 Vertical power pump. *(Ingersoll-Rand Company.)*

Uses and Applications for Reciprocating Pumps

The justification for selecting a reciprocating pump instead of a centrifugal or rotary is cost, including costs of power and maintenance.

Reciprocating pumps are best suited for high-pressure/low-capacity services. Such services include high-pressure water-jet cleaning and cutting; glycol injection and charge, as well as amine and lean oil charge, in gas processing; ammonia and carbamate charging for fertilizer production; nuclear-reactor charging and standby control; oil-field saltwater injection and disposal; oil well blowout preventers; pipelines (slurries and crude oil, and injection of ammonia and light hydrocarbons); steel and aluminum mill hydraulic systems; knockout drums in process plants; hydrostatic testing; process slurries; metering; food and chemical homogenizing; well-drilling mud; and car washes.

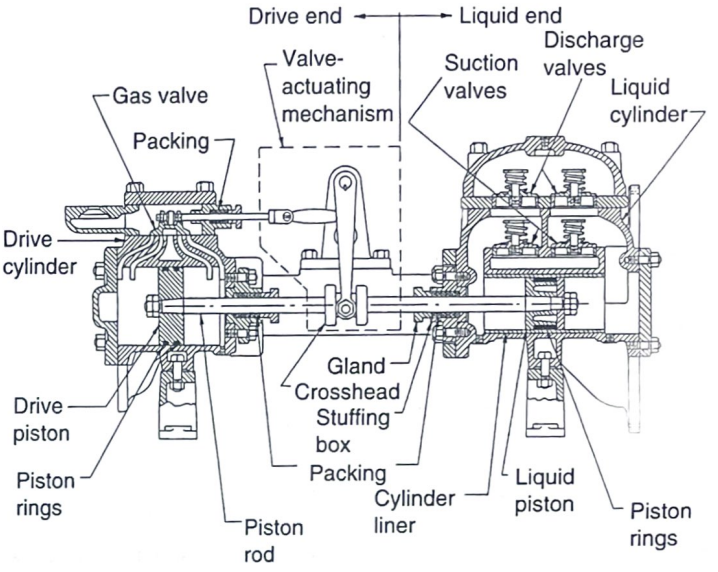


Fig. 14.1.4 Horizontal direct-acting gas-driven pump.

Applications that practically mandate reciprocating units are abrasive and/or viscous slurries above about 500 lb/in^2 (3500 kPa). Examples of these services range from powdered coal to peanut butter.

High-Pressure Applications

Hand-powered pumps, for pressures to 40,000 lb/in^2 (300 MPa), are used for small jacks, deadweight testers, and hydrostatic testing of small components. Small, air-driven direct-acting pumps are used for low-flow, high-pressure systems such as hydrostatic testing. The liquid pressure is approximately the air pressure multiplied ("intensified") by the ratio of the air piston area to the liquid plunger area. They are very simple and will stall and hold a fixed pressure without using power; but when pumping, they consume relatively large amounts of air. Available up to 10 hp (7 kW), the most popular sizes are up to 2 hp (1.5 kW). Pressures range to 100,000 lb/in^2 (700 MPa).

Hydraulically driven **intensifiers** operate on the same principle, but are more efficient, and because of the higher pressures commonly available from hydraulic systems, can produce high pressures with lower intensification ratios. Intensifiers are available to 300,000 lb/in^2 (2100 MPa) for laboratory-scale hydrostatic applications. Intensifier systems for water jetting are available from 30,000 to 60,000 lb/in^2 (210 MPa to 410 MPa) up to 200 hp (150 kW). They are powered by electric motors or diesel engines. Figure 14.1.5 shows the pumping end of an intensifier.

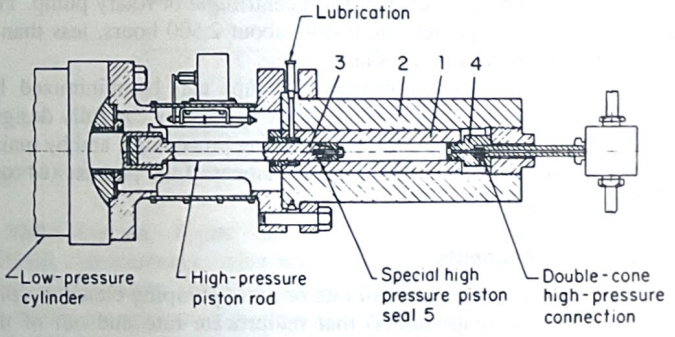


Fig. 14.1.5 Pumping end of a direct-acting, liquid-driven, "intensifier" pump. The check valves are in the tubing.

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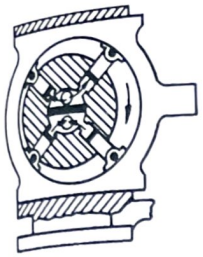


Fig. 14.1.32 Radial-plunger rotary pump.

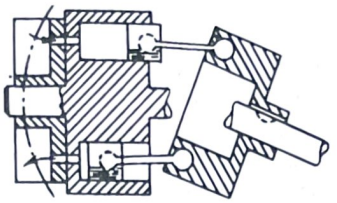


Fig. 14.1.33 Axial-plunger (swash-plate) rotary pump.

of each plunger is established by the eccentricity of the rotor. In the axial pump, the stroke length is established by the angle of the plate or shaft. On some pumps, the eccentricity (or angle) is adjustable, providing variable displacement. Rotary plunger pumps are used in hydraulic systems to provide power to hydraulic motors and cylinders.

Circumferential-Piston Pump Although sometimes considered a lobe pump, this unit (shown in Fig. 14.1.34) differs from the lobe unit in that there is no close clearance between the two rotors. Close clearance does exist, though, between each rotor and adjacent stationary parts.

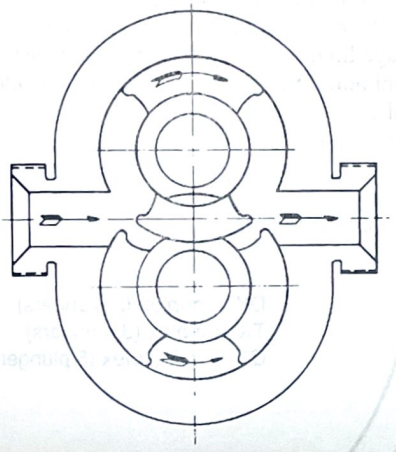


Fig. 14.1.34 Circumferential-piston rotary pump. *(Hydraulic Institute.)*

The **casing** (also called a **body** or **housing**) is the main pressure-retaining component of a rotary pump. It supports the rotor, contains the suction and discharge nozzles, and usually forms a portion of the boundaries of the pumping chambers.

The **bearings** of a rotary pump may be internal or external. If internal, they are lubricated by the pumpage. If the pumpage is not suitable for bearing lubrication, the bearings must be external. External bearings require two pumpage seals on each shaft. Some rotary pumps contain four seals.

Figure 14.1.35 shows typical **rotary pump performance curves.** Capacity, mechanical efficiency, and power are plotted against pressure.

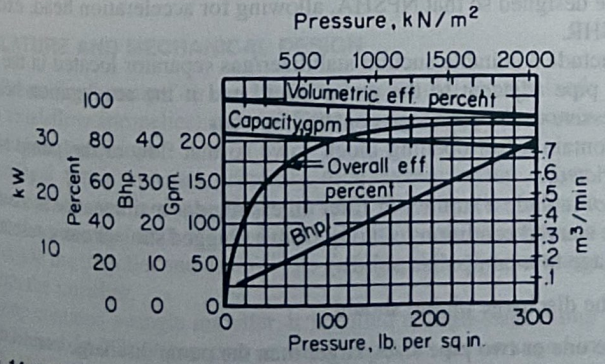


Fig. 14.1.35 Rotary pump performance curves for a herringbone gear pump at 600 r/min with a 400-SSU (90-mm²/s) liquid.

The difference between displacement and capacity is called **slip.** Power input is the sum of power output and power losses (due to friction and slip).

As defined by the Hydraulic Institute, the NPSH required by a rotary pump is that value of NPSHA when cavitation noise, a sharp drop in capacity, or a 5 percent reduction in capacity occurs, whichever occurs first.

Increased Running Clearances An increase in the clearances between the rotating parts, and/or between the rotating and stationary parts, of a rotary pump will cause the slip to increase. An increase in slip causes the volumetric efficiency to drop, the mechanical efficiency to drop, and the NPSHR to rise. If the unit is running at a constant speed, the lower volumetric efficiency (lower capacity) will result in a lower discharge pressure if the system head varies with capacity.

VOLUMETRIC EFFICIENCY

Volumetric efficiency η_v of a displacement pump is the ratio of capacity Q to displacement D: $\eta_v = Q/D$. Capacity is the volume flow rate in the suction pipe. Displacement is the volume swept by the pumping elements per unit time. For a single-acting triplex plunger pump, displacement may be calculated as follows: $D = 3(\pi/4)d^2LN$, where D = displacement; d = diameter of plunger; L = stroke length; and N = crankshaft speed, r/min.

To predict the capacity of a reciprocating pump, in the selection process, it is necessary to predict the volumetric efficiency. Volumetric efficiency may be calculated with the following equation:

$$\eta_v = 1 - \frac{S}{D'} - \frac{C}{D'}\left(\frac{v_s}{v_d} - 1\right) \qquad (14.1.2)$$

where η_v = volumetric efficiency; S = slip through both suction and discharge valves, typically about 3 percent of D'; D' = displacement of one pumping element during one stroke; C = clearance volume (dead space) in the pumping chamber (when the pumping element is at the end of the discharge stroke); v_s = specific volume of pumpage at suction conditions; and v_d = specific volume of the pumpage at discharge conditions.

Although volumetric efficiency has an effect on mechanical efficiency, in reciprocating pumps the two do not necessarily move in unison. It is possible to have a high η_m with a low η_v (due to liquid compressibility), or a low η_m with a high η_v (such as would occur in a high-suction-pressure, low-differential-pressure application).

The same equation can be applied to rotary pumps, but usually $v_s = v_d$, so that Eq. (14.1.2) reduces to $\eta_v = 1 - S/D'$. For a rotary pump, S = leakage of pumpage from the discharge side of the pump to the suction side. Because rotary pumps have no check valves, they must rely on close clearances to restrict slip. Slip is a function of the differential pressure, the viscosity of the pumpage, and the clearance between the parts. Slip in a rotary pump is usually a higher fraction of displacement than in a reciprocating pump.

POWER OUTPUT AND INPUT

The **power output** (also called **hydraulic power, hydraulic horsepower,** and **water horsepower**) of any pump is the usable power it imparts to the pumpage. It is the product of capacity and differential pressure.

In U.S. units, $P_o = Q \times \Delta p/1{,}714 = Q(p_d - p_s)/1{,}714$, where P_o = power output, hp; Q = pump capacity, gal/min; Δp = pump differential pressure, lb/in²; p_d = pump discharge pressure, lb/in²; and p_s = pump suction pressure, lb/in².

In metric units, $P_o = Q \times \Delta p/3{,}600 = Q(p_d - p_s)/3{,}600$, where P_o = power output, kW; Q = pump capacity, m³/h; Δp = pump differential pressure kPa; p_d = pump discharge pressure, kPa; and p_s = pump suction pressure, kPa.

The **power input** (sometimes called **brake horsepower**) is the power required to drive power and rotary pumps. Normally, the power input includes the losses in an integral gear unit, but not the losses in a separate

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gear unit or variable-speed drive. V-belt losses may be included. If the mechanical efficiency is known, power input may be calculated by dividing the power output by the mechanical efficiency.

MECHANICAL EFFICIENCY—POWER AND ROTARY PUMPS

Mechanical efficiency (also called **pump efficiency** and **overall efficiency**) of a power-driven pump is the ratio of output power P_o to input power P_i: $\eta_m = P_o/P_i$.

As shown in Fig. 14.1.36, mechanical efficiency drops as the frame load is reduced, because the power output falls faster than friction losses, and becomes a smaller part of the power input. Mechanical efficiency is zero when the differential pressure is zero.

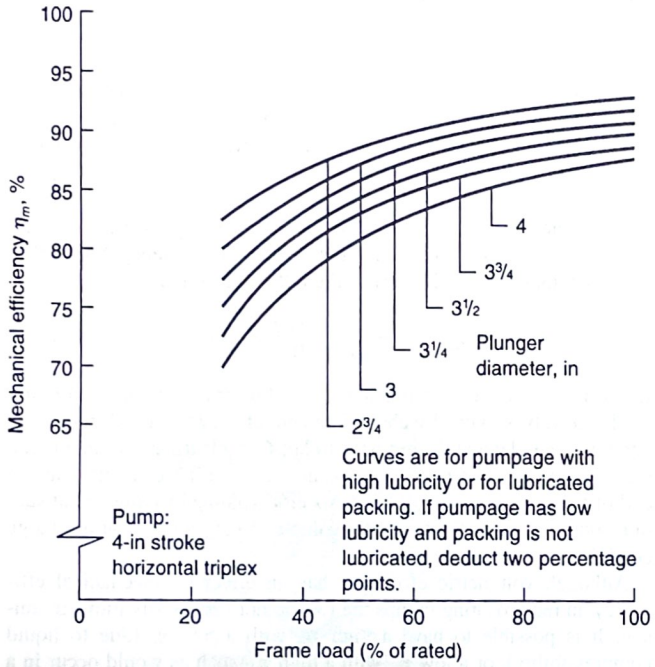


Fig. 14.1.36 Mechanical (overall) efficiency of a triplex power pump.

PULSATION DAMPENERS

A **pulsation dampener** is a device which reduces the pressure pulsations created, in a suction or discharge pipe, by the interaction of a pump and a system. The dampener performs this function by reducing the velocity variations of the pumpage in the pipe.

Most reciprocating pumps, and many rotary pumps, create a variation in the velocity of the pumpage in the piping. Unless the suction and discharge vessels are very close to the pump, these velocity variations will be converted by the system into pressure variations. To reduce these pulses, a small vessel can be installed in the suction and/or discharge line. Such a vessel mounted in the discharge line is called a **pulsation dampener.** If the suction pressure is high enough, this same type of device can be used in the suction line, but a special dampener, called a **suction stabilizer** will perform the dual function of reducing pulses and separating free gas from the pumpage.

For optimum effectiveness, each dampener must be installed close to the pipe and adjacent to the pump. There must be minimal restriction between the dampener and pipe. For a low suction pressure, a simple air-filled standpipe is often adequate. For the discharge side, and for high suction pressures, the dampener often contains a diaphragm or bladder that is precharged with nitrogen to about 70 percent of system pressure.

SYSTEM DESIGN

A properly designed system is necessary for a satisfactory installation. Improper design will result in a system that vibrates and is noisy. Pulsations may be severe enough to damage pump components and instrumentation.

Field experience and information from the Hydraulic Institute standards are condensed below into a list of general design guidelines for the (1) suction vessel, (2) suction piping, and (3) discharge piping. The asterisks in the following lists show features that *do not* apply to rotary pumps that produce a uniform flow rate (constant velocity).

The suction vessel should:

Be large enough to provide sufficient retention time to allow free gas to rise to the liquid surface.

Have the feed and return lines enter below the minimum liquid level.

Include a vortex breaker over the outlet (pump suction) line and/or provide sufficient submergence to preclude vortex formation.

Contain a weir plate to force gas bubbles toward the surface. The top of the weir must be sufficiently below the minimum tank level to avoid disturbance.

The suction piping should:

Be as short and direct as possible.

Be one or two pipe sizes larger than the pump suction connection.

Have an average liquid velocity less than the values in Fig. 14.1.37.*

Contain a minimum number of turns, which should be long-radius elbows or laterals.*

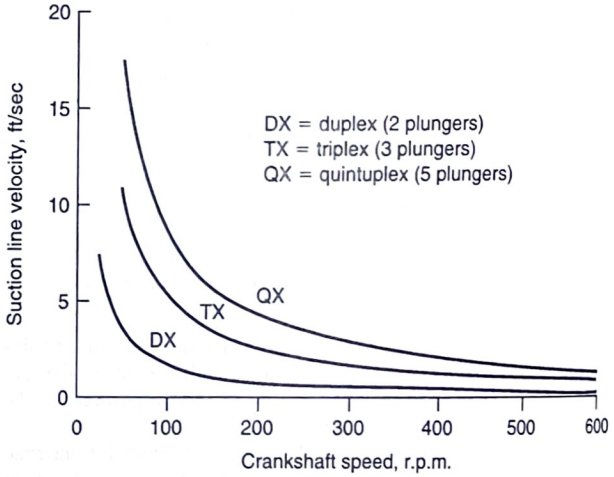


Fig. 14.1.37 Suction line velocities for constant acceleration head, for reciprocating power pumps.

Be designed to preclude the collection of vapor in the piping. (There should be no high points, unless vented. The reducer at the pump should be the eccentric type, installed with the straight side up.)

Be designed so that NPSHA, allowing for acceleration head, exceeds NPSHR.

Include a suitable suction stabilizer/gas separator located in the suction pipe adjacent to the pump liquid end if the acceleration head is excessive or the pumpage contains free gas.

Contain a full-opening block valve so that flow to the pump is not restricted.

Not include a strainer or filter unless regular maintenance is assured. (The starved condition resulting from a plugged strainer can cause more damage to a pump than solids.)

The discharge piping should:

Be one or two pipe sizes larger than the pump discharge connection.

Have an average velocity less than 3 times the maximum suction line velocity.

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