

Section 14

Fans, Pumps, and Compressors

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14.1 DISPLACEMENT PUMPS

by T. 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.

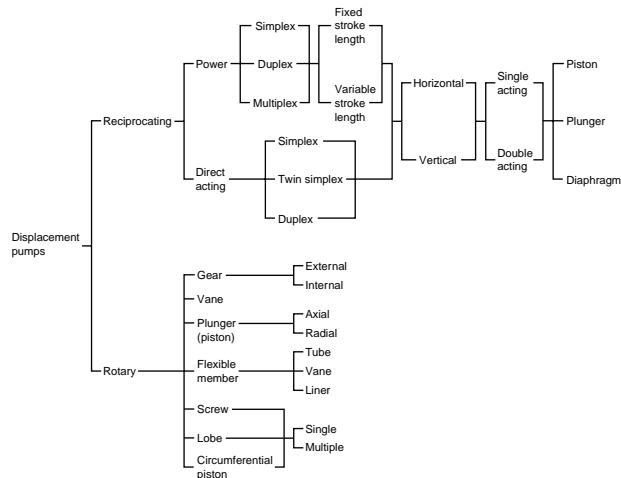


Fig. 14.1.1 Classification diagram of displacement pumps.

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/in² 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/in² (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

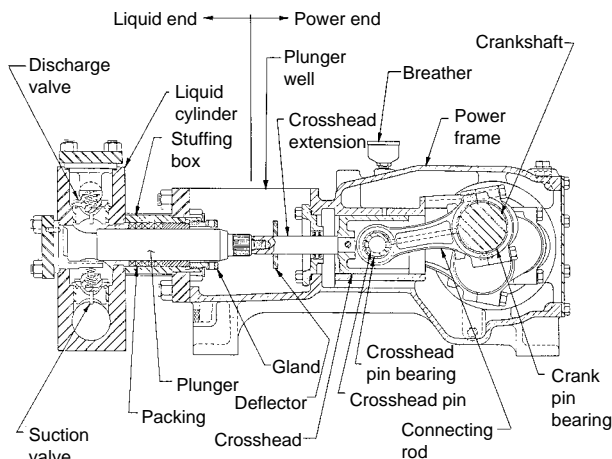


Fig. 14.1.2 Horizontal power pump.

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.)

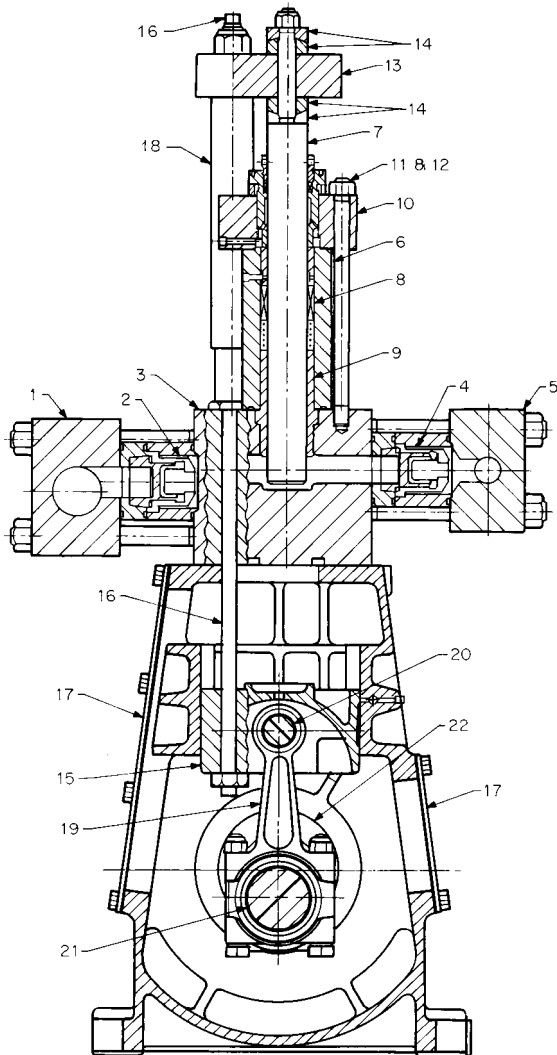


Fig. 14.1.3 Vertical power pump. (Ingersoll-Rand Company.)

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.

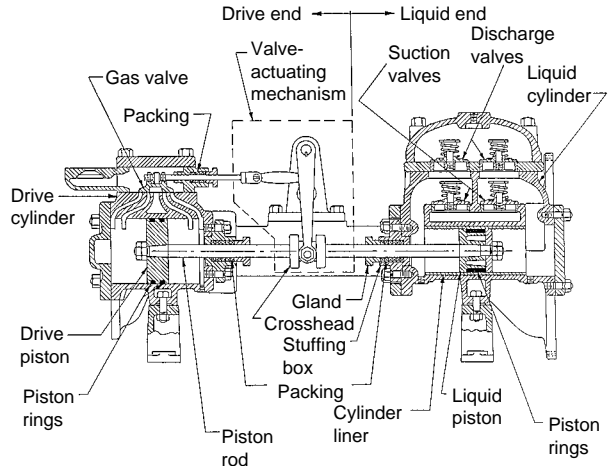


Fig. 14.1.4 Horizontal direct-acting gas-driven pump.

Applications that practically mandate reciprocating units are abrasive and/or viscous slurries above about 500 lb/in² (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/in² (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/in² (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/in² (2100 MPa) for laboratory-scale hydrostatic applications. Intensifier systems for water jetting are available from 30,000 to 60,000 lb/in² (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.

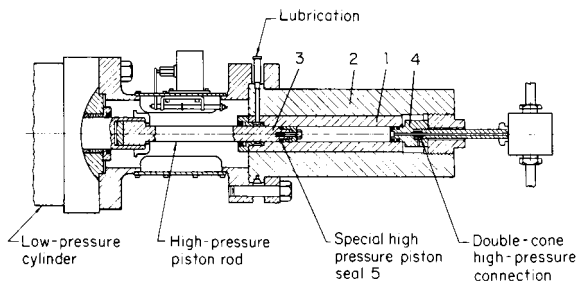


Fig. 14.1.5 Pumping end of a direct-acting, liquid-driven, "intensifier" pump. The check valves are in the tubing.

14-4 DISPLACEMENT PUMPS

Power pumps (usually triplex or quintuplex) are available from about 5 to 150 hp (5 to 100 kW) with pressures to 40,000 lb/in² (280 MPa), and up to 2,000 hp (1500 kW) at 20,000 lb/in² (140 MPa). Power pumps are efficient and mechanically simple. Figure 14.1.6 shows a power pump liquid end for a 36,000 lb/in² (250 MPa) water-jetting pump.

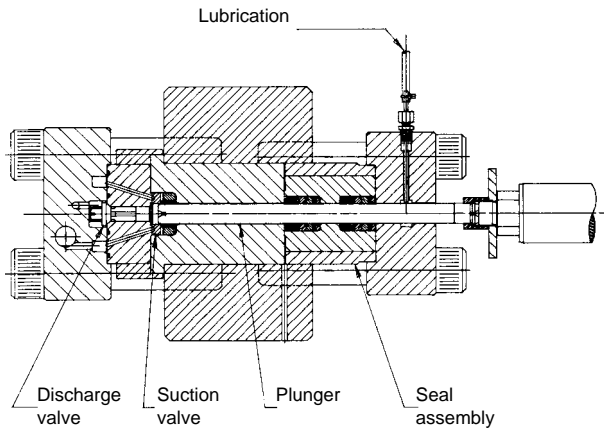


Fig. 14.1.6 Power pump liquid end for 36,000 lb/in² (250 MPa) water-jetting. (NLB Corporation.)

Slurry Applications

The standard reciprocating pump is not designed to handle slurries. Modifications to standard designs, and in some cases special designs, are required to achieve satisfactory operation and component life.

To achieve satisfactory packing and plunger life, abrasive slurry must be prevented from entering the packing. Methods include a wiper ring between the pumpage and packing, a long throat bushing, injection of a clean liquid into the throat area, insertion of a diaphragm or floating piston between the plunger and the pumpage, and complete removal of the valve assembly from the stuffing box area. This last arrangement requires a liquid column between the valve assembly and the stuffing box, which increases the clearance volume and acceleration head within the pump.

Special pump valves are usually required for slurries. Depending on the nature of the solids, they can be ball, bell, bevel seat with elastomer insert, wing-guided with reduced seating area, or disk with special seating surfaces. Special construction can prevent the slurry from contacting the packing, but the valves cannot avoid contact with the slurry.

Problems and How to Avoid Them

Reciprocating pumps have some disadvantages, the most common being pulsating flow. Because of the pulsation, special consideration must be given to system design. Guidelines are provided later in this section.

In most applications, the initial and maintenance costs for a reciprocating pump will be greater than for a centrifugal or rotary pump. The packing in a typical power pump lasts about 2,500 hours, less than a mechanical seal on a rotating shaft.

Most problems with reciprocating pumps can be minimized by selecting pumps to operate at conservative speeds, by carefully designing the pumping system, by careful operating procedures, and by maintenance practices which preserve the alignment of the plunger (or rod) with the stuffing box.

Liquid-End Components

All reciprocating pumps contain one or more pumping elements (pistons, plungers, or diaphragms) that reciprocate into and out of the pumping chambers. (In reciprocating pump terminology, a **piston** is a cylindrical disk which mounts on a rod. A **plunger** is just a smooth rod.)

Each pumping chamber contains at least one suction and one discharge valve.

The liquid cylinder is the major pressure-retaining part of the liquid end, and forms the major portion of the pumping chamber.

A piston pump is normally equipped with a replaceable liner (sleeve) that absorbs the wear from the piston rings. Because a plunger contacts only stuffing-box components, plunger pumps do not require liners.

Sealing between the pumping chamber and atmosphere is accomplished by a **stuffing box** (Fig. 14.1.7). The stuffing box contains rings of packing that conform to and seal against the stuffing box bore and the rod (or plunger).

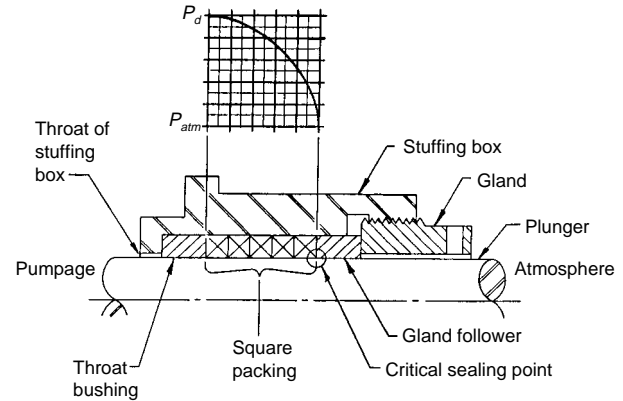


Fig. 14.1.7 Reciprocating pump stuffing box (nonlubricated) showing pressure gradient during discharge stroke.

The valves in a reciprocating pump are simply check valves which are opened by the liquid differential pressure. Most valves are spring-loaded. They have a variety of shapes, including spheres, hemispheres, and disks. Disks may be center-guided, outside-diameter-guided, or wing-guided. Figures 14.1.8 through 14.1.13 illustrate some of the valves used in reciprocating pumps.

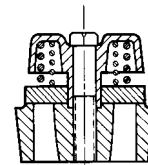


Fig. 14.1.8 Reciprocating pump disk valve. (Ingersoll-Rand Company.)

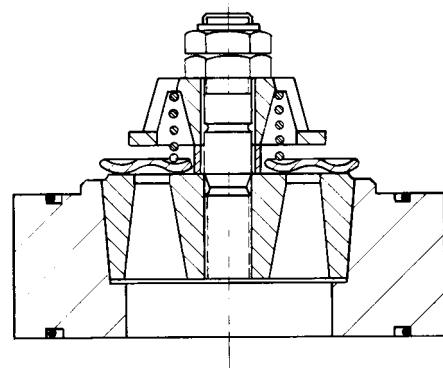


Fig. 14.1.9 Reciprocating pump disk valve. (Durabla Manufacturing Company.)

The Pumping Cycle

As the pumping element reverses from the discharge stroke to the suction stroke, liquid within the pumping chambers expands, and the pressure drops. Since most liquids are relatively incompressible, little movement of the element is required to reduce the pressure. When the

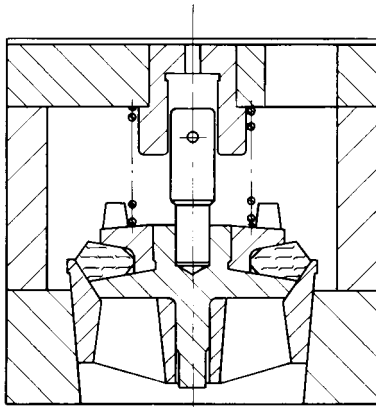


Fig. 14.1.10 Reciprocating pump elastomeric-insert valve. (TRW-Mission Manufacturing Co.)

pressure drops sufficiently below suction pressure, the differential pressure (suction pressure minus chamber pressure) pushes the suction valve open. This occurs when the element is moving slowly, so that the valve opens slowly and smoothly as the velocity of the element increases. Liquid then flows through the valve and follows the element on its suction stroke. As the element decelerates near the end of the stroke, the suction valve returns to its seat. When the element stops, the suction valve closes.

The pumping element then reverses and starts its discharge stroke. The liquid trapped in the pumping chamber is compressed until chamber pressure exceeds discharge pressure by an amount sufficient to push the discharge valve from its seat. The action of the discharge valve, during the discharge stroke, is then the same as that of the suction valve during the suction stroke.

Power Pumps

The drive end of a power pump is called a **power end** (see Fig. 14.1.2). Its function is to convert rotary motion from a driver into reciprocating motion for the liquid end. The main component of the power end is the **power frame**, which supports all other power end parts and, usually, the liquid end. The second major component of the power end is the **crankshaft** (sometimes a camshaft). The crankshaft transmits power from the driver to the connecting rods.

The **connecting rod** is driven by a throw (journal) of the crankshaft on one end, and drives a crosshead on the other. The crankshaft moves in pure rotating motion, the crosshead in pure reciprocating motion. The connecting rod is the link between the two. **Main bearings** support the shaft in the power frame.

The **crosshead** transmits the force from the connecting rod into the pumping element. It is fastened directly to a plunger or piston rod, or to a rod called an **extension, stub, or pony rod**. The other end of this rod is fastened to the pumping element.

Power pump overall efficiencies normally range from 85 to 94 percent, higher than any other type of pump.

Selection Graphs Different manufacturers produce these graphs in different configurations. All graphs consist of straight lines since displacement is directly proportional to speed. Figure 14.1.14 shows one such graph for a triplex power pump. Displacement is plotted against crankshaft speed for the different plunger diameters available in the pump. The pressures listed are those at which each plunger size would operate to load the power end to its rated value (4,500 lbf or 20,000 N).

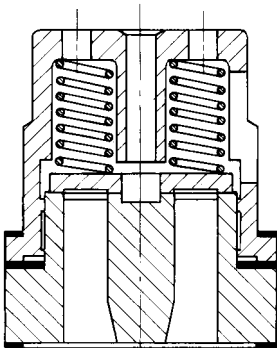


Fig. 14.1.11 Reciprocating pump double-ported disk valve.

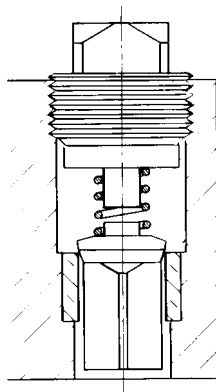


Fig. 14.1.12 Reciprocating pump conical-faced (bevel-seat) wing-guided valve.

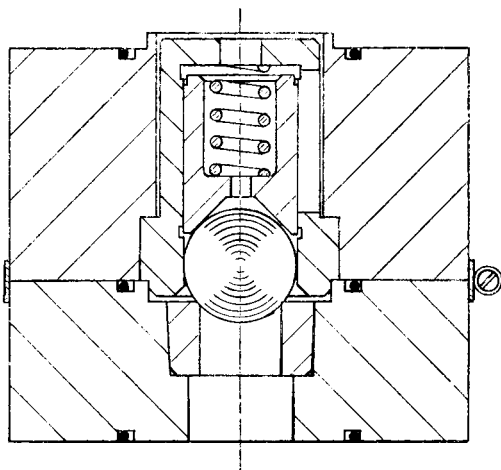


Fig. 14.1.13 Reciprocating pump ball valve. (Ingersoll-Rand Company.)

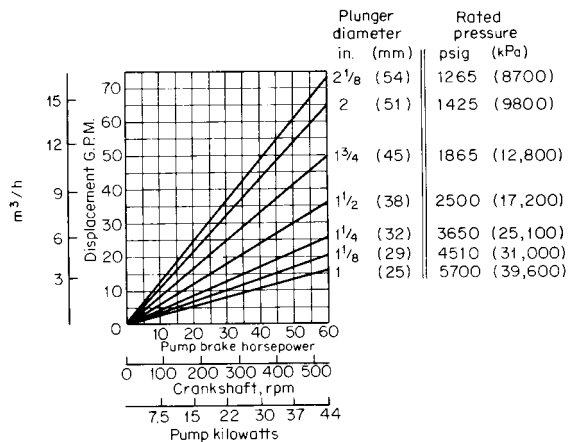


Fig. 14.1.14 Power pump selection graph. (Ingersoll-Rand Company.)

NPSH Curves Figure 14.1.15 provides insight into power pump NPSHR characteristics, valve action, and the effect of valve springs on pump performance. These curves are for a 3-in (76-mm) stroke, horizontal, triplex plunger power pump with 2 1/4-in-diameter (57-mm) plungers, and with suction valves that operate vertically. The valves are wing-guided, and have a diameter approximately equal to the plunger diameter.

Because the axis of the suction valve is vertical, the valve can operate without a spring if the pump speed is kept low. Curve A represents the NPSH requirements with no springs on the suction valves. NPSHR at 100 r/min is only 0.7 lb/in² (5 kPa) (1.6 ft or 0.5 m of water), less than for most centrifugals.

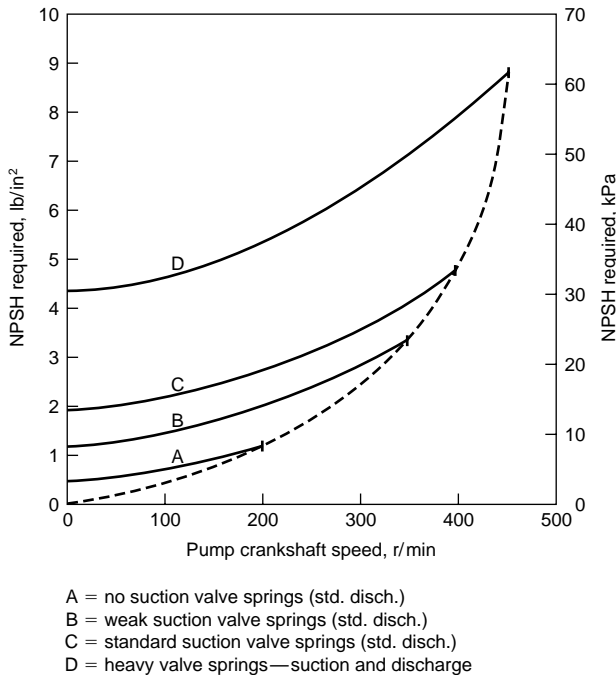


Fig. 14.1.15 NPSH curves for a 3-in- (75-mm-) stroke triplex power pump.

The speed of the pump in this configuration is limited by the ability of the suction valve to keep up with the plunger. Since there is no spring to push the valve back onto its seat, gravity is the only force tending to close the valve against the entering fluid. If the pump is run too fast, the valve will still be off its seat when the plunger reverses to the discharge stroke. The liquid will then momentarily flow backward through the seat, and the valve will be slammed onto the seat, sending a shock wave into the suction manifold and piping. At that moment, the plunger is moving at a finite velocity, but the discharge valve is still closed. The pressure in the pumping chamber will quickly exceed discharge pressure, and the discharge valve will be driven from its seat. A shock wave will be transmitted from the pumping chamber through the discharge manifold, and into the discharge line. The vertical line at the end of Curve A indicates the maximum speed for proper suction valve operation.

Curve B is for weak springs added to the suction valves. Because both spring force and valve weight must now be overcome to open the suction valve, NPSHR has increased about 100 percent over Curve A. These springs get the suction valves onto their seats quicker, so that operation is smooth at higher speeds.

If speeds beyond the end of Curve B are desired, stronger suction valve springs are required. Stronger springs allow operation to 400 r/min. NPSH requirements are about 3 times those of Curve A, ranging from about 2 to 5 lb/in² (14 to 35 kPa).

Curves A through C represent a pump equipped with the same (standard) discharge valve spring. Only the suction spring has been changed.

If operation is required at speeds exceeding the limits of Curve C, extra-strong springs are required on both suction and discharge valves. As shown by Curve D, NPSHR is about double that required for the standard springs, ranging from about 4.5 to 9 lb/in² (31 to 62 kPa).

The dashed line passing through the ends of the NPSH curves illus-

trates the dramatic increase of NPSHR as pump speed is increased. When it is necessary to change suction springs, NPSHR increases approximately as the **square** of speed.

To minimize the problem of dissolved air, the NPSH tests that produced these curves were performed with water at, or near, its boiling point in the suction vessel.

Reciprocating pumps, under the correct conditions, can operate with a suction pressure below atmospheric. Such a situation, though, can lead to air being drawn through the stuffing box packing and into the pumping chamber on the suction stroke. This air will cause as many problems as air entrained in the pumpage. Capacity will drop, the pump may operate noisily, the system may vibrate, and damage may occur to pump and system components.

This inward air leakage can be reduced by an external sealing liquid, such as lubricating oil, being directed onto the plunger surface or into the packing.

Test Criteria for NPSH Power pump NPSH tests are performed by holding the pump speed and discharge pressure constant, and varying the NPSH available (NPSHA) in the system. Capacity remains constant for all NPSHA values above a certain point. Below this NPSHA value, capacity begins to fall. NPSHR is defined as the NPSH available when the capacity has dropped 3 percent.

Acceleration Head Because the velocities in the suction and discharge piping are not constant, the pumpage must accelerate a number of times for each revolution of the crankshaft. Since the liquid has mass and, therefore, inertia, energy is required to produce the acceleration. This energy is returned to the system upon deceleration. Sufficient pressure, however, must be provided to accelerate the liquid on the suction side of the pump to prevent cavitation in the suction pipe and/or the pumping chambers. The drop in pressure, below the average, on the suction side, caused by this acceleration, is called **acceleration head**.

An approximation of acceleration head for power pumps is given by the equation $h_a = LvNC/gK$, where h_a = acceleration head, ft (m) of liquid being pumped; L = actual length of suction line (not equivalent length), ft (m); v = average liquid velocity in suction line, ft/s (m/s); N = speed of pump crankshaft, r/min; C = constant, depending on type of pump; g = gravitational constant, ft/s² (m/s²); and K = fluid compressibility correction factor. Values for the constants C and K are:

Pump type	C
Simplex, single-acting	0.400
Duplex, single-acting	0.200
Triplex	0.066
Quintuplex	0.040
Septuplex	0.028
Nonuplex	0.022

Fluid	K
Slightly compressible liquids such as deaerated water	1.4
Most liquids	1.5
Highly compressible liquids (such as ethane)	2.5

Note that increasing the pump speed, without changing the suction line, increases h_a as the square of speed, because both v and N increase proportionately with speed.

Because this equation does not adequately compensate for pumpage and system elasticity, it is recommended only for short, rigid suction lines.

To reduce acceleration head in a suction line, a bottle or suction stabilizer may be installed in the pipe, adjacent to the pump.

Figure 14.1.16 shows the theoretical flow rates for various types of power pumps during one revolution of the crankshaft. Pumps with an odd number of cranks have the same flow variations for both single-acting and double-acting liquid ends (except the simplex). Flow curves for septuplex and nonuplex pumps are similar to the quintuplex curve, with

more but smaller variations, as indicated in the tabulation. The values tabulated will vary slightly with variations in the ratio of connecting rod length to stroke length. These curves indicate flow-rate variations, not **pressure pulsations**. Pressure pulsations are a product of the acceleration created by the pump and the mass of the pumpage in the system.

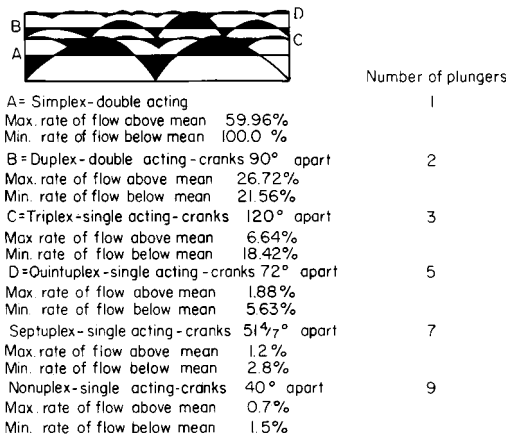


Fig. 14.1.16 Flow-rate variations in reciprocating power pumps.

Actual flow rates may vary considerably from these curves, particularly with high-speed pumps, where valve closing and opening may lag the crankshaft rotation as much as 15°, and at high pressures, where liquid compression may cause a further lag of 5° to 10° (or more for a large clearance volume).

Power Pump Speeds Probably the most controversial factor in the selection of power pumps is the maximum allowable speed. Some manufacturers, for example, offer 3-in- (76-mm-) stroke triplex pumps for operation at 500 r/min. One builder of portable water-jetting units routinely operates this size pump at 600 r/min. Most users of this size pump, in continuous-duty applications, prefer to operate it no faster than about 400 r/min.

Top speeds quoted by one manufacturer for the 3-in-stroke triplex increased over a period of years from 150 to 520 r/min. Although improvements in power end and liquid end construction made the pump capable of operating at higher speeds, albeit with shorter packing life, a major obstacle to higher speeds is system design. Initially, some thought that a smaller pump, running faster to achieve the same capacity, would produce less pulsation in the suction and discharge pipes. Unfortunately, the opposite is true.

The velocity variation in the piping for a triplex pump is 25 percent, regardless of the pump size or speed (see Fig. 14.1.16). For the same capacity, a smaller pump, running faster, will produce the same maximum and minimum flow rates and more pulses per second. Since the acceleration head is proportional to frequency, doubling the pump speed doubles the acceleration head, thereby reducing the NPSH available from the system. Also, doubling the speed requires a much stiffer valve spring, thereby significantly increasing the NPSHR of the pump. If the NPSHA drops below the NPSHR, cavitation and pounding will occur.

The best solution for excessive acceleration is an effective dampener in the suction line; but such devices often are less effective at higher frequencies, due to the inertia of their moving parts and inertia of the liquid inside them, which must oscillate for them to function.

Through field observations, the author developed a set of **maximum recommended speeds** for plunger-type power pumps in continuous-duty services. These speeds, as adopted by API 674, are shown in Fig. 14.1.17, along with the Hydraulic Institute basic speeds. Intermittent and cyclic operation is often satisfactory above these speeds. Lower speeds may be dictated by factors such as pump construction, system design, low NPSHA, entrained gas, high temperature, entrained solids, high viscosity, and requirements for a low sound level.

The **minimum speed** of a power pump is determined by its ability to provide sufficient lubrication to all bearing surfaces in the power end. Some units can run satisfactorily at 20 r/min. Others must be maintained at 100 r/min or more.

Plunger load (also called **frame load** and **rod load**) is the force transmitted to the power end by one plunger. For a single-acting pump, the discharge plunger load is calculated by multiplying discharge pressure by the cross-sectional area of one plunger. Suction plunger load is the suction pressure multiplied by the plunger area.

A power pump is rated by the maximum discharge plunger load that the power end is capable of absorbing when the suction pressure is zero. Some units are rated for continuous operation, some for intermittent operation, and some are rated both ways.

Torque Characteristics For fixed suction and discharge pressures, a power pump requires an **average** input torque that is independent of speed (except for increases at very low and very high speeds). The torque actually has a variation that mirrors the discharge flow-velocity curve. A power pump will require the same torque at one-half or one-quarter rated speed and, therefore, will require one-half or one-quarter rated power, respectively. Figure 14.1.18 illustrates the variation of average torque with speed for a typical triplex power pump.

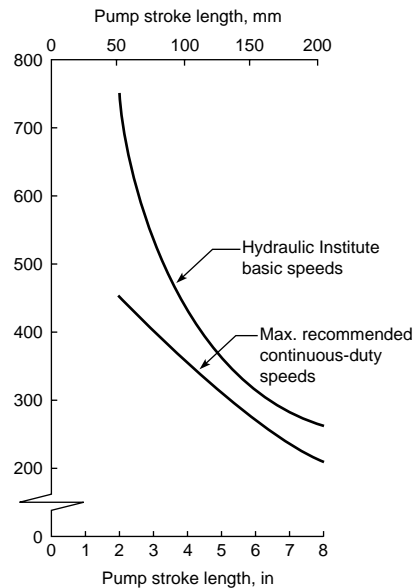


Fig. 14.1.17 Maximum recommended speeds for plunger-type power pumps in continuous-duty services.

The curve for full-load torque in Fig. 14.1.18 is for starting against full discharge pressure. Breakaway torque is about 150 percent of average full-load running torque. As speed is increased, and proper lubrication is established in the power end and packing, torque drops to the full-load, full-speed value, and is thereafter constant up to full speed.

For a start-up that is easier on equipment, pump discharge is piped back to the suction vessel, making the discharge pressure near suction pressure. The no-load curve in Fig. 14.1.18 illustrates the resulting torque requirements imposed by the pump on the drive train. Breakaway torque is approximately 25 percent of full-load torque. This will vary, depending on the type of packing and bearings in the pump and the length of time the pump has been idle. As speed increases, the torque drops to less than 10 percent of the full-load torque.

Packing The component normally requiring the most maintenance on reciprocating pumps is packing. Although the life of packing in a power pump is typically about 2,500 h (3 months), some installations,

with special stuffing box arrangements, have experienced lives of more than 18,000 h (2 years), at discharge pressures to 4,000 lb/in² (28 MPa).

Short packing life can result from any of the following conditions: (1) misalignment of plunger (or rod) with stuffing box; (2) worn plunger, rod, stuffing-box bushings, or stuffing-box bore; (3) improper

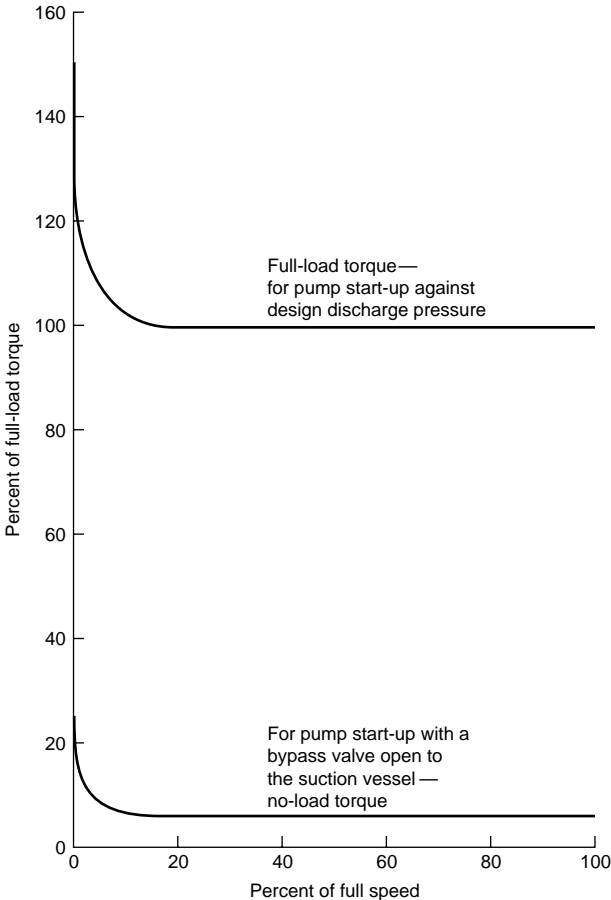


Fig. 14.1.18 Torque-speed curves for a typical triplex power pump.

packing for the application; (4) insufficient or excessive lubrication; (5) packing gland too tight or too loose; (6) excessive speed or pressure; (7) high or low temperature of pumpage; (8) excessive friction (too much packing); (9) packing running dry (pumping chamber gas-bound); (10) shock conditions caused by entrained gas or cavitation, broken or weak valve springs, or system problems; (11) solids from the pumpage, environment, or lubricant; (12) improper packing installation or break-in; and (13) icing caused by volatile liquids that refrigerate and form ice crystals on leakage to atmosphere, or by pumpage at temperatures below 32°F (0°C). As is evident from this list, short packing life can indicate problems elsewhere in the pump or system.

To achieve a low leakage rate, the clearance between the plunger (or rod) and packing must be essentially zero. This requires that the sealing rings be relatively soft and pliant. Because the packing is pliant, it tends to flow into the stuffing box clearances, especially between the plunger and follower bushing. If this bushing does not provide an effective barrier, the packing will extrude, and leakage will increase.

A set of square or V-type packing rings will experience a pressure gradient, during operation, as indicated in Fig. 14.1.7. The last ring of packing, adjacent to the gland-follower bushing, will experience the largest axial loading, resulting in greater deformation, tighter sealing

and, therefore, the largest pressure drop. The gap between the plunger and the follower must be small enough to prevent packing extrusion. Most packing failures originate at this critical sealing point.

Because this last ring of packing is the most critical, does the most sealing, and generates the most friction, it requires more lubrication than the others. In a nonlubricated arrangement (Fig. 14.1.7), this ring must rely on the plunger to drag some of the pumpage back to it in order to provide cooling and lubrication. Therefore, to maximize packing life, the overall stack height of the packing should not exceed the stroke length of the pump.

Because the last ring of packing requires more lubrication than do the others, lubrication of the packing from the atmospheric side is more effective than injecting oil into a lantern ring located in the center of the packing. Care must be exercised to get the lubricant onto the plunger surface and close enough to the last ring, so that the stroke of the plunger will carry the lubricant under the ring.

During the first few hours of pump operation with new packing, each stuffing box should be monitored for temperature. It is normal for some boxes to run warmer than others—as much as 50°F (30°C) above the pumping temperature. Only if this exceeds the maximum temperature rating of the packing are steps required to reduce box temperature.

The best lubricant for packing in most services has been found to be steam cylinder oil.

The concepts that a higher discharge pressure requires more rings of packing and that a larger number of rings lasts longer are true for long-stroke, low-speed machines, but have been disproven in a number of moderate- to high-speed power-pump applications. Unless they are profusely lubricated, the larger number of rings create additional frictional heat and wipe lubricant from the plunger surface, thus depriving some rings of lubrication. On numerous saltwater injection pumps operating at pressures above 4,000 lb/in² (28 MPa), Chesney reported that packing life was only two weeks with twelve rings of packing in each stuffing box. With three rings in each box, packing life was approximately 6 months.

Plungers Next to packing, the plunger is the component of a power pump that typically requires most frequent replacement. The high speed of the plunger and the friction load of the packing tend to wear the plunger surface. For longer life, plungers are sometimes hardened, although it is more common to apply a hard coating. Such coatings are of chrome, various ceramics, nickel-based alloys, or cobalt-based alloys. Desired characteristics of the coatings include hardness, smoothness, high bond strength, low porosity, corrosion resistance, and low cost. No one coating optimizes all characteristics.

Ceramic coatings are harder than the metals, but are brittle, porous, and sometimes lower in bond strength. Porosity contributes to shorter packing life. Mixing of hard particles, such as tungsten carbide, into the less-hard nickel or cobalt alloys has resulted in longer plunger life at the expense of shorter packing life.

Stuffing Boxes Various stuffing box designs (various lubrication and bleed-off arrangements to minimize leakage and extend packing life) are shown in Fig. 14.1.19.

The most significant advance in packing arrangements in recent years has been spring loading. Spring loading is applied most to V-ring (chevron) packing but also works well with square packing. The spring must be located on the pressure side of the packing. Springs of various types can be used, including single-coil, multiple coil, wave-washer, Belleville, and thick-rubber-washer. The force provided by the spring is small compared to the force imposed on the packing by discharge pressure. The major functions of the spring are to provide a small preload to help set the packing, and to hold all bushings and packing in place during operation.

Spring loading of packing has the following advantages:

Requires no adjustment of the gland. The gland is tightened until it bottoms, then is locked. This removes one of the biggest variables in packing life: personnel skill.

Allows expansion. If the packing expands during the initial break-in, the spring allows for the expansion.

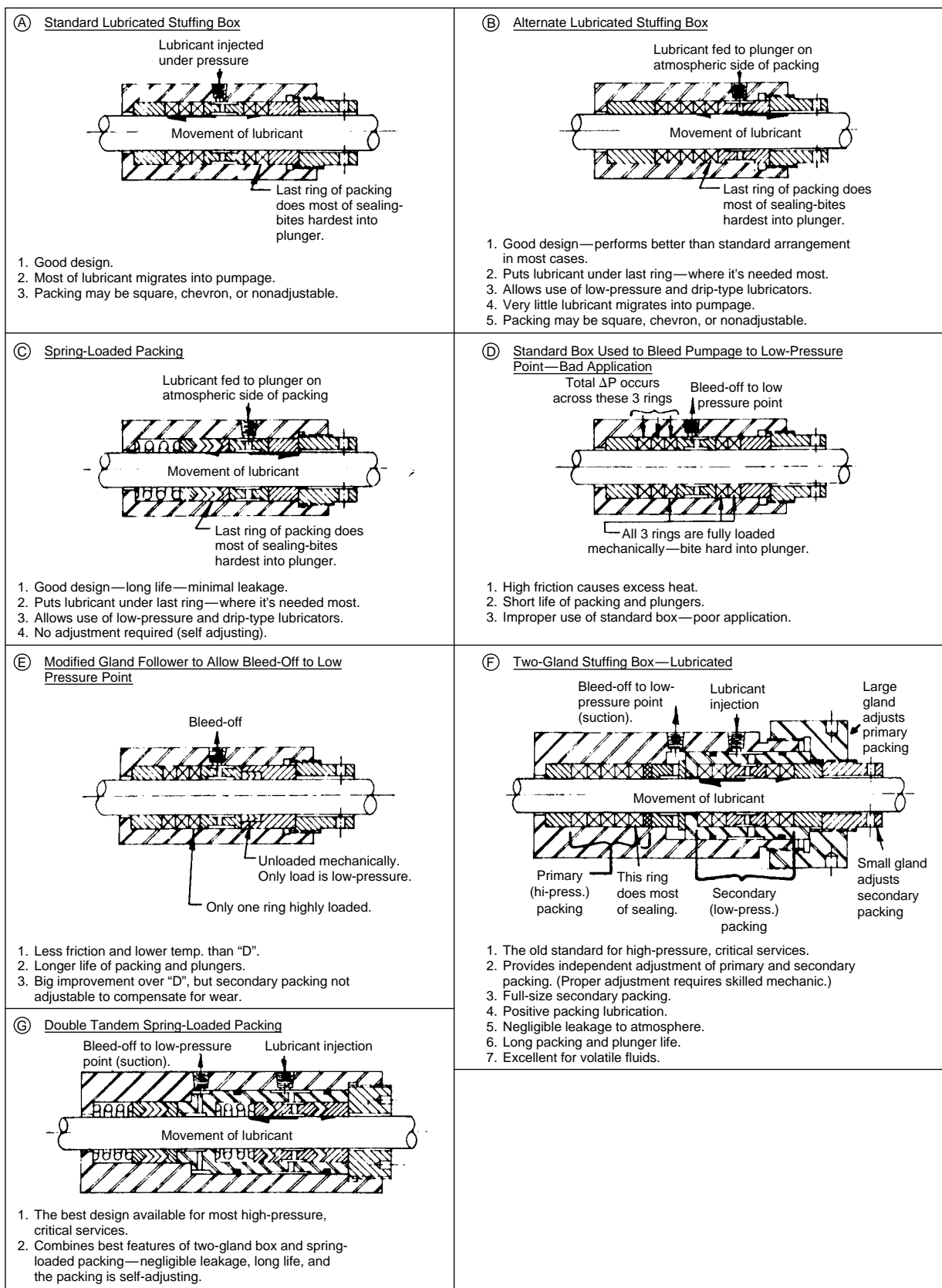


Fig. 14.1.19 Various stuffing box designs for reciprocating pumps.

Adjusts for wear. As the packing wears, adjustment automatically occurs from inside the box. The problem of transmitting gland movement through the top packing ring is eliminated.

Provides a cavity. The spring cavity provides an annular space for the injection of a clean liquid for slurry applications.

Eliminates the need for a gland, if allowed by pump design. The stuffing box assembly (if a separate component) can be disassembled and reassembled on a workbench.

Disadvantages of spring-loading packing are associated with the cavity created by the spring. Since this cavity communicates directly with the pumping chamber, the additional clearance volume will cause a reduction in volumetric efficiency if the pumpage is sufficiently compressible. This cavity also provides a place for vapors to accumulate. If the pump design does not provide for venting this space, a further reduction in volumetric efficiency may occur.

Spring-loaded packing is the reciprocating pump's equivalent to the mechanical seal for rotating shafts. Leakage is low, life is extended, and adjustments are eliminated. Packing sets can be stacked in tandem (they must be independently supported) for a stepped pressure reduction, or to capture leakage from the primary packing that should not escape to the environment.

Controlled-Volume Pumps A controlled-volume pump (also called a **metering, proportioning, or chemical-injection pump**) is a power pump with an adjustable stroke length (or adjustable *effective* stroke length). It is used to provide an accurate, and adjustable, capacity. The capacity can be changed manually, as shown in Fig. 14.1.20, or with an electric, pneumatic, or hydraulic controller.

Such pumps can develop pressures to 30,000 lb/in² (200 MPa) with capacity controlled with ± 1 percent. The piston or plunger may come in direct contact with the pumpage, or may be separated from the pumpage by a diaphragm. The diaphragm may be flat, tubular, or conical. It can be actuated mechanically, as shown in Fig. 14.1.20, or hydraulically, as shown in Fig. 14.1.21. Mechanically actuated diaphragms are generally limited to capacities of 25 gal/h (100 L/h) and pressures of 250 lb/in² (1,700 kPa).

Direct-Acting Pumps

The direct-acting pump (see Fig. 14.1.4) has some of the same advantages as the power pump, plus others. These units are also well-suited for high-pressure, low-flow applications. Discharge pressures normally range from 100 to 5,000 lb/in² (0.7 to 35 MPa), but may exceed 10,000 lb/in² (70 MPa). Capacity is proportional to speed from stall to maximum speed, affected little by discharge pressure. Speed is controlled by controlling the motive fluid. The unit is normally self-priming, particularly the low-clearance-volume type.

Direct-acting pumps are negligibly affected by hostile environments,

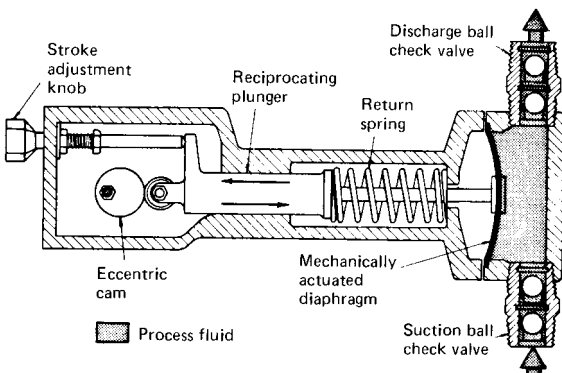


Fig. 14.1.20 Controlled-volume pump with mechanically actuated diaphragm and manual-adjustment stroke length. (Reprinted from James P. Poynton, "Metering Pumps: Selection and Application," Marcel Dekker, Inc., Copyright 1983. Used by permission.)

such as corrosive fumes, because of the absence of a bearing housing, crankcase, or oil reservoir (except for units requiring lubricators). Some direct-acting pumps, inundated by floodwater, have continued to operate without adverse effects. The direct-acting pump is quiet, simple to maintain, and its low speed often results in a long life.

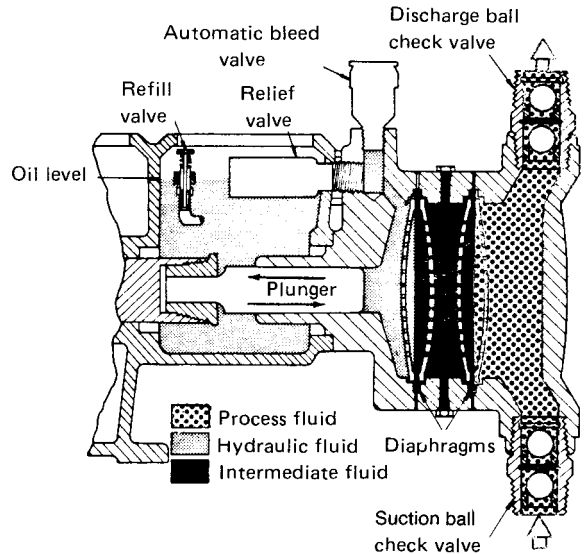


Fig. 14.1.21 Controlled-volume pump with hydraulically actuated double diaphragms. (Reprinted from James P. Poynton, "Metering Pumps: Selection and Application," Marcel Dekker, Inc., Copyright 1983. Used by permission.)

Since the direct-acting pump was originally designed to be driven by steam (initially, most pumps were), it was known as a **steam pump**—not for the pumped fluid, but for the motive fluid. Other fluids are now also used to drive direct-acting pumps. Fuel gas, which would otherwise be throttled through a pressure regulator for plant use, is often piped through a direct-acting pump to provide "free" pumping. Compressed air is frequently used to drive small pumps for services such as hydrostatic testing and chemical metering. Hydraulic oil is used to drive **intensifiers** for high-pressure water-jetting.

Direct-acting air-driven diaphragm pumps handle a variety of liquids at pressures up to the pressure of the driving air. This pump, as shown in Fig. 14.1.22, has no stuffing box or packing, and therefore has no leakage, unless a diaphragm fails. Failure of a diaphragm may result in pumpage in the exhaust air and/or air in the pumpage.

Direct-acting pumps operate at speeds that normally range from 0 to 50 cycles/min, depending on the stroke length. The ability of these units to operate at speeds down to zero (i.e., stall conditions) is desirable for some applications.

The direct-acting pump has a low thermal efficiency when driven by a gas (such as steam or air). The mechanical efficiency (output force divided by input force) is high; but because the unit has no device to store energy (such as a flywheel), the motive gas must remain at full inlet pressure in the drive cylinder through the entire stroke. At the end of the stroke, the gas expands to exhaust pressure, but no work is performed during this expansion. Hence, the thermal energy of the gas is lost to friction. Steam pumps consume about 100 lb/h of steam for each hydraulic horsepower (hhp) (60 kg/h · kW) developed on the liquid end. When natural gas or air is the motive fluid, consumption is about 3,500 std. ft³/h · hhp (130 m³/h · kW).

The **drive end** (or steam end, or gas end) of a direct-acting pump converts the differential pressure of the motive fluid to reciprocating motion for the liquid end. It is similar in construction to the liquid end, containing a piston (or diaphragm) and valving. The major difference is that the valve is mechanically actuated by a control system which senses

the location of the drive piston, causing the valve to reverse the flow of the motive fluid when the drive piston reaches the end of its stroke.

The main component of the drive end is the drive cylinder. This cylinder forms the major portion of the pressure boundary, and supports the other drive end parts.

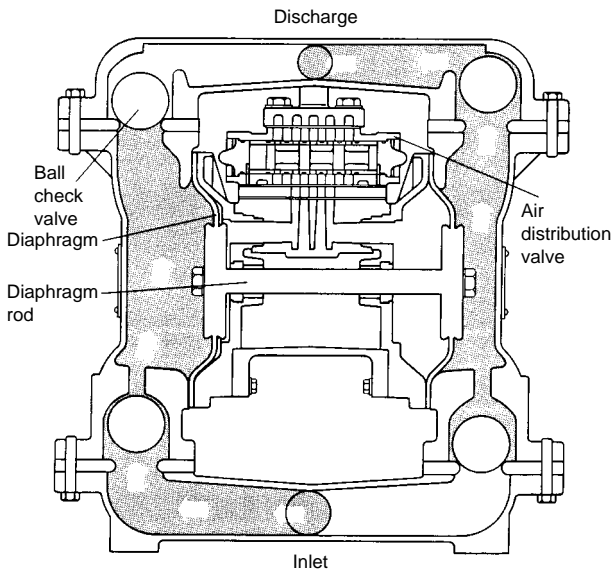


Fig. 14.1.22 Direct-acting, air-driven diaphragm pump. (Used by permission of Warren Rupp, Inc., a unit of IDEX Corp., Mansfield, OH.)

Mechanical Efficiency—Direct-Acting Pumps For a direct-acting pump, mechanical efficiency is the ratio of the force transmitted to the liquid by the piston (or plunger or diaphragm) to the force transmitted to the drive piston (or diaphragm) by the motive fluid. For double-acting

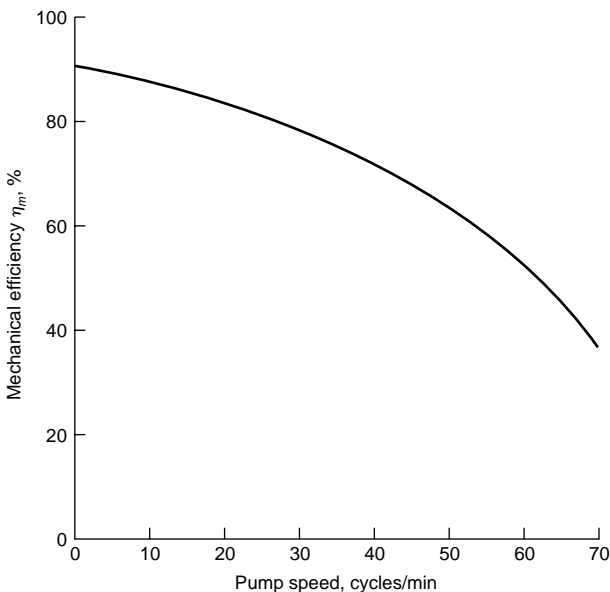


Fig. 14.1.23 Mechanical efficiency of a direct-acting pump.

pumps, the differential pressures are used at both ends of the pump, and since fluid friction losses in valves and ports must be charged to the pump, the pressures are as measured at the inlet and outlet ports. In equation form:

$$\eta_m = \frac{A_L \Delta p_L}{A_{dr} \Delta p_{dr}} \quad (14.1.1)$$

where η_m = mechanical efficiency of pump, A_L = area of liquid piston or plunger, Δp_L = differential pressure across liquid end, A_{dr} = area of drive piston, and Δp_{dr} = differential pressure across drive end.

The area of the piston rod is usually small relative to the piston area, and is often ignored. However, it must be considered when the rod area becomes a significant portion of the piston area.

The mechanical efficiency of a typical direct-acting pump is shown in Fig. 14.1.23. Note that η_m rises as speed is reduced. It is this characteristic that allows the unit to be controlled by throttling the motive fluid. The reduction in the available differential driving pressure forces the pump to operate more efficiently, i.e., at a lower speed.

ROTARY PUMPS

Rotary pumps are displacement pumps which have rotating pumping elements, such as gears, lobes, screws, vanes, or rollers. They do not contain inlet and outlet check valves, as do reciprocating pumps. They are built for capacities from a fraction of a gallon per minute (cubic meter per hour) (for domestic oil burners) to about 10,000 gal/min (2000 m³/h) (for marine cargo service). Though used for pressures up to 5,000 lb/in² (35 MPa), their particular field is for pressures of 25 to 500 lb/in² (170 to 3,500 kPa).

Most rotary pumps rely on close running clearances to prevent the pumpage from leaking from the discharge side back to the suction side of the pump. Because of the close clearances, the pumpage must be clean. Most rotary pumps are noted for their ability to handle viscous liquids, and many actually *require* a viscous liquid to achieve peak performance. Viscosity affects the mechanical efficiency, volumetric efficiency, and NPSHR of a rotary pump.

The **rotor** is the pumping element of the rotary pump, and is usually the feature by which the pump is classified.

Gears The teeth of the gear trap and displace the pumpage. If the teeth are on the outside of the gear, as shown in Fig. 14.1.24, it is called an **external gear**. As the teeth disengage, suction pressure pushes liquid into the cavities between the teeth. The teeth then carry the liquid around to the discharge side of the pump. As the teeth engage, the fluid

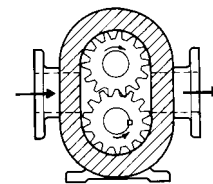


Fig. 14.1.24 External-gear rotary pump. (Reprinted from "Hydraulic Institute Standards for Centrifugal, Rotary and Reciprocating Pumps," 14th ed., copyright 1983, with the permission of Hydraulic Institute.)

is pushed into the discharge line. In the most common type of gear pump, one of the pumping gears drives the other pumping gear. This direct contact between the teeth requires the pumpage to be a good lubricant. In some units, though, the two pumping gears are driven by gears (called **timing gears**) mounted on the two shafts external to the pumpage. With this arrangement, the two pumping gears do not contact each other, and the unit is more suitable for liquids with low lubricity. This arrangement does, however, require a larger, more complex pump.

If the teeth are on the inside of a ring, as shown in Fig. 14.1.25, it is called an **internal gear**. In this unit, the larger internal-tooth gear drives the smaller external-tooth gear. Both gears transfer liquid from the suction to the discharge.

Both external- and internal-gear rotary pumps are used in lubrication systems of engines, compressors, and larger pumps.

Lobe pumps are similar in construction and pumping action to external gear pumps, but one lobe does not drive the other. They do not even

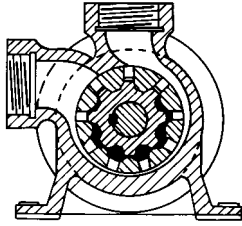


Fig. 14.1.25 Internal-gear rotary pump.

touch each other, and therefore must have their shafts independently driven by external timing gears. The lobes are often made of elastomers, and operate at low speeds. These units are used to transfer delicate items

such as cherries and other foods—and even live fish. Figure 14.1.26 shows a rotary pump with three-lobe rotors.

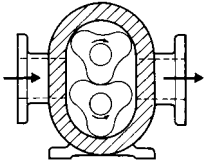


Fig. 14.1.26 Three-lobe rotary pump. (Reprinted from "Hydraulic Institute Standards for Centrifugal Rotary and Reciprocating Pumps," 14th ed., copyright 1983, with the permission of Hydraulic Institute.)

Screw pumps are constructed with one, two, or three screws. The single-screw pump is more commonly called a progressing-cavity pump, and is discussed separately below. The two-screw pump, as illustrated by Fig. 14.1.27, is used to handle liquids with viscosities to greater than 1,000,000 SSU (200,000 mm²/s), with capacities to about 10,000 gal/min (2000 m³/h), pressures to 2500 lb/in² (17 MPa), and temperatures to 600°F (300°C). It is particularly suited for high-viscosity liquids.

The three-screw rotary pump, as illustrated in Fig. 14.1.28, is a high-speed pump (up to 6,000 r/min) used primarily for lubrication systems on turbines, compressors, and centrifugal pumps.

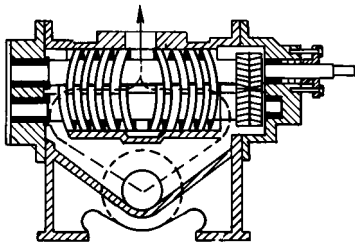


Fig. 14.1.27 Two-screw rotary pump.

A **progressing-cavity pump** is illustrated in Fig. 14.1.29. The rotor is made of polished steel, and in the shape of a single-lead screw. The stator, usually made of elastomer, is in the shape of a double-lead screw. As the rotor turns, its axis prescribes a circle. This characteristic requires the rotor to be driven by an internal universal joint (or equivalent). The pumpage moves through the pump in a spiral. The rubber stator makes this unit well-suited for abrasive services.

Figure 14.1.30 illustrates a **sliding vane** rotary pump. The single rotor contains multiple vanes which slide in radial slots. The rotor and casing are eccentric. The vanes maintain contact with the casing by centrifugal force and pressure. These units are typically available for capacities to 1000 gal/min (200 m³/h), viscosities to 500,000 SSU (100,000 mm²/s), and pressures to 125 lb/in² (900 kPa). Some sliding-vane pumps are suitable for low-lubricity liquids such as light hydrocarbons.

Flexible Member Some rotary pumps are built with flexible vanes, liners, and tubes. The flexible tube pump, also called a **peristaltic pump**, is shown in Fig. 14.1.31.

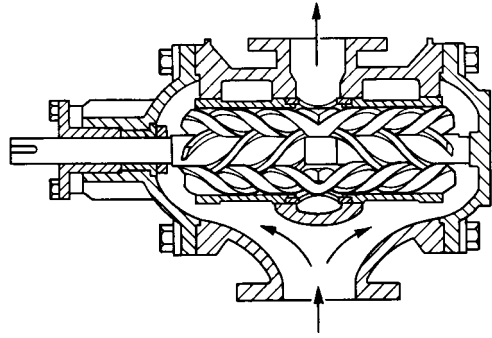


Fig. 14.1.28 Three-screw rotary pump.

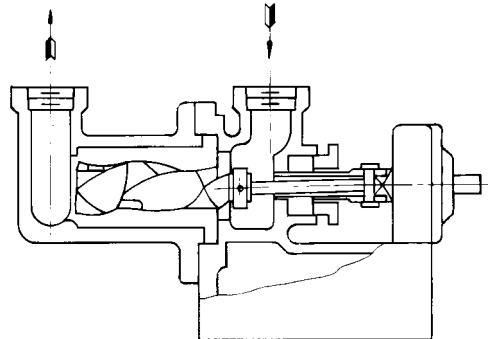


Fig. 14.1.29 Progressing-cavity (single-screw) rotary pump. (Hydraulic Institute.)

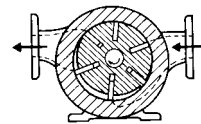


Fig. 14.1.30 Sliding-vane rotary pump. (Reprinted from "Hydraulic Institute Standards for Centrifugal, Rotary and Reciprocating Pumps," 14th ed., copyright 1983, with the permission of Hydraulic Institute.)

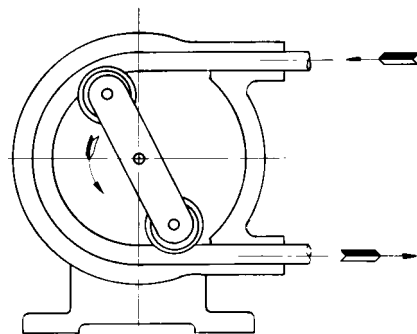


Fig. 14.1.31 Flexible-tube (peristaltic) rotary pump. (Hydraulic Institute.)

A **radial-plunger** rotary pump is shown in Fig. 14.1.32. An **axial-plunger** rotary pump (also called an **axial-piston** or **swash-plate** pump) is shown in Fig. 14.1.33. In these pumps, the inner body rotates, causing each plunger to alternately accept pumpage from the inlet and deliver it

to the discharge port. In the radial pump, the length of stroke 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

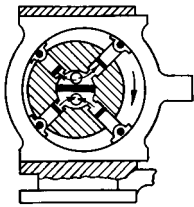


Fig. 14.1.32 Radial-plunger rotary pump.

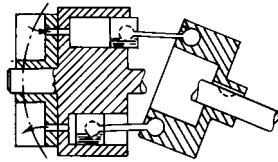


Fig. 14.1.33 Axial-plunger (swash-plate) rotary pump.

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.

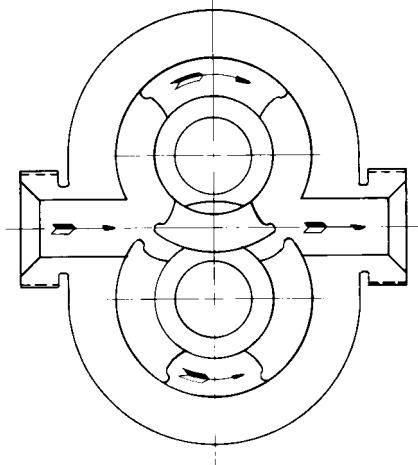


Fig. 14.1.34 Circumferential-piston rotary pump. (Hydraulic Institute.)

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. 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. The definition is not as precise as the reciprocating pump 3 percent capacity drop.

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.

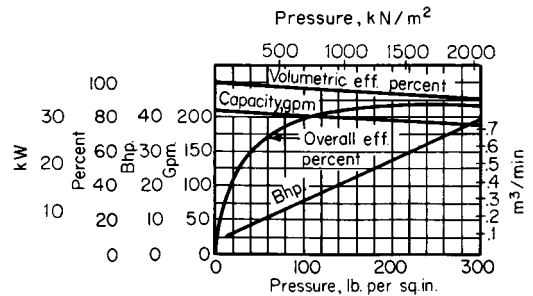


Fig. 14.1.35 Rotary pump performance curves for a herringbone gear pump at 600 r/min with a 400-SSU (90-mm²/s) liquid.

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:

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

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 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.

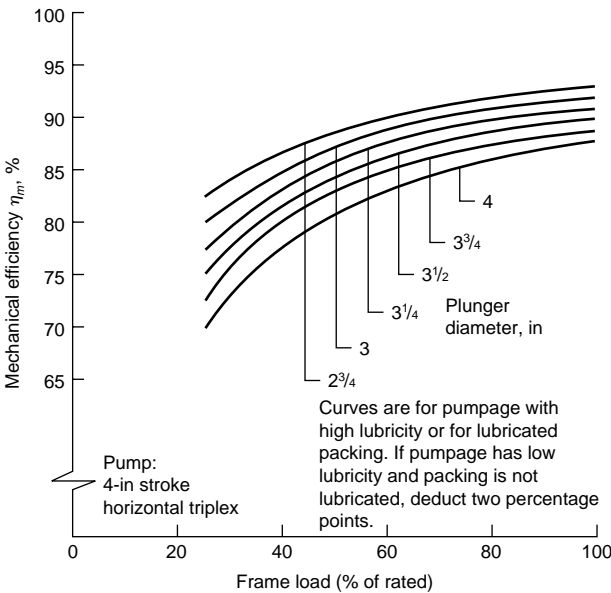


Fig. 14.1.36 Mechanical (overall) efficiency of a triplex power pump.

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.*

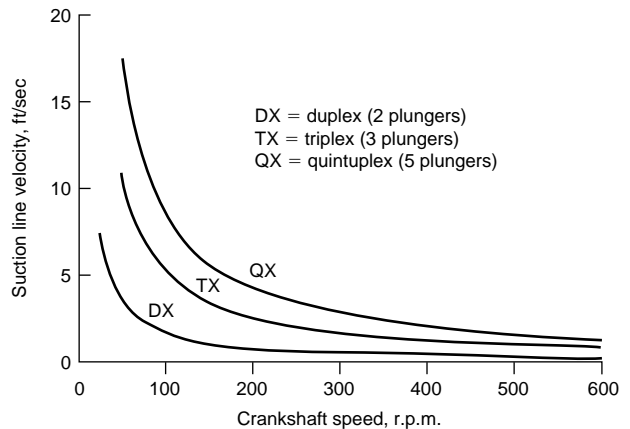


Fig. 14.1.37 Suction line velocities for constant acceleration head, for reciprocating power pumps.

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.

Contain a minimum number of turns, using long-radius elbows and laterals where practical.

Include a suitable pulsation dampener (or provisions for adding one) adjacent to the pump liquid end.*

Contain a relief valve, sized to pass full pump capacity at a pressure that does not exceed 110 percent of cracking pressure (the opening pressure of the relief valve). The discharge from the relief valve should be piped back to the suction vessel so that gases liberated through the valve are not fed back into the pump.

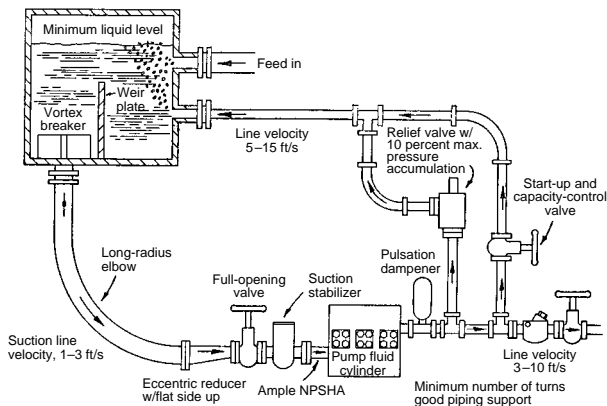


Fig. 14.1.38 A good system for a reciprocating pump.

Contain a bypass line and valve so that the pump may be started against negligible discharge pressure.

Contain a check valve to prevent the imposition of system pressure on the pump during start-up.

The features of a good system are illustrated in Fig. 14.1.38.

Remedies for Low NPSHA/High NPSHR

In designing the suction system for a displacement pump, if NPSHA is found to be less than NPSHR, a remedy may be found in the following list:

Increase the diameter of the suction line.

Reduce the length of the suction line by providing a more direct route, or by moving the pump closer to the suction vessel.

Install a suction bottle or stabilizer adjacent to the pump liquid end. A fabricated bottle has often been successfully used at pressures below about 50 lb/in² (300 kPa), but maintenance of a liquid level is required.

A section of rubber hose in the suction line, adjacent to the pump, will often reduce the acceleration head.

Elevate the suction vessel or the level of liquid in the suction vessel.

Reduce temperature of pumpage. (If the pumpage is at bubble point in the vessel, it must be cooled after it leaves the vessel.)

Reduce speed of the pump, or select a larger pump running slower.

At a lower speed, it may be possible to operate a reciprocating pump with light suction valve springs, or none at all.)

If the above steps are insufficient, impractical, or impossible, a booster pump must be provided. A booster for a displacement pump is normally a centrifugal, although direct-acting reciprocating and rotary pumps are sometimes used. The NPSHR of the booster must be less than that available from the system. The head of the booster should exceed the main pump NPSHR plus suction-line losses by about 20 percent. The booster should be installed near the suction vessel, and a pulsation bottle or stabilizer should be installed in the suction line, adjacent to the main pump, to protect the booster from pulsating flow.

14.2 CENTRIFUGAL AND AXIAL-FLOW PUMPS

by Igor J. Karassik

REFERENCES: (1) Theory: Stepanoff, "Centrifugal and Axial Flow Pumps," Wiley. Wislicenus, "Fluid Mechanics of Turbomachinery," McGraw-Hill. Pfeleiderer, "Die Kreiselpumpen," Springer. Spannake, "Centrifugal Pumps, Turbines and Propellers," Technology Press. (2) Practice: Hicks, "Pump Selection and Application," and "Pump Operation and Maintenance," McGraw-Hill. Standards of the Hydraulic Institute. Karassik, "Centrifugal Pump Clinic," Marcel Dekker, Inc.

NOMENCLATURE AND MECHANICAL DESIGN

Pumps covered in this section fall into three general classes: (1) centrifugal or radial-flow, (2) mixed-flow, and (3) axial-flow or propeller pumps. The essential elements of a centrifugal pump are (1) the rotating element, consisting of the shaft and the impeller, and (2) the stationary element, consisting of the casing, stuffing boxes, and bearings (see Fig. 14.2.1). Other parts, such as wearing rings and shaft sleeves, are generally added to produce better-operating, more economical machines as warranted by the various services on which the pumps are to be used. Names recommended by the Hydraulic Institute for various parts are given in Table 14.2.1.

In a centrifugal pump the liquid is forced, by atmospheric or other pressure, into a set of rotating vanes which constitute an impeller discharging the liquid at a higher pressure and a higher velocity at its

periphery. The major portion of the velocity energy is then converted into pressure energy by means of a volute (Fig. 14.2.2) or by a set of stationary diffusion vanes (Fig. 14.2.3) surrounding the impeller periph-

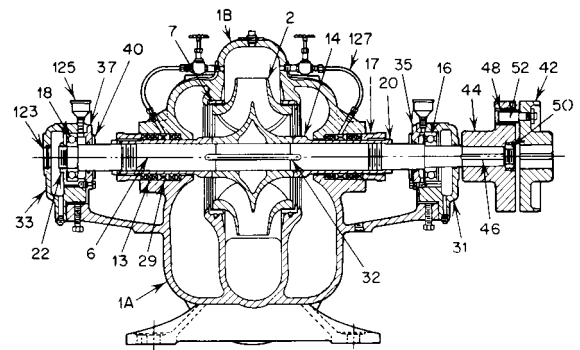


Fig. 14.2.1 Horizontal single-stage double-suction volute pump. (Numbers refer to parts listed in Table 14.2.1.)

Table 14.2.1 Recommended Names of Centrifugal-Pump Parts

(These parts are called out in Figs. 14.2.1, 14.2.4, and 14.2.5)

Item no.	Name of part	Item no.	Name of part
1	Casing	33	Bearing housing (outboard)
1A	Casing (lower half)	35	Bearing cover (inboard)
1B	Casing (upper half)	36	Propeller key
2	Impeller	37	Bearing cover (outboard)
4	Propeller	39	Bearing bushing
6	Pump shaft	40	Deflector
7	Casing ring	42	Coupling (driver half)
8	Impeller ring	44	Coupling (pump half)
9	Suction cover	46	Coupling key
11	Stuffing-box cover	48	Coupling bushing
13	Packing	50	Coupling lock nut
14	Shaft sleeve	52	Coupling pin
15	Discharge bowl	59	Handhole cover
16	Bearing (inboard)	68	Shaft collar
17	Gland	72	Thrust collar
18	Bearing (outboard)	78	Bearing spacer
19	Frame	85	Shaft-enclosing tube
20	Shaft-sleeve nut	89	Seal
22	Bearing lock nut	91	Suction bowl
24	Impeller nut	101	Column pipe
25	Suction-head ring	103	Connector bearing
27	Stuffing-box-cover ring	123	Bearing end cover
29	Lantern ring	125	Grease (oil) cup
31	Bearing housing (inboard)	127	Seal piping (tubing)
32	Impeller key		

ery. Pumps with volute casings are called **volute pumps**; those with diffusion vanes are called **diffuser pumps**. The latter were once commonly called turbine pumps, but this term has recently been more selectively applied to vertical deep-well centrifugal diffuser pumps, now called **vertical turbine pumps**.

Centrifugal pumps are divided into other categories, several of which relate to the impeller. First, impellers are classified according to the major direction of flow in reference to the axis of rotation. Centrifugal pumps may have (1) **radial-flow** impellers (Figs. 14.2.1, 14.2.2, and 14.2.5), (2) **axial-flow** impellers (Fig. 14.2.4), and (3) **mixed-flow** impellers, which combine radial- and axial-flow principles (Fig. 14.2.7).

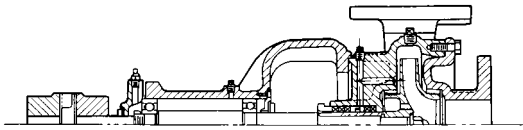


Fig. 14.2.2 Single-stage end-suction volute pump.

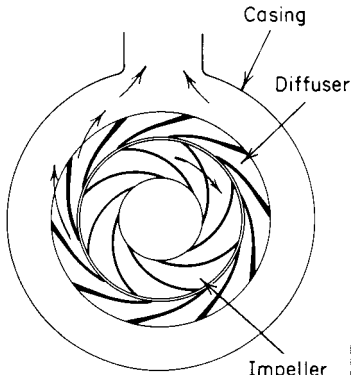


Fig. 14.2.3 Diffuser-type pump.

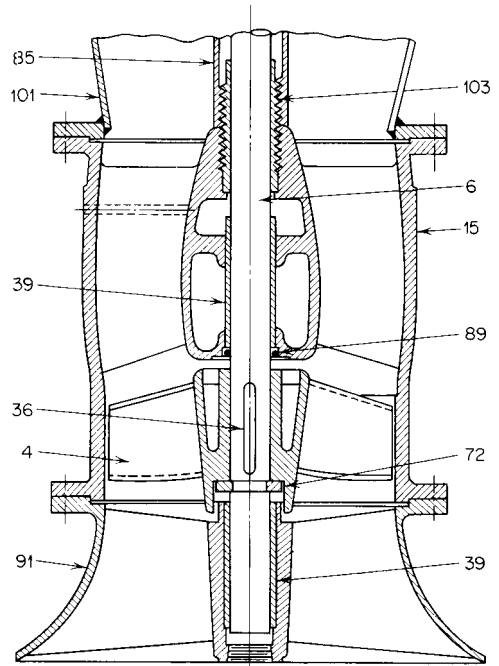


Fig. 14.2.4 Vertical wet-pit diffuser pump bowl. (Numbers refer to parts listed in Table 14.2.1.)

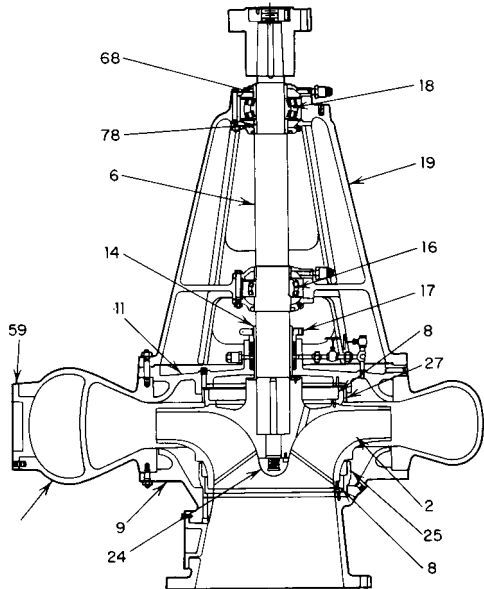


Fig. 14.2.5 Vertical end-suction pump with a double-volute casing. (Numbers refer to parts listed in Table 14.2.1.)

Impellers are further classified according to the flow arrangement into (1) **single-suction**, with a single inlet on one side, and (2) **double-suction**, with water flowing to the impeller symmetrically from both sides. They are categorized according to their mechanical construction into (1) **closed**, with shrouds or sidewalls enclosing the waterways, (2) **open**, with no shrouds, and (3) **semiopen** or semiclosed.

If the pump is one in which the head is developed by a single impeller, it is called a **single-stage pump**. When two or more impellers

operating in series are used, the unit is called a **multistage pump**. The mechanical design of the casing provides the added classification of **axially split** (Fig. 14.2.1) or **radially split** (Figs. 14.2.2 and 14.2.5), and the axis of rotation determines whether the pump is horizontal-shaft, vertical-shaft, or (occasionally) inclined-shaft. Usually these are referred to simply as **horizontal** or **vertical** units.

Horizontal centrifugal pumps are classified still further according to suction-nozzle location into (1) **end-suction**, (2) **side-suction**, (3) **bottom-suction**, and (4) **top-suction**.

Some pumps operate with the liquid conducted to and from the unit by piping. Other pumps, usually vertical types, are submerged in their suction supply. Vertical pumps are therefore called either **dry-pit** or **wet-pit** types. If the wet-pit pumps are axial-flow, mixed-flow, or vertical-turbine types, the liquid is discharged up through the supporting drop or column pipe to a discharge point either above or below the supporting floor. These pumps are consequently designated as **above-ground discharge** or **below-ground discharge** units.

Casings

The pressure acting on the impeller in a single-volute pump-casing design is nearly uniform when the pump is operated at or near its design capacity. At other capacities, the pressures around the impeller are not uniform, causing a radial reaction (or radial thrust) which can substantially increase the pump-shaft deflection. When it becomes impractical to counteract this radial thrust through the use of a heavier shaft and heavier bearings, a **double-** or **twin-volute** design (Fig. 14.2.5) may be used.

End-suction single-stage pumps are made of one-piece solid casings. At least one side of the casing must have an opening with a cover so that the impeller can be assembled in the pump. If the cover is on the suction side, it becomes the casing sidewall and contains the suction opening (Fig. 14.2.2). This is called the **suction cover** or **casing suction head**. Other designs are made with stuffing-box covers (Fig. 14.2.6), and still others have both casing suction covers and stuffing-box covers (Figs. 14.2.5 and 14.2.6).

In the inexpensive open-impeller pump, the impeller rotates within close clearance of the pump casing (Fig. 14.2.6). If the intended service is more severe, a side plate is mounted within the casing to provide a renewable close-clearance guide to the liquid flowing through the open impeller.

The discharge nozzle of end-suction single-stage horizontal pumps is usually in a top-vertical position (Fig. 14.2.2). However, other nozzle positions may be obtained, such as top-horizontal, bottom-horizontal, or bottom-vertical discharge. Practically all double-suction axially split casing pumps have a side-discharge nozzle and either a side- or a bottom-suction nozzle. Single-stage bottom-suction pumps are rarely made in sizes below 10 in discharge-nozzle diameter.

Both axially split (Fig. 14.2.8) and radially split (Fig. 14.2.9) casings are used for multistage pumps. The choice between the two designs is

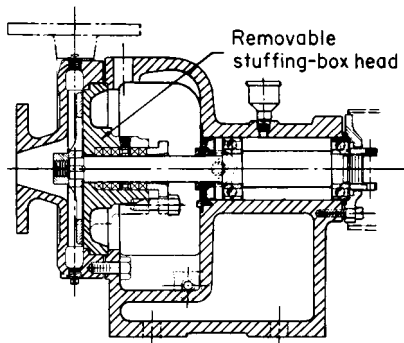


Fig. 14.2.6 End-suction pump with removable stuffing-box head.

dictated by the discharge pressure, with 1,300 to 2,000 lb/in² (90 to 140 kg/cm²) forming the approximate limit between the two. Radially split casings are normally designed as double casings; the working parts of the pump are enclosed in an inner casing, which is then inserted into a second, or outer, casing. The space between the two casings is maintained at the discharge pressure of the last pump stage.

Impellers and Wearing Rings

In addition to being classified with reference to the suction flow into the impeller, the basic flow component, and their mechanical features, impellers are also classified with reference to their profile and to their head-capacity characteristics at a given speed. This last relationship will be described later, in the discussion of **specific speed**.

Many impellers are designed for specific applications. Special non-clogging impellers with blunt edges and large waterways are used for sewage which ordinarily contains rags or stringy material. Impellers designed for handling paper-pulp stock are fully open and nonclogging and have screw conveyor vanes which project into the suction nozzle.

Wearing rings provide an easily and economically renewable leakage joint between the impeller and casing. A leakage joint without renewable parts is used only in very small, inexpensive pumps. The stationary ring is called (1) **casing ring** if mounted in the casing, (2) **suction-cover ring** or **suction-head ring** if mounted in a suction cover or head, and (3) **stuffing-box cover ring** if mounted in the stuffing-box cover. A renewable part for the impeller wearing surface is called the **impeller ring**. Pumps with both stationary and rotating rings are said to have **double-ring** construction.

There are various types of wearing-ring designs, and selection of the most desirable type depends on the liquid being handled, the pressure differential across the leakage joint, the surface speed, and the particular pump design. In general, centrifugal-pump designers use the ring construction they have found suitable for each particular pump service. The most common ring constructions are the **flat type** (Fig. 14.2.2) and the **L type** (Figs. 14.2.1 and 14.2.7).

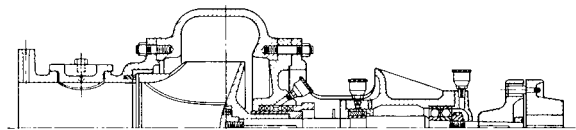


Fig. 14.2.7 End-suction pump with removable suction and stuffing-box heads.

Axial Thrust in Single-Stage and Multistage Pumps

Axial hydraulic thrust is the summation of unbalanced impeller forces acting in the axial direction. Theoretically, a double-suction impeller is in hydraulic balance, with the pressures on one side equal to and counterbalancing the pressures on the other. In practice, some slight unbalance may exist, and even double-suction pumps are provided with thrust bearings.

The single-suction radial-flow impeller is subject to axial thrust because a portion of the front wall is exposed to suction pressure, with a greater back-wall surface subject to discharge pressure. In addition, an overhung single-suction impeller with a single stuffing box is subject to an axial force equivalent to the product of the shaft area through the stuffing box and the difference between suction and atmospheric pressure. This force acts toward the impeller suction when the suction pressure is less than the atmospheric and in the opposite direction when it is higher than the atmospheric.

To eliminate the axial thrust of a single-suction impeller, a pump can be provided with both front and back wearing rings (Figs. 14.2.2 and 14.2.5). Pressure approximately equal to the suction pressure is maintained in a chamber located on the inner side of the back wearing ring by providing so-called **balancing holes** through the impeller. Leakage past the back wearing ring is returned into the suction area through these holes. In large pumps, a piped connection usually replaces the balancing holes.

Most multistage pumps are built with single-suction impellers. To balance the axial thrust of these impellers, two arrangements are used: (1) The impellers all face in the same direction and are mounted in the ascending order of the stages. The axial thrust is balanced by a hydraulic balancing device (Figs. 14.2.8 and 14.2.9). (2) An even number of single-suction impellers is used, one half of these facing in a direction opposite to the second half (Fig. 14.2.10). This mounting of single-suction impellers back to back is frequently called **opposed impellers**.

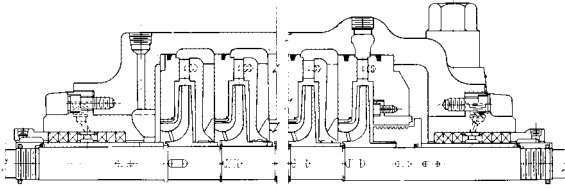


Fig. 14.2.8 Multistage pump with single-suction impellers facing in one direction and with hydraulic balancing device.

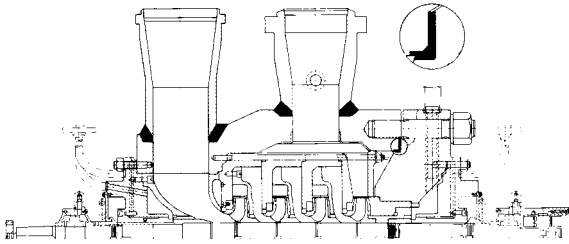


Fig. 14.2.9 Double-casing multistage pump with radially split inner casing.

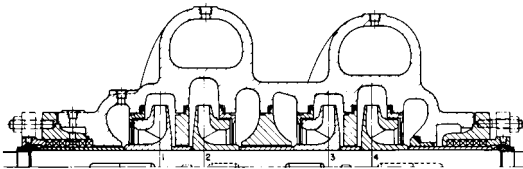


Fig. 14.2.10 Four-stage pump with opposed impellers.

Hydraulic balancing devices may take the form of (1) a **balancing drum**, (2) a **balancing disk**, or (3) a combination of these two. The **balancing drum** is illustrated in Fig. 14.2.11. The balancing chamber at the back of the last-stage impeller is separated from the pump interior by a drum mounted on the shaft. The drum is separated by a small radial clearance from the stationary portion of the balancing device, called the **balancing**

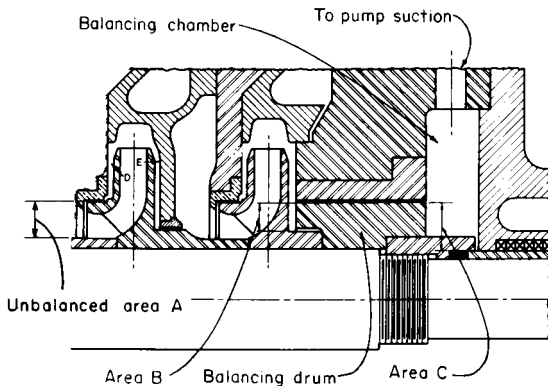


Fig. 14.2.11 Balancing drum.

drum head, which is fixed to the pump casing. The balancing chamber is connected either to the pump suction or to the vessel from which the pump takes its suction. The forces acting on the balancing drum are (1) toward the discharge end—the discharge pressure multiplied by the front balancing area (area B) of the drum; (2) toward the suction end—the back pressure in the balancing chamber multiplied by the back balancing area (area C) of the drum. The first force is greater than the second, thereby counterbalancing the axial thrust exerted upon the single-suction impellers. The drum diameter can be selected to balance the axial thrust completely or to balance 90 to 95 percent of this thrust, depending on whether a slight thrust load in a specific direction on the thrust bearing is desirable.

The operation of the simple **balancing disk** is illustrated in Fig. 14.2.12. The rotating disk is separated from the balancing-disk head by a small axial clearance. The leakage through this clearance flows into the balancing chamber and from there either to the pump suction or to the suction vessel. The back of the balancing disk is subject to the balancing-chamber back pressure, whereas the disk face experiences a range of pressures. These vary from discharge pressure at its smallest diameter to back pressure at its periphery. The inner and outer diameters are chosen so that the difference between the total force act-

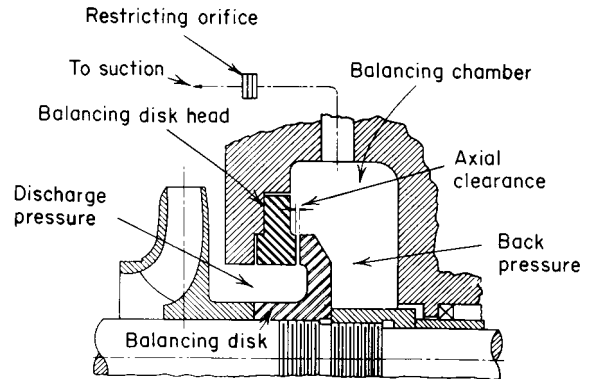


Fig. 14.2.12 Simple balancing disk.

ing on the disk face and that acting on its back will balance the impeller axial thrust. If the axial thrust of the impellers should exceed the thrust acting on the disk during operation, the latter is moved toward the disk head, reducing the axial clearance. The amount of leakage through this clearance is reduced so that the friction losses in the leakage return line are also reduced, lowering the back pressure in the balancing chamber. This automatically increases the pressure difference acting on the disk and moves it away from the disk head, increasing the clearance. Now the pressure builds up in the balancing chamber, and the disk is again moved toward the disk head until an equilibrium is reached. To ensure proper balancing-disk operation, the change in back pressure must be of an appreciable magnitude. This is accomplished by introducing a restricting orifice in the leakage return line.

The **combination disk and drum** (Fig. 14.2.9) is the most commonly used hydraulic balancing device. It incorporates portions rotating within radial clearances of stationary portions and a disk face rotating within an axial clearance of another portion of the stationary part. The radial clearance remains constant regardless of any axial displacement of the rotor within the casing. Such displacement, however, changes the axial clearance within the balancing device. These changes cause changes in the leakage, which in turn change the pressure drop across the radial clearances and thus increase or decrease the average value of the pressure acting on the disk face. These changes in the intermediate pressure on the disk face act to move the balancing device in whichever direction is required to restore equilibrium and axial balance.

Shafts and Shaft Sleeves

Pump-shaft diameters are usually larger than actually needed to transmit the torque because their size is dictated by the maximum permissible or

desirable shaft deflection. This deflection is itself chosen to prevent possible contact at the wearing surfaces while maintaining reasonable clearances that will not affect pump efficiency too unfavorably. The first **critical speed** of a shaft is related to its deflection. It follows that a shaft design permitting a deflection of, for instance, 0.005 to 0.006 in (0.13 to 0.15 mm) will have a first critical speed of 2,400 to 2,650 r/min. This is the reason for using rigid shafts (operating below their first critical speed) for pumps that operate at 1,750 r/min or lower. Multistage pumps operating at 3,600 r/min or higher use shafts of equal stiffness (for the same purpose of avoiding wearing-ring contact). However, their corresponding critical speed is about 25 to 40 percent less than their operating speed. This margin is sufficient to avoid any danger to the operation caused by critical-speed effect.

Pump shafts are usually protected from erosion, corrosion, and wear at the stuffing boxes and leakage joints and in the waterways by renewable **sleeves**. The most common shaft-sleeve function is that of protecting the shaft from wear at a stuffing box. Shaft sleeves serving other functions are given specific names to indicate their purpose. For example, a shaft sleeve used between two multistage impellers in conjunction with an interstage bushing to form an interstage leakage joint is called an **interstage** or **distance sleeve**.

Stuffing Boxes

Stuffing boxes have the primary function of protecting the pump against leakage at the point where the shaft passes out through the pump casing. If the pump handles a suction lift and the pressure at the interior stuffing-box end is below atmospheric, the stuffing-box function is to prevent air leakage into the pump. If this pressure is above atmospheric, the function is to prevent liquid leakage out of the pump. The stuffing box takes the form of a cylindrical recess that accommodates a number of rings of **packing** around the shaft or shaft sleeve. If sealing the box is desired, a **lantern ring** or **seal cage** is used to separate the rings of packing into approximately equal sections. The packing is compressed to give the desired fit on the shaft or sleeve by a gland that can be adjusted in an axial direction.

Water or some other sealing fluid can be introduced under pressure into the space provided by the seal cage, causing flow of sealing fluid in both axial directions. This is useful for pumps handling flammable or chemically active and dangerous liquids since it prevents outflow of the pumped liquid.

When a pump handles clean, cool water, stuffing-box seals are usually connected to the pump discharge or, in multistage pumps, to an intermediate stage. An **independent supply of sealing water** should be provided if any of the following conditions exist: (1) a suction lift in excess of 15 ft (4.5 m); (2) a discharge pressure under 10 lb/in² (0.7 kg/cm²); (3) hot water handled without adequate cooling (except for boiler feed pumps, in which seal cages are not used); (4) muddy, sandy, or gritty water handled; (5) for all hot-well pumps; (6) no leakage to atmosphere permitted of the liquid handled. When sealing water is taken from the pump discharge, an external connection is generally made to the seal cage through small-diameter piping (Fig. 14.2.1), or an internal-passage connection is made within the pump itself (Fig. 14.2.2).

High temperatures or pressures complicate the problem of maintaining stuffing-box packing. Pumps in these more difficult services are usually provided with jacketed, **water-cooled stuffing boxes**. If the pressure ahead of the stuffing box makes it impractical to pack the stuffing box satisfactorily, a pressure-reducing breakdown or **labyrinth** may be located ahead of the box, with the leakage past the pressure-reducing breakdown being returned to some point of lower pressure in the pumping cycle.

Basically, **stuffing-box packing** is a pressure-breakdown device that is sufficiently plastic to be adjusted for proper operation. The most common types are asbestos packing and metallic packing, the latter being composed of flexible metallic strands or foil with graphite or oil lubricant and with either an asbestos or a plastic core. Other types of packing used may be hemp, cord, braided, duck fabric, chevron, etc. Packing is supplied either in continuous coils of square cross section or in preformed, die-molded rings.

Mechanical seals are used in centrifugal pumps when it becomes im-

practical to use conventional packing with radial sealing surfaces. The sealing surfaces of a mechanical seal are located in a plane perpendicular to the pump shaft and consist of two highly polished surfaces running adjacently, one surface being connected to the shaft and the other to the stationary portion of the pump. These surfaces are held essentially in contact by a spring, the axial clearance between the surfaces being provided by a thin film of liquid. The flow of liquid may be only a drop every few minutes or even a haze of escaping vapor.

There are two basic seal arrangements: (1) internal assembly and (2) external assembly. Two mechanical seals may be mounted inside a stuffing box to make a **double seal assembly** (Fig. 14.2.13). Such an arrangement is used for pumps handling toxic or highly inflammable liquids. A clear, filtered, and generally inert sealing liquid is injected between two seals at a pressure slightly in excess of the pressure in the pump ahead of the seal.

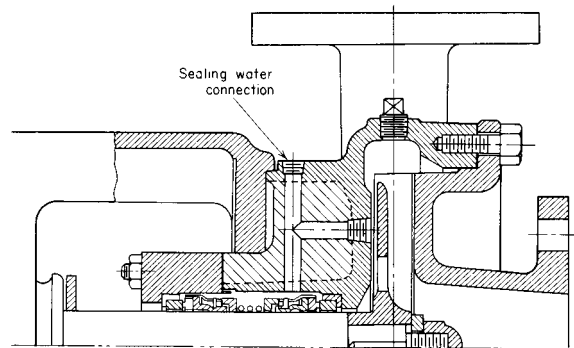


Fig. 14.2.13 Single-stage end-suction pump with double mechanical seal.

Mechanical seals can reduce the escape of toxic or flammable liquids very effectively, but lately we have seen constantly more stringent regulations by both federal and local authorities. This has led to a greater use of hermetic, or so-called **sealless** pumps. Two solutions have emerged to meet this requirement. The first is the canned-motor pump in which a portion of the pumped liquid is permitted to flow into the motor. The liquid is isolated from contact with the windings, insulation, and cores of rotor and stator by metal "cans." The second solution is the use of magnetic couplings. These consist of two sets of magnets, the outer (or driver) magnets, and the inner (or driven) magnets. A motor is connected to the outer portion of the coupling and the pump impeller is connected to the inner portion. A containment shell separates the two magnets and hermetically seals the pump proper. This latter solution is finding preference because it permits the use of standard motors, making pumps more readily maintainable in the field, while canned-motor pumps usually have to be returned to the manufacturer's plant for rebuilding.

For some power-plant services, **condensate-injection sealing** is superior to either conventional packing or mechanical seals. A labyrinth breakdown bushing is substituted for the conventional packing, and the pump-shaft sleeve runs within this bushing with a small radial clearance. Cold condensate at a pressure in excess of the internal pump pressure is introduced centrally in this breakdown bushing (Fig. 14.2.9). A small portion of the injection water flows inwardly into the pump proper; the remainder flows out into a collecting chamber vented to the atmosphere and is piped back to the condenser.

Bearings

(See also Sec. 8.)

All types of **bearings** are used in centrifugal pumps. Even the same basic design of pump is often made with two or more different bearings, required by varying service conditions. Two external bearings are usually used for the double-suction single-stage general-service pump, one on either side of the casing. In horizontal pumps with bearings on each end, the **inboard bearing** is the one between the casing and the coupling

and the **outboard bearing** is located at the opposite end. Pumps with overhung impellers have both bearings on the same side of the casing; the bearing nearest the impeller is the inboard bearing, and the one farthest away the outboard bearing.

Ball bearings are the most common antifriction bearings used on centrifugal pumps. Roller bearings are used less often, although the spherical roller bearing is used frequently for large shaft sizes. Ball bearings used in centrifugal pumps are usually grease-lubricated, although some services use oil lubrication.

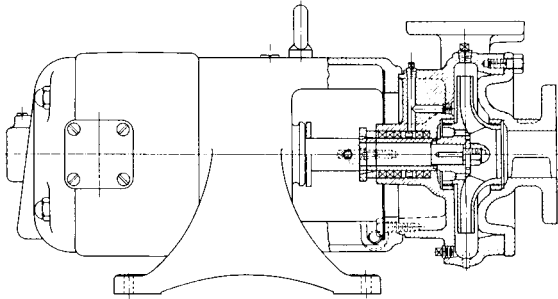


Fig. 14.2.14 Close-coupled (motor-mounted) pump.

Sleeve bearings are used for large, heavy-duty pumps with shaft diameters of such proportions that the necessary antifriction bearings are not commonly available. Another application is for high-pressure multi-stage pumps operating at speeds of 3,600 to 9,000 r/min. Still another application is in vertical submerged pumps, such as vertical turbine pumps, in which the bearings are subject to a water contact. Most sleeve bearings are oil-lubricated.

Thrust bearings used in combination with sleeve bearings are generally Kingsbury or Kingsbury-type bearings.

Couplings

Centrifugal pumps are connected to their drivers through **couplings** of one sort or another, except for close-coupled units (Fig. 14.2.14), in which the impeller is mounted on an extension of the shaft of the driver. Couplings used with centrifugal pumps can be either rigid (of the clamp or compressor type) or **flexible** (pin-and-buffer, gear, grid, or flexible-disk type). (See also Sec. 8.)

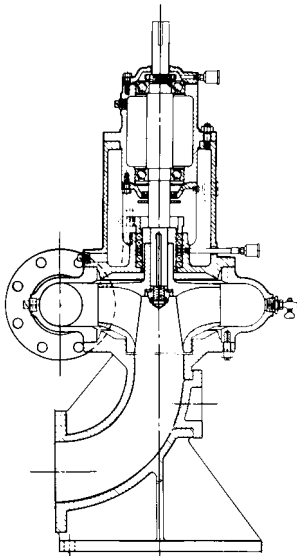


Fig. 14.2.15 Small vertical sewage pump with intermediate shafting.

Pump Mounting

It is desirable that pumps and their drivers be removable from their mountings. Consequently, they are usually bolted and doweled to machined surfaces that, in turn, are firmly connected to the foundations.

These machined surfaces are usually part of a common **bedplate** on which the pump and its driver have been prealigned. Bedplates are made of either cast iron or structural steel. Cast-iron or steel **soleplates** are customarily used for vertical dry-pit pumps and also for some of the larger horizontal units.

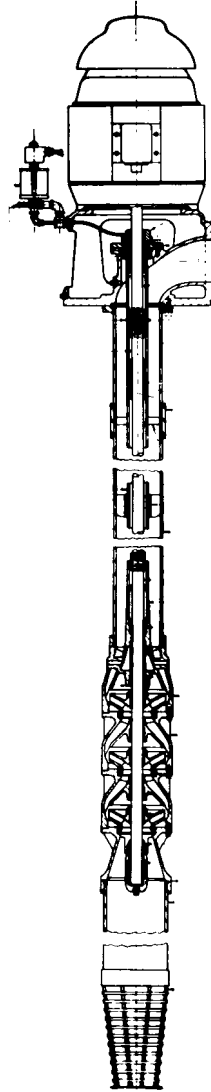


Fig. 14.2.16 Vertical turbine pump with closed impellers and oil-lubricated enclosed shafting.

Vertical Pumps

Dry-pit pumps with external bearings include most sewage pumps, most medium and large drainage and irrigation pumps for medium and high head, many large condenser-circulating and water-supply pumps, and many marine pumps. Some vertical dry-pit pumps are basically horizontal designs with minor modifications to adapt them for vertical-shaft drive. Other applications, such as small- and medium-sized sewage pumps, employ a purely vertical design. Most of these sewage pumps have elbow suction nozzles (Fig. 14.2.15) containing a handle to provide easy access to the impeller. Although the driving motors are frequently mounted right on top of the pump casing, the use of vertical-shaft design permits mounting the motor at an elevation sufficiently above the pump to prevent accidental flooding. For such applications, the pump and its driver are separated by a length of shafting, which may require steady bearings between the two units.

Vertical wet-pit centrifugal pumps can be classified as (1) vertical turbine pumps, (2) propeller or modified propeller pumps, (3) sewage pumps, (4) volute pumps, and (5) sump pumps. The first of these is the most common type. **Vertical turbine pumps** (Fig. 14.2.16) are built with either closed or semiopen impellers and with either enclosed or open line shafting. The bowl assembly consists of the suction head, the impeller or impellers, the discharge bowl, the intermediate bowl or bowls, the discharge case, the various bearings, the shaft, and miscellaneous parts such as keys and impeller locking devices. The column-pipe assembly consists of the column pipe itself, the shafting above the bowl assembly, the shaft bearings, and the cover pipe or bearing retainers. The pump is suspended from the driving head, which consists of the discharge elbow, the motor or driver support, and either the stuffing box (in open-shaft construction) or the assembly for providing tension on and the introduction of lubricant to the cover pipe.

self, the shafting above the bowl assembly, the shaft bearings, and the cover pipe or bearing retainers. The pump is suspended from the driving head, which consists of the discharge elbow, the motor or driver support, and either the stuffing box (in open-shaft construction) or the assembly for providing tension on and the introduction of lubricant to the cover pipe.

MATERIALS OF CONSTRUCTION

Centrifugal pumps can be fabricated of almost any of the known common metals or metal alloys, as well as of porcelain, glass, and even synthetics. A listing of materials commonly recommended for various

Table 14.2.2 Materials for Various Fittings

(Materials for bearing housings, bearings, and other parts are not usually affected by the liquid handled)

Part	Standard fitting	All-iron fitting	All-bronze fitting
Casing	Cast iron	Cast iron	Bronze
Suction head	Cast iron	Cast iron	Bronze
Impeller	Bronze	Cast iron	Bronze
Impeller ring	Bronze	Cast iron or steel	Bronze
Casing ring	Bronze	Cast iron	Bronze
Diffuser	Cast iron or bronze	Cast iron	Bronze
Stage piece	Cast iron or bronze	Cast iron	Bronze
Shaft, with sleeve	Steel	Steel	Steel, bronze, or Monel
Without sleeve	Stainless steel or steel	Stainless steel or steel	Bronze or Monel
Shaft sleeve	Bronze	Steel or stainless steel	Bronze
Gland	Bronze	Cast iron	Bronze

liquids can be found in the Standards of the Hydraulic Institute. Table 14.2.2 indicates the materials most commonly used for various pump parts.

PUMP PERFORMANCE

The performance of a centrifugal pump is generally described in terms of the following of its **characteristics**: (1) rate of flow, or **capacity** Q , expressed in units of volume per unit of time, most frequently ft³/s, gal/min, or m³/h (1 ft³/s = 449 gal/min; 1 m³/h = 4.403 gal/min); (2) increase of energy content in the fluid pumped, or **head** H , expressed in units of energy per unit of mass, usually ft · lb/lb or, more simply, ft, or m; (3) input **power** P , expressed in units of work per unit of time, bhp; (4) **efficiency** η , the ratio of useful work performed to power input, (5) **rotative speed** N , in r/min.

Since the parameters indicated are all mutually interdependent, it is customary to represent the performance of a centrifugal pump by means of **characteristic curves** similar to that shown in Fig. 14.2.17. While it is possible, within certain limits, for the pump designer to regulate the shape of these curves to suit the needs of a particular application, this is essentially a function of design capacity, head, and speed and hence for

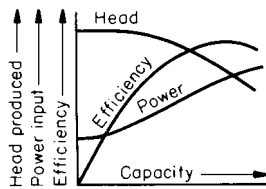


Fig. 14.2.17 Typical characteristic curves at constant speed.

a standard production unit is determined by experiment and is not subject to modification. For any given capacity on such a characteristic curve, the relationship between performance characteristics is expressed by the equation $\eta = \gamma QH/3,960P$, where η is efficiency expressed as a decimal, γ is the specific gravity of the fluid pumped, Q is in gal/min, H in ft, and P in bhp. In metric units, $\eta = \gamma QH/270P$, where Q is in m³/h and H in meters.

Inasmuch as the actual performance of centrifugal pumps is determined largely by experimental means, it is highly desirable to be able to use the results of past tests as a basis for predicting the performance of future designs. To this end, an interesting and widely used characteristic number known as **specific speed**, $N_s = N\sqrt{Q}/H^{3/4}$, has been developed. In this expression, values of N , Q , and H are all for the point of best efficiency. While not dimensionless, N_s is generally expressed simply as a number since its practical application is such that units are of no consequence except for their influence on the absolute magnitude of the number itself. Specific speed is of interest to both the pump designer and the pump user essentially in two ways: (1) All geometrically similar pumps, regardless of their size, will have identical specific speeds (but

all pumps of the same specific speed will not necessarily be geometrically similar). (2) Within reasonable limits, pump geometry and performance can be predicted as a function of N_s and Q . For N in r/min, Q in gal/min, and H in ft, the practical range of N_s is approximately 500 to 15,000. In metric units Q is in m³/s and H is in meters, so that this range

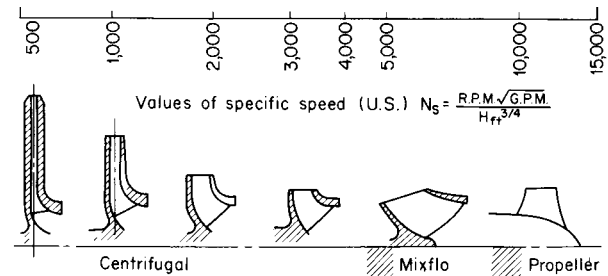


Fig. 14.2.18 Approximate relative impeller shapes versus specific speed.

becomes approximately 10 to 300 ($N_{sm} = N_s/51.66$). The shape of the profiles for impellers varies over the range of specific speeds, as shown in Fig. 14.2.18. The commercially attainable efficiencies also vary with the specific speed (Fig. 14.2.19). It must be remembered that these efficiencies are attainable at the capacity at best efficiency. The actual specified capacity may differ from this best-efficiency flow. In addition, the general shape of the pump-characteristic curves will vary widely from one end of this range to the other, as illustrated by Figs. 14.2.24 and 14.2.25.

Pump Theory

The basic purpose of a centrifugal pump in any fluid-handling system is to add energy to the fluid, and since it is a dynamic machine, the pump depends entirely on changes in velocity relationships to provide the energy. While the measurable evidence of energy addition is in most cases largely in the form of static pressure, this is partially the result of velocity reductions and constraints occurring in the diffuser or casing and to this extent represents a conversion from the velocity energy produced by the impeller. Thus, any discussion of centrifugal-pump theory generally becomes a discussion of velocities occurring at various points within the pump.

The true velocity relationships existing within a pump are extremely complex and, to a substantial degree, still unknown in their ultimate detail; but for practical purposes, a one-dimensional analysis serves to illustrate the basic concepts and, indeed, has served as the basis of design for virtually all centrifugal pumps ever built.

For **radial and mixed-flow impellers** ($500 \leq N_s \leq 7,500$ or $10 \leq N_{sm} \leq 150$) the velocities at the inlet and outlet of the impeller are shown by the vector diagrams in Fig. 14.2.20. The head produced by such an impeller is represented by

$$H = \eta_H(U_2V_{i2} - U_1V_{i1})/g \tag{14.2.1}$$

where H = head, ft (m); U = circumferential velocity of impeller at radius being considered, ft/s (m/s), V_u = average value of the circumferential component of absolute fluid velocity, ft/s (m/s), and $g = 32.174 \text{ ft/s}^2$ (9.80 m/s^2). Subscript 1 refers to the impeller inlet section, and subscript 2 to the impeller discharge section. The coefficient η_H is

$$H = \eta_H U_2 V_{u2} / g \tag{14.2.2}$$

In practice, Eq. (14.2.2) will provide a close approximation to total head for any pump up to a specific speed of 2,000 ($N_{sm} \approx 40$) since the term $U_1 V_{u1}$ in Eq. (14.2.1) is very small compared with $U_2 V_{u2}$.

It should also be noted in Fig. 14.2.20 that the relative velocity v_2 does not coincide in direction with the vane angle at the impeller discharge. The angular difference between v_2 and the direction of the vane is due to the irrotational nature of the flow between the vanes. The effect of this difference can be taken into account as the vector difference $V_{u2}^* - V_{u2} = Na/229$, where a is the shortest distance in inches taken in the radial plane between the discharge tip of any vane and the upper surface of the following vane. ($V_{u2}^* - V_{u2} = Na/19,100$, where a is in mm and velocities are in m/s.)

The foregoing relationships provide the basis for determining impeller diameters and vane angles required to produce a given total head requirement at a specified rotative speed. It is equally necessary, of course, to provide in the design for handling a specified flow volume, and this is readily accomplished by providing the necessary cross-sectional area A between vanes to pass the required flow at velocities previously determined. A useful relationship for this purpose, in light of the units commonly employed in pump design, is $Q = AV/0.321$, where Q is in gal/min, V in ft/s, and A in in^2 ($Q = AV/278$, where Q is in m^3/h , V in m/s, and A in mm^2).

Upon leaving the impeller, the liquid pumped enters either (1) a system of diffusing vanes surrounded by an outer casing or (2) directly into a casing designed to contain the fluid and control its velocities. Where diffusing vanes are used, they are designed on the basis of velocity relationships very similar to those employed in impeller design but with the objective being to slow the fluid down to convert velocity energy to pressure energy and, further, to reduce frictional losses in the discharge system following the diffuser. Where the impeller discharges directly into the casing, this component of the machine is most frequently designed in the form of a volute to provide constant velocity all around the impeller periphery up to the point of entry into the discharge nozzle. From this point, commonly called the **casing throat**, to the discharge flange or to the inlet of a succeeding stage, the velocity is gradually reduced. Special circumstances related to pump design or application often result in modifications to the constant-velocity design of such a casing, and variations may be found covering the entire range from constant velocity to constant area. In addition, many casings are now designed with one or more spiral vanes placed in such a way as to approximate a condition of geometric similarity in relation to the impeller, which is advantageous when a pump is operated at capacities other than those for which it is designed. A casing of this nature represents an effort by the designer to obtain an optimum balance between the desirable geometric similarity of the diffuser discharge and the manufacturing simplicity and generally high efficiency of the volute-type casing. The most common form of such a casing is the twin-volute type discussed under Nomenclature and Mechanical Design (see Fig. 14.2.1 and Table 14.2.1).

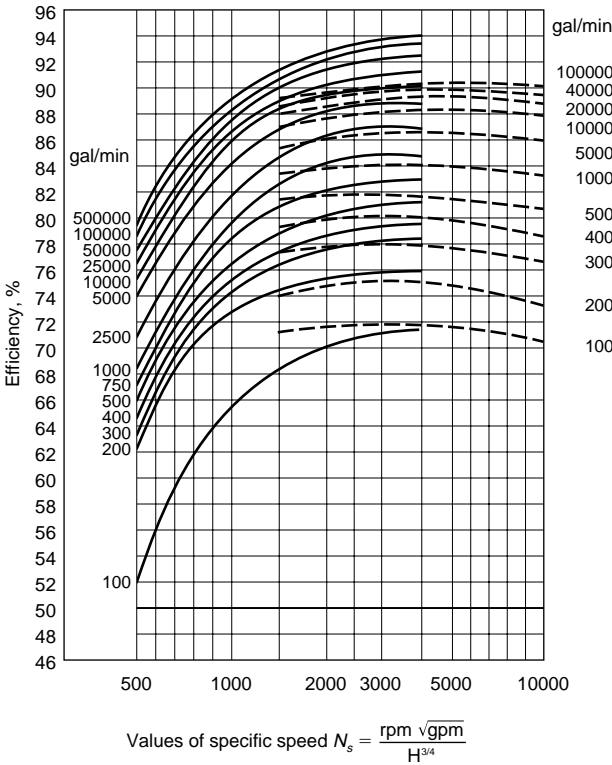


Fig. 14.2.19 Efficiencies of single-stage end-suction and double-suction centrifugal pumps (solid lines) and wet pit pumps (dashed lines).

the hydraulic efficiency of the rotating-vane system and, for the range of specific speeds indicated above, will generally fall between 0.85 and 0.95. This hydraulic efficiency is considerably higher than pump efficiency η since it does not include mechanical losses due to bearing or packing friction, volumetric losses due to internal wearing-ring clearances, impeller-disk friction, or fluid-friction losses due to velocity conversion or boundary-layer considerations ahead of or following the impeller. For pumps in this specific-speed range, the hydraulic losses $1 - \eta_H$ will generally be between one-quarter and three-quarters of total pump losses $1 - \eta$.

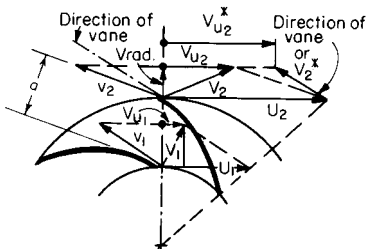


Fig. 14.2.20 Velocity diagram of a radial-flow impeller.

For pumps arranged with an axial inlet to the impeller (such as that shown in Fig. 14.2.5), it is generally assumed that the entering flow will

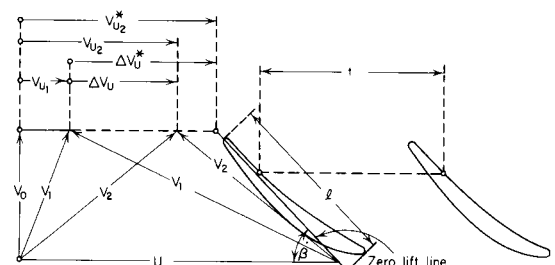


Fig. 14.2.21 Velocity diagram of an axial-flow vane system.

For axial-flow impellers ($7,500 \leq N_s \leq 15,000$ or $150 \leq N_{sm} \leq 300$) velocity relationships can be approximated in a manner similar to that used for lower-specific-speed pumps, but refinement of these approximations is approached in a somewhat different manner, largely because of the considerable body of knowledge available in the form of airfoil data which can be applied. Velocity diagrams for pumps of this type are shown in Fig. 14.2.21.

Considering a cylindrical stream tube intersecting the vanes of an axial-flow impeller, we can rewrite Eq. (14.2.1) in the form

$$H = \eta_H U \Delta V_u / g \quad (14.2.3)$$

where ΔV_u represents the increase in the tangential component of the absolute velocity as the fluid passes through the impeller. For pumps in this specific-speed range, η_H will generally fall between 0.80 and 0.90 and $1 - \eta_H$ will generally be between one-half and three-quarters of $1 - \eta$.

As in the case of radial-flow impellers, the relative velocity v_2 at the impeller exit does not coincide with the vane angle. In this case, the necessary correction can be applied by means of the expression

$$\Delta V_u = 2\Delta V_{u^*} [(t/l)(2/\pi K)(1/\sin \beta) + 1] \quad (14.2.4)$$

where t is vane spacing, l is vane length, β is the discharge vane angle, and K is the coefficient determined from Fig. 14.2.22, which provides for the fact that the impeller blades are, in effect, arranged in a continuous lattice.

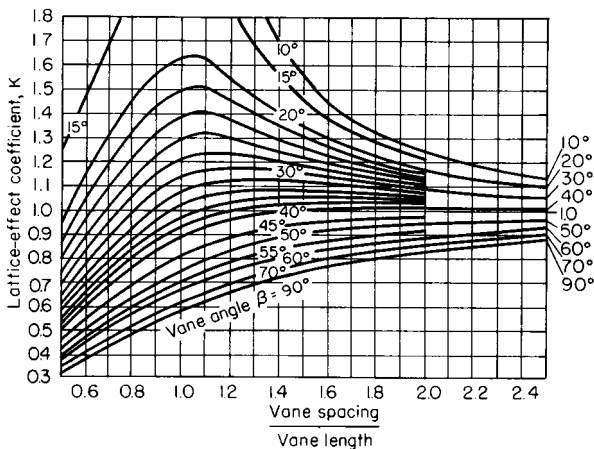


Fig. 14.2.22 Lattice-effect coefficient. (Weinig.)

In the design of axial-flow pumps, it is generally assumed that the head developed by the blade elements within all the cylindrical stream tubes between the impeller hub and its outer diameter will be the same. For this condition to be achieved, it will be evident from Eq. (14.2.3) that since U will vary directly with the radius, ΔV_u must vary inversely with the radius. Thus vane camber (or curvature) will be greater near the hub than at the periphery. It is further generally assumed that the axial velocity is constant throughout the impeller, and to satisfy this condition, the blade angles will be greater at the hub than at the periphery, giving rise to the twist of the vane.

By designing the vane element at the hub, the periphery, and any reasonable number of radial stations between, it is possible to define the vane over its entire surface to produce the total head desired. The design capacity can be simply established from the constant axial velocity and the annular area between the hub and the periphery.

Axial-flow impellers will almost invariably discharge into a vaned diffuser designed primarily to convert into pressure the tangential component of the absolute velocity leaving the impeller, thus producing a uniform, nonrotating velocity profile at the pump discharge or at the entrance to any succeeding stage.

Pump Application

In applying any centrifugal pump to a practical fluid-handling system, the engineer generally has two variables in the pump which may be used to effect a match between it and the system, namely, speed of operation and impeller diameter. (In axial-flow pumps, impeller diameter cannot conveniently be changed, but similar results can be accomplished within a limited range by reducing vane length.) To take advantage of these variables, it is necessary to understand the similarity relationships which govern pump behavior.

The effect of change in speed can be most readily explained by referring back to Eq. (14.2.1). For a fixed impeller diameter, it will immediately be evident that any increase in rotative speed will result in a directly proportional increase in the peripheral velocities of the impeller at both inlet and outlet, U_1 and U_2 , respectively. Furthermore, since the directions of both the relative and absolute velocities v and V , respectively, are controlled by the vane angles, it follows that both the inlet and outlet velocity diagrams remain geometrically similar, in other words, as pump speed changes, all fluid velocities change in direct proportion. Thus the effect of a speed change on pump capacity can be represented for two speeds, N_1 and N_2 by

$$Q_1/Q_2 = N_1/N_2$$

Referring again to Eq. (14.2.1), it will be evident that the change in head will be proportional to the square of the change in speed, resulting in

$$H_1/H_2 = N_1^2/N_2^2$$

Also, since P is proportional to QH , then

$$P_1/P_2 = Q_1 H_1 / Q_2 H_2 = N_1^3 / N_2^3$$

The effect of a reduction in impeller diameter is for practical purposes, identical to that of a reduction in speed, and the above equations can all be rewritten in the same form, substituting D_1 and D_2 for N_1 and N_2 . In this case, however, there is somewhat less latitude for change, the practical limit for cut-down being approximately 25 to 30 percent in the case of low-specific-speed impellers and decreasing from this value as specific speed increases.

From the foregoing it should be noted that running a pump at speeds far below its normal rated speed would be uneconomical since the pump would obviously be oversized for the service. Conversely, the same pump could not be run at speeds far above its rating since it would soon exceed its horsepower capability. Thus the normal approach to pump application calls for selection of a unit which will meet the system requirements at or near the pump's maximum rated speed and preferably as close to maximum impeller diameter as possible.

A further aspect of the similarity laws, of interest primarily to pump designers, is the matter of geometrically similar pumps, or as they are commonly called, **factors of each other**. Assuming two such pumps with a size ratio f_1/f_2 , then at the same rotative speed $H_1/H_2 = f_1^2/f_2^2$, which is the same as for a change in speed of the same ratio. In the case of capacity, however, we have, in addition to the linear increase due to velocity change, a further increase due to the change in the cross-sectional area of all fluid channels which is proportional to the square of the size factor, resulting in the relationship $Q_1/Q_2 = f_1^3/f_2^3$. The ratio of horsepower requirements therefore becomes $P_1/P_2 = f_1^5/f_2^5$.

Lines of geometrically similar pumps are more frequently designed, however, for constant head than for constant speed, and to have this condition apply for the size ratio indicated in the preceding paragraph, it is necessary that the speeds vary in exact inverse proportion to size ratio. Thus, for constant head, $N_2/N_1 = f_1/f_2$ and

$$\begin{aligned} H_1/H_2 &= 1 = (f_1/f_2)^2 (N_2/N_1)^2 \\ Q_1/Q_2 &= (f_1/f_2)^3 (N_2/N_1) = (f_1/f_2)^2 \\ P_1/P_2 &= (H_1/H_2)(Q_1/Q_2) = (f_1/f_2)^2 \end{aligned}$$

To simplify selection of pumps, it is customary to plot, for any given speed, performance curves to show pump characteristics over the available range of impeller diameters rather than at the single diameter which would be implicit in a curve of the type shown in Fig. 14.2.17. A typical

rating curve of this nature is shown in Fig. 14.2.23. Such rating curves will vary widely in their nature with changes in N_s , and an understanding of these variations is essential to the selection of properly sized pump drivers and, in many cases, proper design of discharge piping and/or the determination of limiting ranges of pump operation. Some

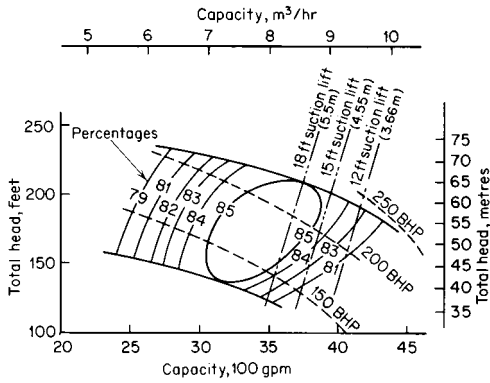


Fig. 14.2.23 Rating curve of 10-in double-suction single-stage pump.

idea of the diversity of characteristics which is available can be obtained by a comparison between Figs. 14.2.24 and 14.2.25, both of which are plotted entirely in percentages of design values occurring at the point of best efficiency.

In the case of the lower-specific-speed pump shown in Fig. 14.2.24, a

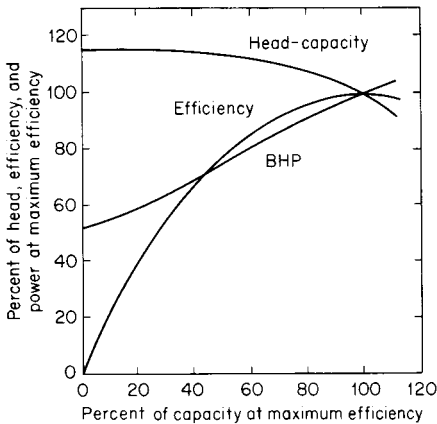


Fig. 14.2.24 Type characteristics for $N_s = 1,550$ single-suction impeller ($N_{sm} = 30$).

driving motor selected for the rated condition would be more than adequate at lower capacities, and discharge piping would be subjected to only modest overpressures. On the other hand, if the pump shown in Fig. 14.2.25 is to be operated at reduced capacities, then both motor size and discharge piping must be designed for the minimum flows to be encountered.

Selection of drivers, as well as pumps, is also influenced by **properties of the fluid handled**. Pump horsepower varies directly with the specific gravity of the liquid and is influenced in a more complex manner by viscosity. The effect of the latter is illustrated in Fig. 14.2.26. For the example shown on the chart, the pump in question is rated for 750 gal/min (170 m³/h) at 100-ft (30-m) head. When handling a fluid having a viscosity of 1,000 SSU (220 cS), its capacity is reduced to 95 percent of that on water over its complete range of operation. Its head is reduced to 96, 94, 92, and 89 percent of its head on cold water at 60, 80, 100, and 120 percent of these reduced capacities, and its efficiency is

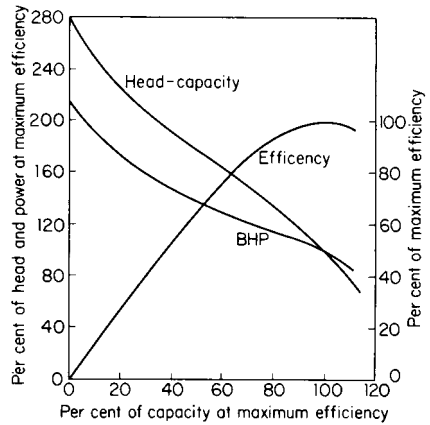


Fig. 14.2.25 Type characteristics for $N_s = 10,000$ single-suction impeller ($N_{sm} = 194$).

reduced to 63.5 percent of its efficiency on water over its reduced capacity range. Thus, if the pump had a rated efficiency on cold water of 70 percent, it would now deliver 712.5 gal/min (161.8 m³/h) at 92-ft (28.0-m), head with an efficiency of 44.5 percent.

In order for a pump to deliver its rated output, it is obviously necessary that it be supplied with fluid at its inlet at the same rate. It is further necessary that the absolute pressure (including velocity head, $V^2/2g$) of the fluid at the inlet exceed the vapor pressure by an amount sufficient

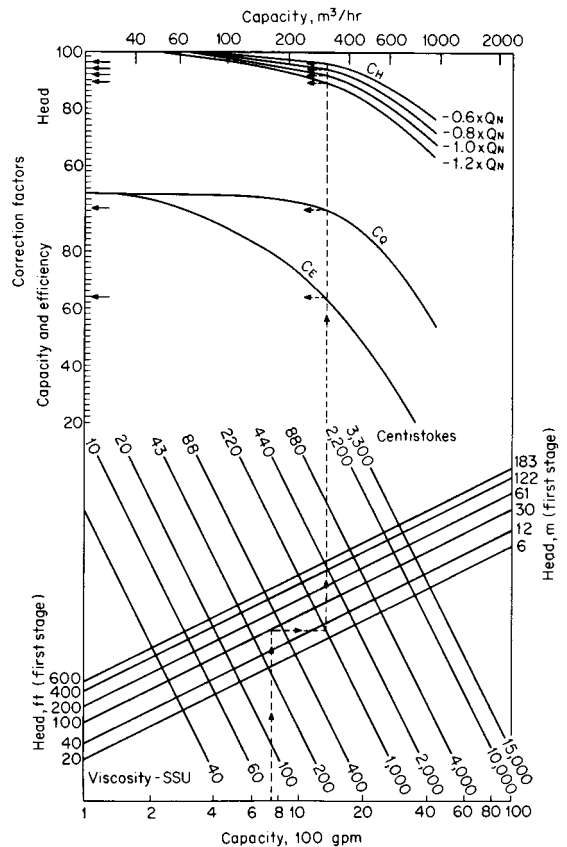


Fig. 14.2.26 Performance-correction chart for effect of viscosity. (Hydraulic Institute.)

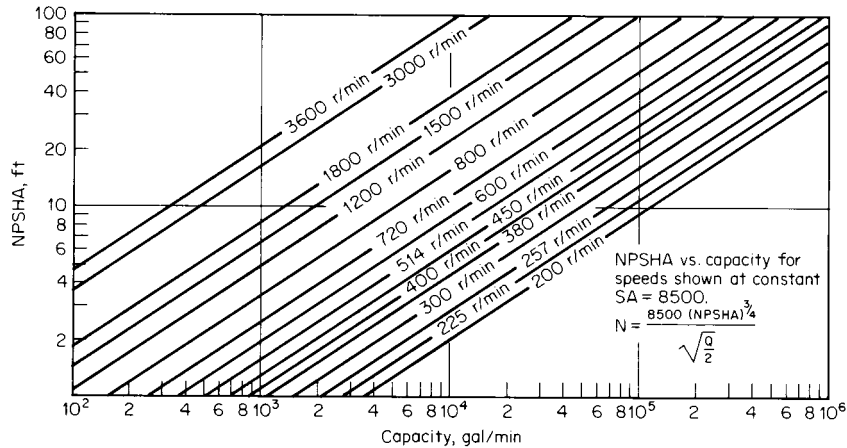


Fig. 14.2.27 Recommended maximum operating speeds for double-suction pumps. (Hydraulic Institute Standards.)

to overcome (1) any entrance or frictional losses between the point of entry into the pump and the impeller, and (2) the shock losses occurring at the impeller inlet. This gives rise to the definition of **net positive suction head (NPSH)** which is the absolute pressure at the pump inlet expressed in feet of liquid, plus velocity head, minus the vapor pressure of the fluid at pumping temperature, and corrected to the elevation of the pump centerline in the case of horizontal pumps or to the entrance to the first-stage impeller for vertical pumps.

NPSH required is determined by the pump manufacturer and is a function of both pump speed and pump capacity. **NPSH available** represents the energy level of the fluid over the vapor pressure at the pump inlet and is determined entirely by the system preceding the pump. Unless NPSH available at least equals NPSH required at any condition of operation, some of the fluid will vaporize in the pump inlet and bubbles of vapor will be carried into the impeller. These bubbles will collapse violently at some point downstream of the pump inlet (usually at some point within the impeller) and produce very sharp, crackling noises, frequently accompanied by physical damage of adjacent metal surfaces. This phenomenon is known as **cavitation** and is generally highly undesirable.

A term similar to that used for pump specific speed has been developed for pump inlet characteristics and has been identified as **suction specific speed**, $S = N \sqrt{Q} / H_{sv}^{3/4}$, where H_{sv} is NPSH. For double-suction impellers, $S = N \sqrt{Q/2} / H_{sv}^{3/4}$. The value S , like N_s , is expressed simply as a number and, for general-purpose pumps, does not generally exceed 10,000, where N is in r/min, Q is in gal/min, and H_{sv} is in ft. When Q is in m^3/s and H_{sv} is in meters, the equivalent value S_m is 194.

Within the limits of NPSH capabilities, it is desirable to select a pump of the highest specific speed since this will produce the highest pump speed and consequently use the smallest pump. As a convenience in accomplishing this, the Hydraulic Institute has published two charts indicating the recommended NPSH as a function of pump capacity and operating speed and, conversely, the maximum recommended operating speed for different capacities if the NPSH value is known or first selected. These charts are based on a recommended value of suction specific speed of 8,500 ($S_m = 165$). The chart for double-suction pumps is reproduced in Fig. 14.2.27.

Recirculation A better understanding has been recently reached of the flow pattern in a pump impeller when the pump is operated at capacities below that at its best efficiency. All pumps exhibit the phenomenon of recirculation which is a flow reversal at the inlet or at the discharge tips of the impeller. The capacities at which flow recirculation occurs at the suction or discharge are not necessarily coincidental. It is now possible to predict the flow patterns that must exist to produce this flow reversal. Depending on the size and speed of the pump, the effects of recirculation can be very damaging not only to the operation but also

to the life of the impeller and casing. The symptoms associated with recirculation and a method for calculating the capacity at which it takes place at both the suction and discharge of the impeller are given in Fraser, "Recirculation in Centrifugal Pumps," ASME Winter Annual Meeting, Nov. 16, 1981, Washington, D.C.

Just as the head-capacity characteristic of a pump is represented by a curve indicating reducing head developed with increasing capacity, the head-capacity requirements of the system served by the pump can be depicted by a curve showing increased head requirements for increased flow, i.e., a **system-head curve** (Fig. 14.2.28). The capacity at which a pump will operate in this system is that at which the pump head-capacity curve intersects the system-head curve.

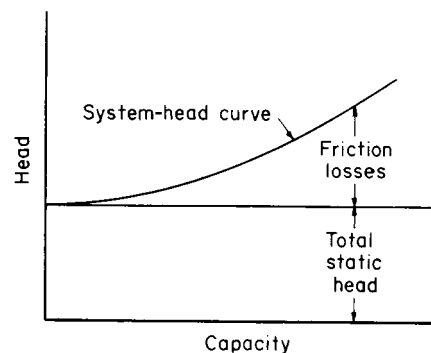


Fig. 14.2.28 System-head curve.

Every system-head curve consists of (1) a total static head (which may in some cases be equal to zero) plus (2) friction losses. The total static head $H_{stat} = H_d - H_s + P_d - P_s$, where H_d and H_s are the elevations at the system's termination and origin, respectively, and P_d and P_s are the gage pressures at the same points. In the special case of a continuous freshwater system where $P_d = P_s =$ zero, i.e., atmospheric, it must be remembered that siphon recovery cannot exceed 1 atm (34 ft less vapor pressure at sea level) and that recovery will not occur if the siphon leg is broken. Thus, any system over this height will have a total static-head component at least equal to the difference between its highest elevation and H_s minus the static-head equivalent of 1 atm plus the vapor pressure, even where $H_d = H_s$.

The **friction losses** H_{fr} are the sum of the line losses H_1 , fitting losses H_f , and changes in velocity head across the system, H_v . (See Sec. 3 for calculation methods.) H_v is simply the velocity-head difference between the system's termination and origin; $H_v = V_d^2/2g - V_s^2/2g$.

In complex pumping systems, such as municipal water-supply operations, both the total static-head and the friction-loss components may be variable, e.g., the former by changes in reservoir levels and the latter by the particular combination of lines being served at any given moment. This results in upper and lower limits of system-head requirements, and the intersection of these limiting curves with the pump head-capacity curve will define the capacity range over which the pump will be required to operate.

Since the capacity at which a pump will operate corresponds to the intersection of the system-head curve with the pump head-capacity curve, any changes in pump capacity can only be obtained by varying one or the other of these two curves (Fig. 14.2.29). Thus the capacity of a centrifugal pump operating in a system can be regulated by (1) changing the pump speed or (2) throttling in the discharge piping. The former method is preferable whenever the driver permits it, because throttling always involves an appreciable waste of power.

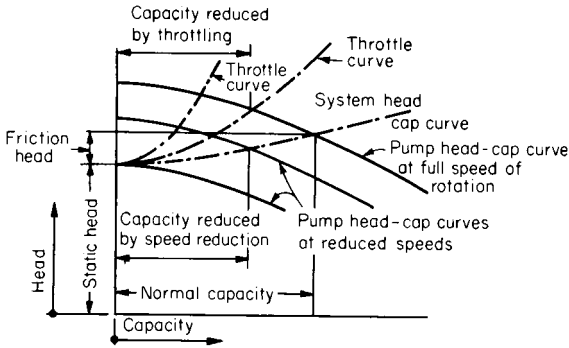


Fig. 14.2.29 Pump operation in a system.

Pump capacity should never be permitted to be reduced to zero because the fluid within the pump would have to absorb the entire power input and therefore would heat up rapidly with injurious effects on the pump. A small bypass line should be provided in the pump discharge line to permit a predetermined amount of liquid to flow through the pump if the discharge valve is closed entirely. This bypass may be operated manually or automatically.

Centrifugal pumps may sometimes be operated in **parallel** or in **series**. To construct the head-capacity curve of two pumps in parallel, it is merely necessary to add the capacities of the individual pumps for various total heads. The head-capacity curve of two pumps in series is constructed by adding the individual pump heads for various capacities. Figure 14.2.30 shows series and parallel operation of two centrifugal pumps with both flat and steep system-head curves.

Priming

A centrifugal pump is primed when the waterways of the pump are filled with the liquid to be pumped. When first put into service, the

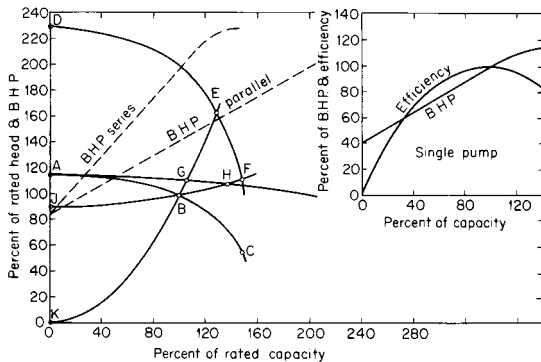


Fig. 14.2.30 Series and parallel operation of centrifugal pumps.

waterways are filled with air. If the suction supply is above atmospheric pressure, priming is accomplished by venting the entrapped air out of the pump through a valve provided for this purpose. If the pump takes its suction from a supply located below the pump itself, the air in the pump must be evacuated by some vacuum-producing device, by placing a foot valve in the suction line so that the pump and suction piping can be filled with liquid, or by providing a priming chamber in the suction line. Almost every commercially made vacuum-producing device can be used to prime pumps. Formerly, water- and steam-jet primers had wide application, but today electric-motor-driven vacuum pumps are most frequently used.

INSTALLATION, OPERATION, MAINTENANCE

Proper installation, operation, and maintenance of centrifugal pumps will vary widely over the complete range of services to which the pumps may be applied, and satisfactory results in these areas can only be fully achieved by following the manufacturer's instructions for the size and type of unit involved. There are, however, certain general considerations which should be observed and which will seldom need to be modified under any circumstances.

In general, the location selected for installation should be as close to the source of the fluid as possible, consistent with the requirement that adequate space be made available to provide accessibility for operation, inspection, and maintenance. The pumping unit should be mounted on a foundation of sufficient size and rigidity to support the unit itself plus the weight of the fluid it will contain during operation and to maintain accurate alignment. Piping should be independently supported and anchored to avoid imposing stresses on the pump, and suction piping in particular must be designed to minimize friction losses and to present a uniform velocity profile at the pump inlet. Suction and discharge (and/or check) valves must be suitable for the pressures involved and, in the case of large units, may also require independent support. If the pump will be required to operate against a suction lift, a suitable priming system must be installed, and where it is to be provided with a head-on suction, it will often be necessary to provide a venting arrangement. Care must also be exercised to ensure that all auxiliary connections for sealing, cooling, flushing, and drainage are made as required for the particular unit being installed.

Prior to initial operation of any centrifugal pump, it is necessary to make sure that the driver is connected to provide proper direction of rotation, that any shaft couplings between separate components of the entire unit are aligned within the manufacturer's stated limits, and that all bearings are provided with the proper amounts and grades of lubricants. The normal starting sequence will then be as follows: (1) open valves in all auxiliary sealing, cooling, flushing, and bypass lines; (2) open suction valve; (3) close discharge valve for low-specific-speed pumps where no check valve is installed after the pump, or open discharge valve for high-specific-speed pumps or wherever a discharge check valve has been provided; (4) prime or vent as necessary; (5) energize the driver; and (6) open discharge valve if it was previously closed in step 3.

Following start-up and until proper operation has been adequately established, it is desirable to monitor bearing temperature, stuffing-box leakage, and other outward symptoms of the unit's behavior. Securing of the pump is accomplished by a reversal of the start-up sequence, encompassing steps 6, 5, 3, and 1, in that order.

Until the early 1970s there were only four factors to consider when setting an acceptable minimum flow for centrifugal pumps:

Higher radial thrust developed by single-volute pumps at reduced flows.

Temperature rise in the liquid pumped as capacity is reduced.

Desire to avoid overload of drivers on high-specific-speed axial-flow pumps, which have a rising power consumption as capacity is reduced.

For pumps handling liquids with significant amounts of dissolved or entrained air or gas, there is the need to maintain sufficiently high velocities in the pump casing to wash out this air or gas with the liquid.

Since then, the phenomenon of internal recirculation in the impeller has been discovered. The recirculation occurs at flows less than the capacity at best efficiency. Problems include pressure pulsations at the suction and discharge and rapid deterioration of the pump and casing. The unfavorable effects of this internal recirculation have brought a fifth factor into the picture when setting minimum flows. This is discussed more fully above under "Recirculation."

Each of the effects mentioned above may dictate a different minimum operating capacity. Obviously, the final decision must be based on the greatest of the individual minimums. The internal recirculation usually sets the recommended minimum. Assuming that the pump is properly furnished with the necessary instrumentation, such as flowmeters, pressure gages with sufficient sensitivity to show pulsations, and vibration- and noise-monitoring equipment, an experienced test engineer should be able to pinpoint the onset of internal recirculation.

If the flow demand of the service in which the pump is installed is expected to fall below the minimum permissible flow, some means

must be provided to accommodate the difference between the minimum permissible flow and that required by the service. This is accomplished by installing a bypass in the discharge line from the pump, located on the pump side of the check and gate valves and leading to some lower pressure point in the installation where excess heat absorbed through operation at low flows may be dissipated. A valve is located in this bypass line. For small installations, the bypass valve is operated as fully open or fully closed. For large installations, a modulating valve is usually used to bypass the difference between the required flow and the minimum permissible.

In the matter of pump maintenance, a generally accepted cardinal rule is that as long as operation continues normal, the unit should be left alone. Thus, except in special circumstances, periodic overhauls are not recommended. The amount and degree of maintenance likely to be required are influenced primarily by the nature of the service to which the pump is applied, and maintenance practices must therefore be determined largely by the user as a result of personal experience.

14.3 COMPRESSORS

James L. Bowman

REFERENCES: Chlumsky, "Reciprocating and Rotary Compressors," E&FN Spons Ltd., Loomis, "Compressed Air and Gas Data," 3rd ed., Ingersoll-Rand. Rollins, "Compressed Air and Gas Handbook," 5th ed., Compressed Air and Gas Institute. Sheppard, "Principles of Turbomachinery," Macmillan. Stepanoff, "Turboblowers," Wiley. Koppers Research and Engineering Staff, "Engineers Handbook of Piston Rings, Seal Rings, Mechanical Shaft Seals," 8th ed., Koppers Company, Inc. Lundberg and Glanvall, "A Comparison of SRM and Globoid Type Screw Compressors at Full Load," Proceedings of the 1978 Purdue Compressor Technology Conference, Purdue University, W. Lafayette, IN. Von Nimitz, "Pulsation and Vibration Control Requirements in the Design of Reciprocating Compressor and Pump Installations," Proceedings of the 1982 Purdue Compressor Technology Conference, Purdue University, W. Lafayette, IN. Mechanics, Simulation and Design of Compressor Valves, Gas Passages and Pulsation Mufflers, short course, July 12-14, 1992, Purdue University, West Lafayette, IN.

Notation

a = speed of sound
 A = piston area
 A_v = valve flow area
 c_p = constant pressure specific heat
 c_v = constant volume specific heat
 C = clearance volume (decimal)
 d = depth of ring section, diameter
 D = diameter
 e = eccentricity
 E = modulus of elasticity (lb/in²)
 f = valve resistance, in velocity heads
 g = ring free gap less end clearance (in)
 g_c = gravitational constant
 ghp = gas horsepower
 h = enthalpy
 H_p = polytropic head (ft · lb/lbm or kJ/kg)
 k = ratio of specific heats
 K = area constant (decimal)
 L = length, leakage coefficient, connecting rod length
 L_{eq} = equivalent length of pipe
 m = mass
 MW = molecular weight
 n = polytropic coefficient
 N_m = number of lobes on main rotor
 n_s = number of stages

n_v = number of vanes
 N = rotational speed
 N_s = specific speed
 P = pressure
 Q = heat flow, volume flow at inlet conditions
 r = compression ratio (P_2/P_1)
 r_s = pressure ratio per stage
 r_t = pressure ratio across entire compressor
 r_δ = ratio of cylinder pressure at bottom dead center to inlet pressure
 R = gas constant [for air 53.34 ft · lbf/(lb_m · R)], crank radius
 s = entropy, stress
 sg = specific gravity relative to air
 T = absolute temperature, thrust
 T_v = vane thickness
 U = piston velocity, peripheral velocity
 V = volume
 V_i = built-in volume ratio
 \bar{V} = velocity
 W = work, mass flow
 Z = compressibility factor

Greek

α = ratio of piston area over valve flow area
 β = blade angle, degrees
 λ = wave length
 Δ = pressure loss, differential pressure (lb/in²)
 η_c = adiabatic efficiency
 η_p = polytropic efficiency
 η_v = volumetric efficiency
 μ = pressure coefficient (gH_p/U^2)
 ϕ = flow coefficient (Q/ND^3)
 ω = angular speed (radians/second)

Subscripts

1 = inlet conditions
 2 = discharge conditions
 A = axial
 R = radial

Compressed-Air and Gas Usage

Compressed-air is used for machine and tool operation, drilling, painting, soot blowing, pneumatic conveying, food processing, instrument operations, and in situ operations (e.g., underground combustion). Pressures range from 25 psig (172 kPa) to 60,000 psig (413,790 kPa). The largest usage is at 90 to 110 psig which is the normal plant air-pressure range.

Gas compressors are used for refrigeration, air conditioning, heating, pipeline conveying, natural gas gathering, catalytic cracking, polymerization, and in other chemical processes.

Standard Units and Conditions

In the ISO system, the standard unit of pressure for compressors is the kilopascal (kPa). In some countries this is the only unit which can appear on the compressor pressure gages by law. In Europe, the European Committee of Manufacturers of Compressors, Vacuum Pumps and Pneumatic Tools (PNEUROP) and, in the United States, the Compressed Air and Gas Institute (CAGI) prefer the bar as the standard unit of pressure. PNEUROP and CAGI have selected as standard conditions 1 bar (14.5 lb/in²) (100 kPa), 20°C (68°F), and 0 percent relative humidity. The unit of flow in the ISO system is m³/s. Other units still in common usage are m³/h, m³/min, and L/s. In the United States the most commonly used units are ft³/min (cfm) and ft³/h (cfh). Power is normally expressed in kilowatts (PNEUROP) and horsepower (CAGI).

Thermodynamics of Compression

Most compressors are analyzed using the ideal-gas law and assuming constant specific heat. Real-gas deviations are handled by applying a compressibility factor (also called *super compressibility*). The ideal-gas law will give satisfactory results for nonhydrocarbon gases for pressures to approximately 1,000 psig (6,900 kPa) at normal temperatures. Most hydrocarbon gases and refrigerants deviate strongly from the ideal-gas laws even at moderate pressures. For these cases, thermodynamics property tables, Mollier Charts, or compressibility charts should be used. Unfortunately these are not available for many gases of industrial importance. In this case a generalized compressibility chart is used. Compressibility charts for various gases can be found in Loomis, "Compressed Air and Gas Data," 3d ed., Ingersoll-Rand, and Rollins, "Compressed Air and Gas Handbook," 5th ed., Compressed Air and Gas Institute.

Adiabatic Analysis

The equation of state for an ideal gas is:

$$PV = mRT \quad (14.3.1)$$

The first law of thermodynamics for a steady-flow process is (per unit mass):

$$Q = (h_2 - h_1) + \frac{\bar{V}_2^2 - \bar{V}_1^2}{2g_c} + W \quad (14.3.2)$$

Neglecting the kinetic energy of the gas and assuming constant specific heat, for a reversible adiabatic process, Eq. (14.3.2) becomes:

$$W = c_p(T_1 - T_2) \quad (14.3.3)$$

Substituting $(P_2/P_1)^{(k-1)/k}$ for T_2/T_1 and $k/(k-1)$ for c_p/R , Eq. (14.3.3) becomes:

$$W = P_1 V_1 \frac{k}{1-k} \left[\left(\frac{P_2}{P_1} \right)^{(k-1)/k} - 1 \right] \quad (14.3.4)$$

The same results can be obtained by evaluating $W = \int v dp$ on an idealized P-V diagram (Fig. 14.3.1).

The isentropic efficiency (ratio of isentropic work to actual work) is given by:

$$\eta_c = T_1 \frac{(P_2/P_1)^{(k-1)/k} - 1}{T_2 - T_1} \quad (14.3.5)$$

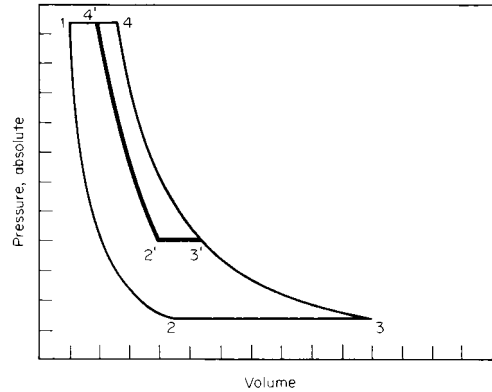


Fig. 14.3.1 Idealized indicator diagram for a positive-displacement compressor.

Polytropic Process

As an alternative to the adiabatic analysis, a polytropic process can be defined as

$$PV^n = C \quad (14.3.6)$$

All the adiabatic equations apply if $(n-1)/n$ is substituted for $(k-1)/k$ where

$$\frac{n-1}{n} = \frac{k-1}{k} \frac{1}{\eta_p}$$

If the inlet and discharge temperatures and pressures are known, n can be calculated from:

$$n = \frac{1}{1 - \ln(T_2/T_1) / \ln(P_2/P_1)} \quad (14.3.7)$$

where n is directly related to the heat transferred during the process by

$$n = \frac{C_p(T_2 - T_1) - Q}{C_v(T_2 - T_1) - Q} \quad (14.3.8)$$

The reversible work done along the polytropic path is called the **polytropic work**.

The polytropic efficiency η_p (ratio of polytropic work to actual work) is given by:

$$\eta_p = \frac{\frac{k-1}{k} \ln(P_2/P_1)}{\ln(T_2/T_1)} \quad (14.3.9)$$

The polytropic efficiency and the isentropic efficiency can be related by:

$$\eta_c = \frac{r^{(k-1)/k} - 1}{r^{(k-1)/(k\eta_p)} - 1} \quad (14.3.10)$$

Equation (14.3.10) is strictly valid only for ideal gases with constant specific heats. However, because the real-gas errors occur in both the numerator and denominator, the equation gives reasonably accurate results for real gases which deviate moderately from ideal gases.

Positive-displacement compressors are almost universally analyzed using the adiabatic model. Dynamic compressors are analyzed using both the adiabatic and polytropic models with the trend toward the polytropic model. The advantage of the polytropic model is that the discharge temperature is immediately available and η_p is essentially constant for different gases. The polytropic efficiency is independent of the thermodynamic state of the gas. Also, the total polytropic head is the sum of the polytropic heads for the stages. This is not the case for isentropic heads. The advantage of the isentropic model is that the isen-

tropic work can be read immediately from thermodynamic tables or charts. Comparing adiabatic efficiencies at different pressure ratios is not a valid procedure.

Real-Gas Effects

Deviations from the ideal-gas law are accounted for by introducing a compressibility factor

$$Z = \frac{PV}{RT} \quad (14.3.11)$$

The isentropic work of compression for a real gas becomes:

$$W = P_1 V_1 \frac{k}{1-k} \left[\left(\frac{P_2}{P_1} \right)^{(k-1)/k} - 1 \right] \frac{Z_1 + Z_2}{2Z_1} \quad (14.3.12)$$

where Z_1 and Z_2 are the compressibilities at conditions 1 and 2. Compressibility will also affect the volume flow of the compressor because of the reexpansion of the clearance volume gas.

Multistaging and Intercooling

Temperature rise and mechanical stresses limit the maximum pressure differential across a single stage of any compressor type. The pressure rise across a dynamic compressor stage is further limited by the available polytropic head which the stage can develop. The volumetric efficiency of a reciprocating compressor decreases with increasing pressure ratio placing a practical limit on the maximum ratio per stage.

Multistaging is used to overcome these limitations and to save power. With perfect intercooling and no pressure losses between stages, theoretically the minimum power is obtained when

$$r_s = \sqrt[n_s]{r_t} \quad (14.3.13)$$

where n_s = the number of stages; r_s = the pressure ratio per stage; and r_t = the ratio across the compressor.

For every 10°F of imperfect intercooling, the horsepower increases approximately 1 percent. Perfect intercooling is achieved when the temperature of the gas leaving the intercooler equals the temperature of the gas at the compressor inlet. The maximum theoretical work that can be saved by perfect intercooling is:

$$W = \left(\frac{k}{1-k} P_1 V_1 \right) \left[\left(\frac{P_2}{P_1} \right)^{(k-1)/k} - n_s \left(\frac{P_2}{P_1} \right)^{(k-1)/n_s k} + n_s - 1 \right] \quad (14.3.14)$$

On the ideal indicator diagram (Fig. 14.3.1), this is area 3'4'4'2'.

In practice, perfect intercooling is seldom obtained. Normal practice is to cool the interstage gas to within 15°F to 20°F of the inlet gas. For water-cooled machines, the coldest water available should be used for intercooling.

Positive-Displacement Compressors Versus Dynamic Compressors

Positive-displacement compressors collect a fixed volume of gas within a chamber and compress it by reducing the chamber volume. The ideal work for such a process is given by Eq. (14.3.4). The work could also be obtained from the area enclosed by an ideal indicator diagram (Fig. 14.3.1). The actual horsepower, taking into account real-gas deviations, is

$$\text{hp} = \frac{144 P_1 V_1 k}{33,000(1-k)} \left[\left(\frac{P_2}{P_1} \right)^{(k-1)/k} - 1 \right] \times \frac{1}{\eta_c} \left(\frac{Z_1 + Z_2}{2Z_1} \right) \quad (14.3.15)$$

with pressures in lbf/in² and volumes in ft³/min. With the volume rate V in m³/s and the pressures in pascals (dropping the 144/33,000), the power will be in watts.

For a given volume flow, the ideal power is affected by inlet pressure, the k value of the gas, and the compression ratio. The ideal power is not affected by the inlet temperature of the gas or its molecular weight.

The actual power is increased due to losses through the intake and discharge valves or ports as can be seen on the indicator diagram Fig. 14.3.2.

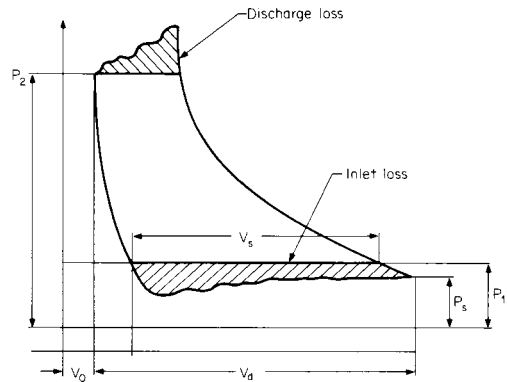


Fig. 14.3.2 Typical indicator diagram showing intake and discharge losses.

Additional losses are caused by turbulence within the compression chamber, preheating of the inlet gas, and leakage from the compression chamber. The isentropic compression efficiency (ratio of isentropic work to actual compression work) can be determined from the indicator diagram or calculated from Eq. (14.3.5) provided that there is no liquid injection. In addition to the thermodynamic losses, the mechanical losses must be added to the power requirement.

Volumetric efficiency is defined for positive-displacement compressors as

$$\eta_v = \frac{\text{actual delivery at intake conditions}}{\text{compressor displacement}}$$

For multistaging compressors, only the first-stage displacement is used.

Positive-displacement compressors are essentially constant-volume, variable-pressure machines as shown in Fig. 14.3.3. For fixed inlet and discharge pressures and varying speed, positive-displacement compressors are essentially constant-torque machines.

Dynamic compressors operate by transferring momentum to the gas via a high-speed rotor. The process is steady flow with the work given by Eq. (14.3.4). Head is defined as the energy per unit mass of the fluid. Head is imparted to the fluid through the change in magnitude and radius of the velocity components of the fluid as it passes through the rotor. The head developed by the rotor is

$$H = (U_1 V_{U1} - U_2 V_{U2}) / g_c \quad (14.3.16a)$$

Where U_1 and U_2 are the linear velocity of the rotor at radii 1 and 2. V_{U1} and V_{U2} are the fluid tangential velocity at radii 1 and 2.

The polytropic head required for a given compression ratio is

$$H_p = \frac{Z_{\text{ave}} RT_1 [r^{(n-1)/n} - 1]}{(n-1)n} \quad (14.3.16b)$$

The gas horsepower on a CFM basis is

$$\text{ghp} = Q_1 P_1 [r^{(n-1)/n} - 1] \frac{n}{229 \eta_p (n-1)} \left(\frac{Z_1 + Z_2}{2Z_1} \right) \quad (14.3.17)$$

and on a mass flow basis:

$$\text{ghp} = \frac{WH_p}{33,000 \eta_p} \quad (14.3.18)$$

With H_p in J/kg and W in kg/s, the power in kilowatts is

$$\text{kW} = H_p \left(\frac{W}{\eta_p} \right) \quad (14.3.19)$$

Mechanical losses must be added to the gas power to obtain the shaft power. Volumetric efficiency is not defined for dynamic compressors.

For a fixed impeller design and a fixed speed, the energy that is transferred to a unit mass of fluid as it passes through the impeller (i.e., the head) is constant. Pressure rise and power vary directly with inlet gas density independent of the cause of the density change. An increase in k will cause the pressure rise to decrease but will not affect the power.

A centrifugal compressor is essentially a constant-pressure, variable-capacity machine (see Fig. 14.3.3). An axial compressor is a constant-capacity, variable-pressure machine over a significant discharge pressure range. Dynamic compressors at least qualitatively follow the fan laws.

$$\frac{Q_1}{Q_2} = \frac{N_1}{N_2} \quad \frac{H_{p1}}{H_{p2}} = \left(\frac{N_1}{N_2}\right)^2 \quad \frac{ghp_1}{ghp_2} = \left(\frac{N_1}{N_2}\right)^3$$

The geometry of a dynamic compressor is dictated by the required flow, polytropic head, and speed. The specific speed is a dimensionless parameter used to select the proper geometry.

$$N_s = \frac{NQ^{(1/2)}}{H^{(3/4)}} \quad (14.3.20)$$

For specific speeds from 400 to 900, radial impellers are used. Between 800 and 1,400, mixed-flow impellers are preferred. For specific speeds above 1,400, axial impellers are normally used.

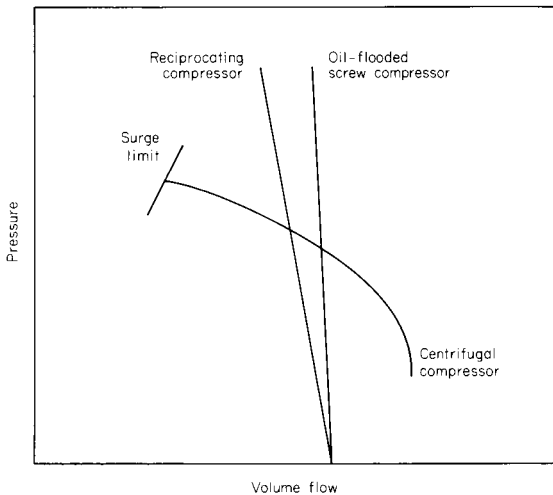


Fig. 14.3.3 Pressure capacity curves for different compressor types.

Surging

All dynamic compressors have a minimum flow point called the **surge limit** below which the operation of the machine is unstable. The surge limit is a function of the compressor type, design pressure ratio, gas properties, inlet temperature, blade angle, and speed. Operation at or below the surge limit must be avoided. The existence of the surge limit makes it essential to know the demand curve for the particular installation involved.

Reciprocating Compressors

Reciprocating compressors as shown in Figs. 14.3.4 and 14.3.5 range in size from fractional cfm to 15,000 cfm (25,485 m³/h) with discharge pressures as high as 60,000 psig (413,790 kPa). The majority of applications fall in the pressure range of 10 to 300 psig (690 to 2,069 kPa) and capacities less than 2,500 cfm (4,250 m³/h). Single-acting compressors (which compress gas on one side of the piston only) have their widest application below 50 hp (37 kW). Larger compressors are usually double-acting (both sides of the piston are used to compress gas).

Most plant air systems operate at 90 to 110 psig. Reciprocating and oil-flooded rotary-screw compressors share this market about equally. Above 200 psig, reciprocating compressors dominate.

Single-stage compressors of 100 hp (75 kW) operating at 125 psig were once common. Modern practice is to use multistaging with intercooling for pressures above 80 psig and sizes as low as 10 hp (7.5 kW). The intercooling saves power, and the reduced discharge temperatures improve safety and prolong compressor life.

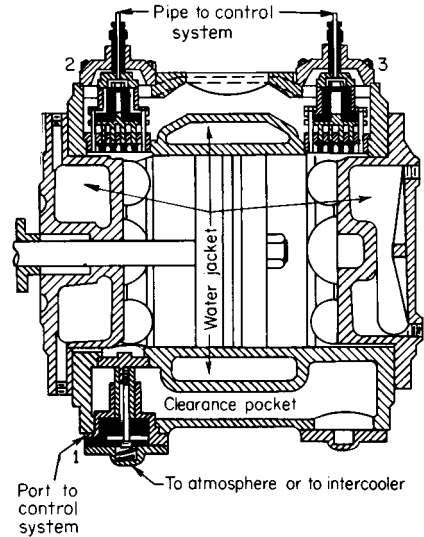


Fig. 14.3.4 Cross section of gas-compressor cylinder, showing water jackets, clearance pocket with pneumatic control, and suction-valve lift devices.

The capacity of a reciprocating compressor is determined by the first-stage displacement multiplied by the volumetric efficiency.

$$\eta_v = (1 + C)(r_\delta)^{(1/k)} - \frac{C(r)^{(1/k)}}{(Z_2/Z_1)} - L \quad (14.3.21)$$

where $r_\delta = P_s/P_1$; $r = P_2/P_1$; and $L =$ a leakage coefficient = 0.05 for lubricated compressors and 0.10 for nonlubricated. P_s = the pressure that exists in the cylinder when the piston is at bottom dead center. C is the clearance volume as a decimal fraction and includes valve passage volumes up to the sealing elements (point P_2V_c in Fig. 14.3.2). Clearance volume for normal service is 5 to 15 percent. High-pressure machines have as small a clearance volume as practical whereas pipeline compressors may have 100 percent clearance. The indicator volumetric efficiency from Fig. 14.3.2 is V_s/V_d . In general, this will be greater than the value obtained by Eq. (14.3.21).

The compression efficiency varies between 0.85 and .95. The mechanical efficiency falls in the range 0.88 to 0.95. The overall efficiency is the product of the compression and mechanical efficiencies.

The discharge temperature can be calculated from Eq. (14.3.22):

$$T_2 = T_1 \left[1 + \frac{(P_2/P_1)^{(k-1)/k} - 1}{\eta_c} \right] \quad (14.3.22)$$

If the inlet pressure is varied while holding the discharge pressure constant, the horsepower of the compressor will pass through a maximum value. This can be seen by substituting displacement times η_v for V_1 in the horsepower equation (14.3.12). For correct sizing of the driver, it is necessary to know the range of inlet pressures as well as discharge pressure. Interstage pressures are selected approximately by Eq. (14.3.13) with empirical adjustments made for interstage piping, intercooler, and valve losses. The loss of volume due to condensation in the intercoolers should be accounted for.

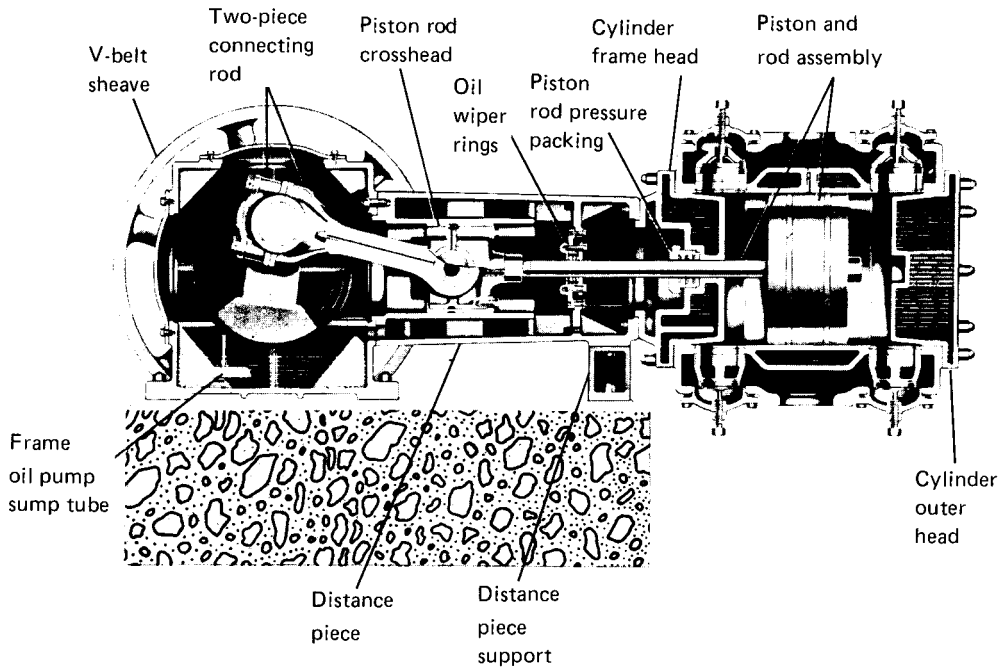


Fig. 14.3.5 Single-stage double-acting reciprocating compressor.

Valve losses can be approximated by:

$$\Delta = 1.26 \times 10^{-6} \alpha f U_{ave}^2 sg \left(\frac{P}{Z} \right) \text{ lbs/in}^2 \quad (14.3.23)$$

where α = ratio of piston area to valve flow area; f = valve resistance in velocity heads ($\cong 4$); sg = specific gravity relative to air; and U_{ave} = average piston speed, ft/s.

The piston velocity as a function of time is:

$$U = R\omega \left(\sin \omega t + \frac{R}{2L} \sin 2\omega t \right) \quad (14.3.24)$$

where R = crank radius; L = connecting rod length; and ω = crank speed (rad/s).

If there were no sealing elements in the valves, the velocity through the valves would be:

$$U_v = \left(\frac{A}{A_v} \right) U$$

This pulsating velocity is amplified by the valve sealing elements so that pulsating flow is inherent in reciprocating compressors. The full wavelength for a double-acting cylinder is $\lambda = 60 a/2N$, where a is the speed of sound. For a single-acting cylinder, it is two times this value. Pipes with total equivalent lengths equal to multiples of $\lambda/4$ should be avoided.

For a double-acting cylinder, the major problems occur at $\lambda/4$ and $3\lambda/4$ although resonance can occur at the other multiples of $\lambda/4$. Equivalent length L_{eq} is given by

$$L_{eq} = \frac{\lambda}{360} \tan^{-1} \left(\frac{2\pi V}{S\lambda} \right) \quad (14.3.25)$$

where V = equivalent cylinder and passage volume; and S = cross-sectional flow area of the pipe.

If it is not feasible to avoid the designated lengths, a volume bottle or an orifice can be installed in the piping system. If an orifice is used, it should be $1/2$ the pipe diameter and located at a node (one node occurs at the open end of the pipe). A volume bottle should be located as close to the cylinder as possible. Von Nimitz ("Pulsation and Vibration Control

Requirements in the Design of Reciprocating Compressor and Pump Installations," Proceedings of the 1982 Purdue Compressor Technology Conference, Purdue University) recommends the following equations for sizing:

$$V_s = 4PD \left(\frac{kT_s}{MW} \right)^{1/4} \quad V_d = \frac{V_s}{R^{1/k}} \quad (14.3.26)$$

PD is the total double-acting displacement volume of all cylinders to be manifolded to the surge bottle (ft³). Subscript s refers to suction, d to discharge. R is the compression ratio. Temperatures are in degrees Rankine.

Compressor Valves

The effective valve area is defined as the product of the element lift and the sum of the valve-seat peripheries or strip edges, less the guide and end contacting surfaces. The plate and strip-type valves (Figs. 14.3.6 and 14.3.7) are used in air and low-pressure service. The concentric-

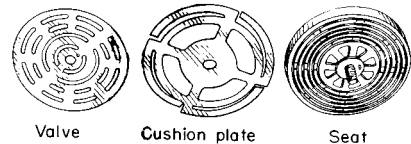


Fig. 14.3.6 Plate valve, illustrating floating element, cushion spring, and seat. (Ingersoll-Rand.)

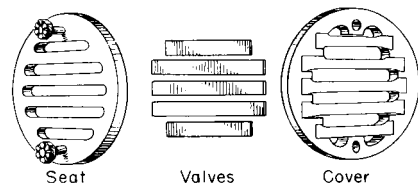


Fig. 14.3.7 Feather valve, illustrating seat, flexing elements, and valve cover. (Worthington.)

disk type (Fig. 14.3.8) is used in high-pressure, chemical, and natural-gas operations. The lift varies from 0.035 in (0.9 mm) for high-pressure, high-speed operation to 0.180 in (4.6 mm) for low-pressure, low-speed operation, with 0.100 in (2.5 mm) for general purposes. Springs are used to reduce the impact load. Element thickness ranges from 0.050 to 0.125 in (1.3 to 3.2 mm) for high-pressure service. The thickness, lift,

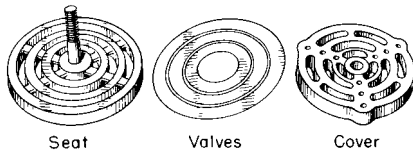


Fig. 14.3.8 Concentric-disk valve, illustrating seat, floating elements, and valve cover. (Chicago Pneumatic.)

and spring load are selected by trial and error. The feather valve element (Fig. 14.3.7) functions as a spring and sealing element. An 8-in (203-mm) strip has an average lift of 0.100 in (2.5 mm). The strip thickness varies from 0.02 to 0.09 in (0.5 to 2.0 mm). Multiple nylon poppet valves [lift = 0.25 in (6.3 mm)] have been successfully applied to pipeline and other services where the discharge temperature does not exceed 250°F (121°C).

Piston Rings

Reciprocating compressors employ compression rings, oil rings, and rider rings. Compression rings are always present, but whether oil rings or rider rings are used will depend on the type of compressor and its service.

The standard compression ring is the one-piece square-cut or angle-cut ring although two-piece and segmental rings are sometimes used. The one-piece ring usually has a rectangular cross section, but occasionally the cylinder contact face is crowned to prevent edge loading and to improve lubrication. Cast-iron rings are still widely used but are being displaced in many applications by filled TFE rings. Carbon-filled TFE rings are preferred for process compressors. For air compressors, carbon, glass, and bronze fillers are used. TFE rings can be used from -450 to 500°F (-268 to 260°C). Bronze-filled TFE is good for higher-temperature operations because of its higher thermal conductivity. Carbon-filled TFE is preferred for oil-free service. Many compressors that handle gases with corrosive components (e.g., H₂S) use piston rings made of cloth-reinforced thermoset laminates with molybdenum disulfide added to improve the lubricity. Polyamide, polyimide, and poly(amide-imide) rings work well with marginal or no lubrication.

A one-piece ring must be stretched over the piston to install it in the piston groove. The maximum stress occurs 180° from the joint and is given by Koppers Research and Engineering Staff ("Engineers Handbook of Piston Rings, Seal Rings, Mechanical Shaft Seals," 8th ed., Koppers Company, Inc.) as:

$$S_o = \frac{0.424 E(8d - g)/d}{(D/d - 1)^2} \quad \text{lb/in}^2 \quad (14.3.27)$$

When closed to the cylinder diameter, the ring is stressed to supply the initial seating pressure. The maximum stress (180° from the joint) is given by the Koppers "Engineers Handbook" as:

$$S_c = \frac{0.482 E (g/d)}{(D/d - 1)^2} \quad \text{lb/in}^2 \quad (14.3.28)$$

Oil rings are used in lubricated compressors to scrape oil from the cylinder wall and to meter a small quantity of oil to the compression rings. The oil rings may be above or below the wrist pin. Operating conditions for oil rings are less severe than for compression rings.

Nonlubricant compressors use rider rings to prevent piston cylinder contact. To prevent pressure buildup in front of the rider rings, longitudinal grooves are cut into the rings. Usually two or more rider rings are required.

Below 300 lb/in² differential, two compression rings and one or two oil rings are normally used. From 300 to 600 lb/in² differential, three to four compression rings are used. From 600 to 1,500 lb/in² differential, four to five rings are used. Above 1,500 lb/in², six or more rings may be required.

Piston-Rod Packing

Double-acting compressors are driven through a crosshead (Fig. 14.3.5) and require a rod packing to seal the piston rod connecting the piston to the crosshead. The operating conditions for rod packings are more severe than those for piston rings. In a double-acting compressor, the piston rings seal against the pressure differential across the stage whereas the rod packing seals the differential between discharge pressure and atmospheric pressure.

Rod packings use the same materials as piston rings. Metallic packing can tolerate rod wear of 0.15 percent of the rod diameter, with less tolerance above 2,000 lb/in² (13,790 kPa). The rod should be hardened to Rockwell 40 C and ground to a 10-rms or less finish.

Nonlubricated Cylinders

NL-1 class does not permit any direct lubrication and tolerates the minute contamination introduced along the piston rod. This can be eliminated by an additional spacer and rod wiper, qualifying the cylinder as a class NL-2. The pistons for this service use various Teflon rings and riders. The life of carbon rings is reduced by low humidities. The cylinder should have a ground 10- to 16-rms finish and a minimum hardness of Rockwell 20 C.

Lubrication

Lubricants serve to (1) prevent wear by providing a supporting film between the rubbing surfaces, (2) seal close clearances, (3) protect against corrosion, and (4) transmit heat of friction and minute wear particles away from points of contact. Industrial cylinders and packings require force-feed lubricators. Viscosity is the best index of suitability: general service at pressures below 500 lb/in² (3,448 kPa) requires 400 SSU (86 centistokes) at 100°F (38°C) (SAE 40); 700 SSU (151 centistokes) for 2,000 lb/in² (13,793 kPa); and 1,000 SSU (216 centistokes) (SAE 50) for 8,000 lb/in² (55,000 kPa) operations. These oils have very high viscosity at 40°F (45°C), which will result in poor lubrication. Good winterizing practice provides continuous circulation of warm jacket water and an immersed electric heater in the oil reservoir. Crankcase oils should include a foam inhibitor, sludge dispersant, and rust inhibitor. Phosphorous extreme-pressure agents are added to high-pressure lubricants to avoid wear and scuff damage. Castor-bean and rapeseed oils are additives resistant to solvent action of condensing hydrocarbons.

Cellulube, Houghto-safe, Pydraul AC, Anderol, and Fluorolubes are used to resist the hazard of exothermic reaction in air compression. The crankcase lubricant consumption for 10 plants over a score of years averaged 20,000 bhp/(h)(gal). Table 14.3.1 shows lubricant requirements for general-process and natural-gas operations.

Table 14.3.1 Lubrication Required for Compressor Cylinders

Range of cylinder diam, in	Film thickness, μm	10 ³ hp/(h) per gal	Pints per cylinder per day per 100 hp of load	Oil drops/min
24-36	0.40	13	1.5	13
15-23	0.45	19	1.0	9
10-14	0.55	24	0.8	7
7-9	0.60	32	0.6	5
4-7	1.20	24	0.8	7
3-5	1.40	27	0.7	6

NOTE: Based on 12,500 drops per pint of SAE 40 oil at 75°F with vacuum-type lubricator. Reduced drop count to roughly one-half for pressure-type lubricator. Example: 500 bhp, 20-in cylinder, requires 5 pint/day and feed of 45 drops/min. Each installation will vary on lubrication requirements. Manufacturers' recommendations on oil type and quantity should be followed.

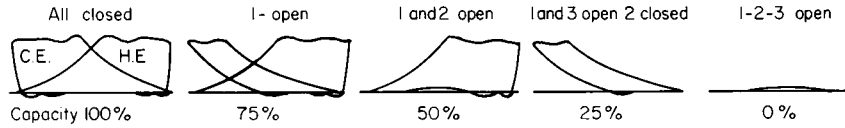


Fig. 14.3.9 Cards from a compressor with combination clearance and suction-valve bypass control.

Compressor Accessories

Most state laws and safe practice require a relief valve ahead of the first stop valve in every positive-displacement compressor. It is set to release at 1.25 times normal discharge or at the maximum working pressure of the cylinder, whichever is lower. The relief-valve piping system sometimes includes a manual vent valve and/or a bypass valve to the suction to facilitate start-up and shutdown operations. Quick line-sizing equations are (1) line connection, $d/1.75$; (2) bypass, $d/4.5$; (3) vent, $d/6.3$; (4) relief-valve port, $d/9$.

Volume flow is controlled by variable-speed drivers; steam engines can operate at 20 percent of rated speed and gas engines at 60 percent, and electric-motor speed can be varied by means of eddy-current and hydraulic couplings and special wound rotors with rheostats, which are both costly and inefficient. Unloading can be applied by means of valve-lift and clearance pockets, as shown in Figs. 14.3.4 and 14.3.9. Utility air plants are throttled with suction unloaders, as shown in Fig. 14.3.10. Suction throttling and bypass controls are used in process operations.

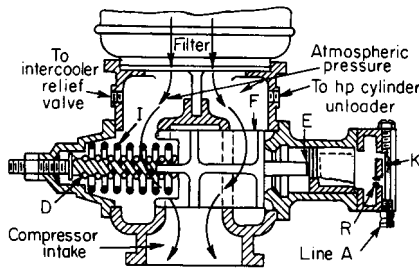


Fig. 14.3.10 Air suction unloader. Piston E moves to closed position as line A pressure reaches design point.

Cylinder Cooling

Single-acting compressors, especially smaller ones, are usually air cooled by fins cast into the cylinder. Cooling air is drawn across the fins by a fan. Double-acting compressors usually have cylinders that are water jacketed.

The principal purpose of a jacket-water system is to normalize cylinder-casting strains. It has some effect on compressor efficiency due to reduced preheating of the air. This effect is reduced as the compressor speed is increased. The heat rejection to the jacket water is approximated by:

$$H_j = 4(t_{ag} - t_{aw}) + 100 \text{ Btu}/(\text{bhp} \cdot \text{h}) \quad (14.3.29)$$

where t_{ag} and t_{aw} represent the average gas and water temperature, respectively, usually 155 and 140°F, respectively, and a rejection of 160 Btu/(bhp·h). Where the temperature rise is between 2 and 5°F, the jacket-water requirement is $160 \text{ bhp}/(500 \times 3.2 \times t_w)$, or 0.1 gal/(min·bhp). The use of cold water in jacket-water systems causes condensation on the cylinder walls, washing of lubricant, and excessive ring wear.

Rotary-Vane Compressors

Figure 14.3.11 is a section through a typical rotary-vane compressor. The displacement can be calculated using the following formula.

$$V_{th} = 2eL(\pi D - T_v N_v)N \quad (14.3.30)$$

Rotary-vane compressors have a built-in volume ratio V_i . This means that they will compress the gas within the compression chamber to $P_1(V_i)^n$ before opening the compression chamber to the discharge port. P_1 is the inlet pressure, and V_i is the ratio of the volume of the compression chamber at the inlet cutoff point to the volume at the point just prior

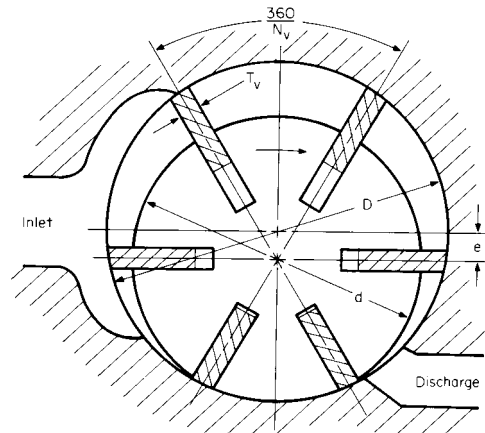


Fig. 14.3.11 Section through a typical vane compressor with N_v vanes.

to the opening of the discharge port. If the operating-pressure ratio does not match the built-in ratio, overcompression or undercompression will take place as indicated in Fig. 14.3.12. For this case the ideal work should be calculated as:

$$W_{ideal} = \frac{P_1 V_1}{(1-n)} [V_i^{n-1} - n] - P_2 \frac{V_1}{V_i} \quad (14.3.31)$$

The ratio of this work over the ideal work required for a perfectly matched compressor can be read from Fig. 14.3.16. For example, the theoretical work for an air compressor with $V_i = 2$ operating at $P_2 = 0.8P_1$ theoretically will require 2.5 percent more work than it would if it were designed so that the built-in ratio matched the operating ratio.

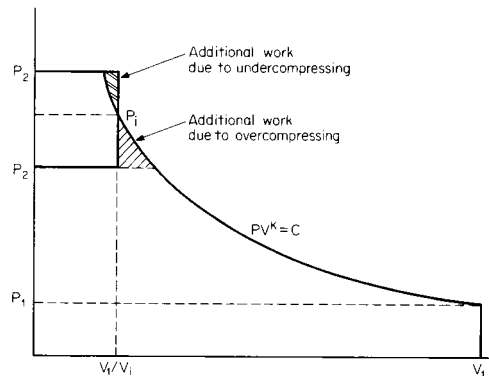


Fig. 14.3.12 Idealized P - V diagram for a compressor with a built-in volume ratio (also referred to as a built-in pressure ratio).

However, normal practice is to base efficiencies on a power calculated as though the compressor built-in ratio matches the operating ratio.

Rotary-vane compressors may be dry, lubricated, or oil-flooded. Prior to 1960 two-stage, oil-flooded vane compressors up to 900 cfm (1,530 m³/h) compressing to 150 psig (10.3 bar) (1,034 kPa) were used in portable compressors. They are now usually limited to less than 100 cfm (170 m³/h) air compressors and to small refrigeration compressors and boosters.

Vane materials may be phenolic (micarta, polyamid-imides, poly-imides, Ryton) or metal (aluminum-silicon alloy).

The major limitations of vane compressors are imposed by vane-tip friction, vane-bending stress, and vane-length limitations. Maximum vane tip speeds are approximately 65 ft/s (20 m/s). Normally 6 to 8 vanes are used although as many as 20 vanes have been used. The trade-off is between reduced pressure differential per cell and increased friction. The eccentricity ratio is normally $e = 0.07D$ for low pressure (25 psig) (172 kPa) and $e = 0.05D$ for higher pressures (50 psig) (345 kPa). For service over 50 psig, the compressors are usually multi-staged.

Assuming an injected oil temperature of 140°F (60°C) and no inter-cooling, a two-stage vane air compressor operating at 100 psig will have an adiabatic efficiency (including mechanical losses) of 60 to 72 percent. The discharge temperature will be approximately 90 to 95°C or 200°F. Oil-flooded vane compressors normally use synthetic lubricants. The high shearing rates and vane tip temperatures rapidly oxidize petroleum based lubricants.

Rolling-Piston Compressors

Superficially, the rolling-piston compressor (Fig. 14.3.13) resembles the sliding-vane compressor. However, its kinematics and performance are quite different from those of the sliding-vane compressor. The rolling-piston compressor consists of an eccentric, a rolling piston, a vane, a cylinder, and a discharge valve. The motion of the rolling piston with

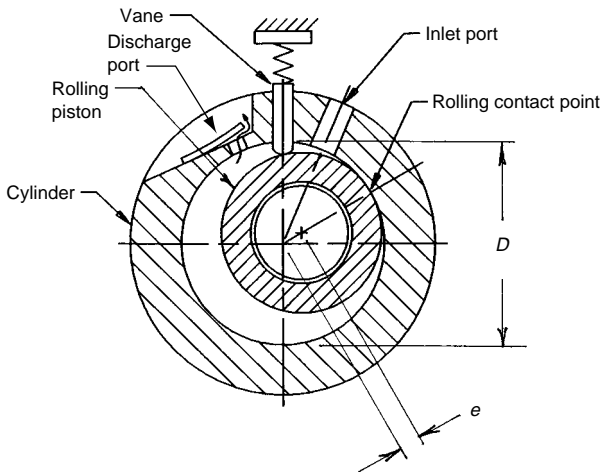


Fig. 14.3.13 Section through a typical rolling-piston compressor.

respect to the eccentric and cylinder is controlled by the various friction forces action on the piston. Compression and suction occur simultaneously in different chambers. A cycle covers two revolutions of the eccentric. Rolling-piston compressors are usually used in refrigeration systems, typically 0.25 hp (180 W) to 0.75 hp (550 W). Typical values for volumetric and power efficiencies are 0.90 and 0.84 to 0.86, respectively. The eccentricity is normally about $0.09 \times D$. The piston diameter is approximately $0.81 \times D$ and the length-to-diameter ratio is approximately 0.5. The compressor displacement can be calculated from Eq. (14.3.30) with $N_v = 1$.

Rotary, Twin-Screw, Oil-Flooded Compressors

Figure 14.3.14 is a cross section of an oil-flooded, rotary-screw compressor. The principle of operation can be determined from Fig. 14.3.15. The displacement of a twin-screw compressor is

$$V_{th} = KD^3 \frac{L}{D} C_{th} N \quad (14.3.32)$$

where $K = N_m(A_{mg} + A_{fg})D^2$; D = the rotor diameter; L = the rotor length; N_m = the number of lobes on the main rotor; A_{mg} = the area of one groove on the main rotor; and A_{fg} = the area of one groove on the secondary rotor. C_{th} is an overlap constant (between 0.95 and 1.0 for most designs) which corrects for the overlapping of the filling and compressing processes.

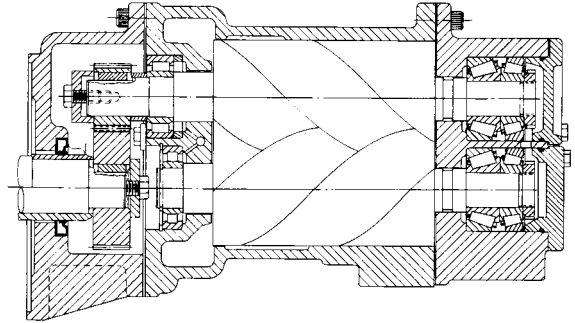


Fig. 14.3.14 Cross section of a typical single-stage, oil-flooded, twin-screw compressor.

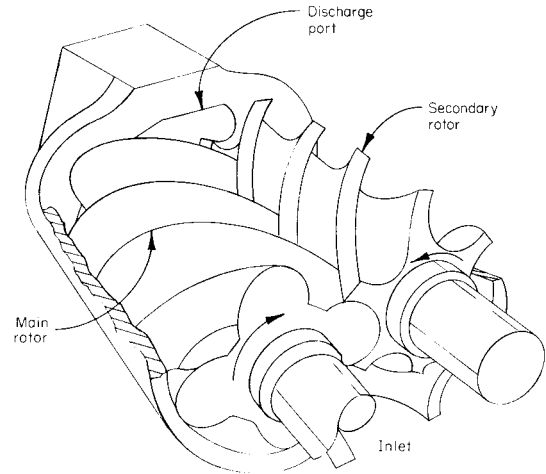


Fig. 14.3.15 Operating principle of a twin-screw compressor.

C_{th} depends on the wrap angle, the profile, and the number of lobes. Lobe combinations of (4 and 5), (4 and 6), (5 and 6), and (5 and 7) are used. The most common design uses the (4 and 6) combination. Typical values for C_{th} and K are given in Table 14.3.2. A complete cycle (filling and compressing) takes place over approximately 750° of main rotor rotation. There are four cycles per revolution of the male rotor. The discharge process for adjacent grooves overlap for approximately 40° giving an inherently smooth discharge process.

Rotary-screw compressors have a built-in volume ratio V_i (typically 4.4 for single-stage air compressors and 1.9 to 3.5 for refrigeration compressors). For an inlet pressure of P_1 , the compressor will have a built-in pressure of $P_1 V_i^\gamma$. The effect on the P-V diagram and on the theoretical power can be found from Fig. 14.3.12 and Fig. 14.3.16. In practice, the increased discharge port losses associated with higher

built-in volume ratios (smaller port area) skew the results in Fig. 14.3.16. In practice, it is more efficient, at least in air compressors, to have the built-in ratio slightly lower than the theoretical ratio. Single-stage, twin-screw air compressors rarely have built-in pressure ratios

Table 14.3.2 Typical Constants for Different Lobe Combinations (300° Male Wrap)

N_m/N_f	3/4	4/5	4/6	5/6	5/7
C_{th}	0.954	0.973	0.969	0.984	0.980
K	0.530	0.439	0.507	0.454	0.468

greater than 9, and most are designed with ratios of 7 to 8. An air compressor designed for 100 psig can operate at 175 psig with a loss in adiabatic efficiency of approximately 5 and 4 percent loss of volumetric efficiency.

Adiabatic efficiency (including mechanical losses in the air end) vary from 70 percent for small sizes (80 mm) to 88 percent for large (300 mm) slow-running compressors. An average value for a midsize compressor is 0.78 to 0.75 based on an injection temperature of in-

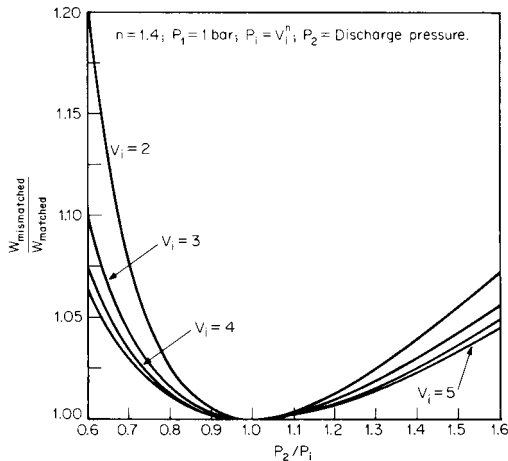


Fig. 14.3.16 The effect of the built-in volume ratio on theoretical work of compression for a compressor operating off the design point.

let temperature plus 50°F. Accuracy in manufacturing the rotors is critical to achieving good efficiency. The heat absorbed by the cooling oil can be estimated by:

$$Q_h = \text{power} \left[1 - \frac{\eta_c(T_2 - T_1)}{T_1(r^{(k-1)/k} - 1)} \right] \quad (14.3.33)$$

where power = the shaft power to the compressor. T_2 is normally 70 to 90°F over T_1 .

Tip speeds range from 15 to 50 m/s. Rotor sizes range from 40 to 400 mm diameters with L/D from 1 to 2. Oil-flooded screw compressors dominate the portable-compressor market and share the plant-air market about equally with the reciprocating compressor up to about 3,000 cfm. In the refrigeration industry, the screw compressor finds its widest application in the 300 to 500 kW range.

In heat-pump applications, the screw compressor's zero-clearance volume is an advantage over reciprocating compressors. Reciprocating compressors lose volumetric efficiency as the outside temperature drops (the compression ratio increases) requiring supplemental heat to compensate for the lost capacity. Refrigeration screw compressors are now available with a secondary suction port known as an **economizer connection**. This allows for two-temperature-level operation. At full-load operation, the system COP is improved significantly. As the compressor is unloaded, the improvement is lost.

Capacity can be controlled by on-line/off-line operation, inlet throttling, slide-valve or turn-valve operation, or speed regulation. Stationary air-compressors are normally controlled by on-line/off-line operation or inlet throttling with unload below 50 percent capacity.

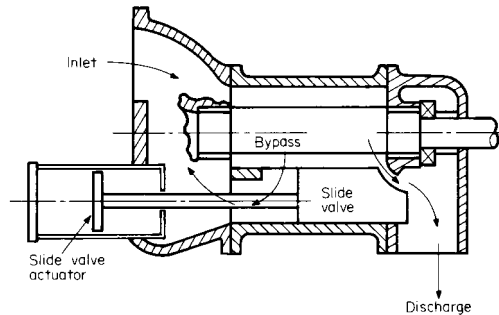


Fig. 14.3.17 Principle of slide-valve operation. As the valve is moved toward the discharge end of the compressor, the point of inlet cutoff is delayed.

The air-oil sump is usually blown down to atmospheric pressure at unload.

Portable compressors are normally controlled by a combination of inlet throttling and speed regulation; the air-oil sumps are not blown down.

Refrigeration compressors are controlled by inlet throttling or more often by a slide valve or turn valve. A compressor with a slide valve is shown in Fig. 14.3.17. The effect on power consumption for each type of capacity control can be found from Fig. 14.3.18.

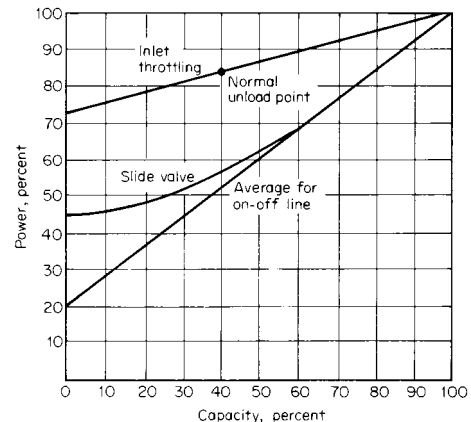


Fig. 14.3.18 Comparison of different capacity control methods for oil-flooded screw compressors operating at constant speed.

Oil-flooded, twin-screw compressors have been operated successfully on petroleum-based oils, automatic transmission fluids, synthetic diesters, polyalphaolefins, synthesized hydrocarbons, and polyglycols. The trend is to synthetic lubricants.

Regardless of the lubricant selected, the discharge temperature for an air compressor must be kept high enough to prevent condensation in the lubricant during part-load operation. This is usually accomplished by a thermal valve set to maintain the minimum injection temperature at 140 to 165°F (60 to 76°C). Standard oil separators limit oil carryover to 5 ppm or less. Optional treatment can reduce the levels further.

The capacity loss of an oil-flooded, twin-screw compressor with altitude is quite small compared to a normal reciprocating compressor. The approximate capacity correction for altitude can be read from Fig. 14.3.19.

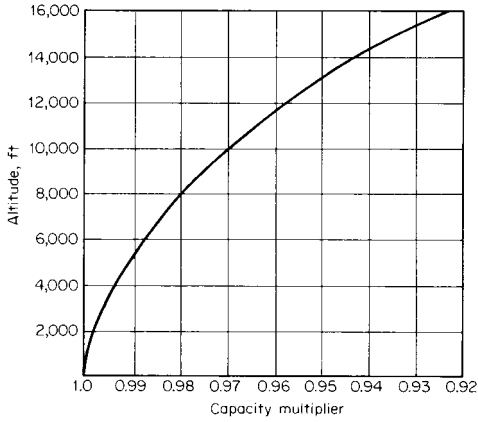


Fig. 14.3.19 Altitude correction for capacity, single-stage, oil-flooded screw compressor.

Rotary Single-Screw Compressors

The operating principle of the single-screw compressor can be determined from Fig. 14.3.20. Lundberg and Glanvall give the displacement as:

$$\beta = \arcsin \frac{B - D/2}{R}$$

$$\Omega = \pi - 2\beta$$

$$C = 2R \sin \frac{\Omega}{2}$$

$$\gamma = \frac{\alpha\Omega}{\xi}$$

$$t = R - \frac{B - D/2}{\sin(\beta + \gamma)}$$

$$u = B - \frac{D}{2} + \frac{t}{2} \sin(\beta + \gamma)$$

$$V = \sum_{\alpha=\alpha_1}^{\xi} AU \Delta\alpha$$

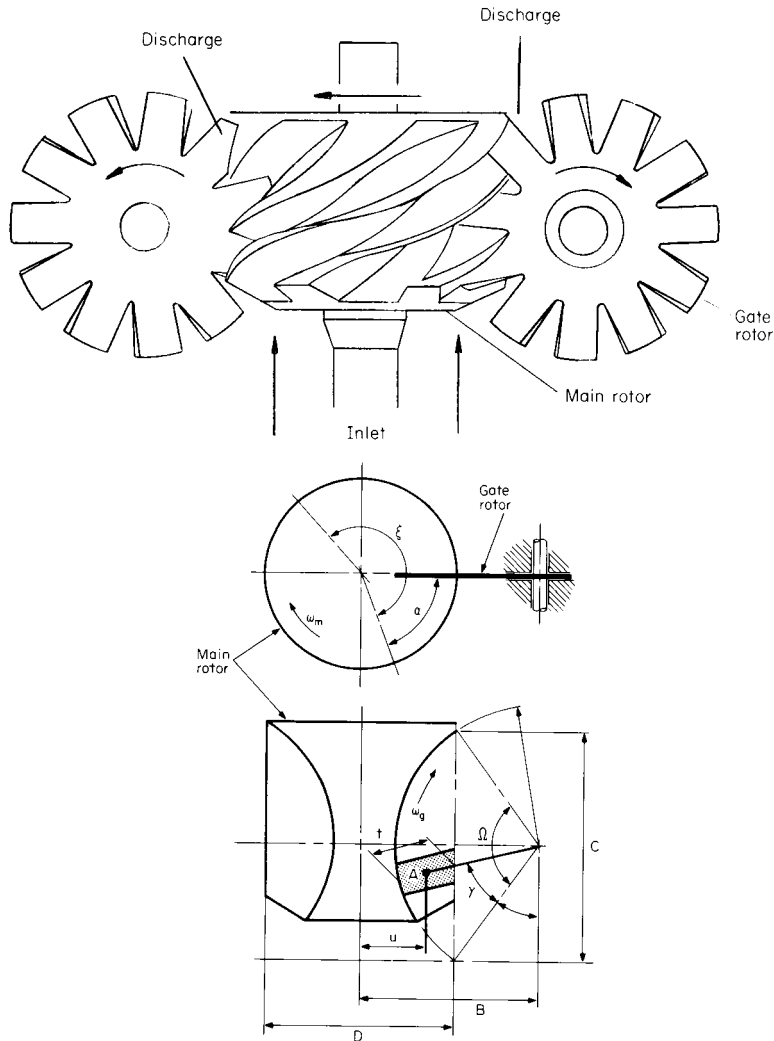


Fig. 14.3.20 Principle of operation and geometry of a single-screw compressor. (Lundberg and Glanvall, "A Comparison of SRM and Globoid Type Screw Compressors at Full Load," Proceedings of the 1978 Purdue Compressor Technology Conference, Purdue University, West Lafayette, IN.)

The rotary single-screw compressor is an oil-flooded compressor with a built-in volume ratio. The compressions for each star wheel are almost in phase so the gross effect as far as the discharge piping is that there are six discharges per revolution of the main rotor.

The single-screw compressor is used in the refrigeration industry where the low bearing load on the main rotor is an advantage. It is also being used as an air compressor in the smaller sizes.

Adiabatic efficiency of a single-screw compressor will be 2 to 5 percent lower than that of a comparable twin-screw compressor. The effect of off design operation on theoretical power can be found from Fig. 14.3.16. Capacity control is identical to the twin-screw compressor.

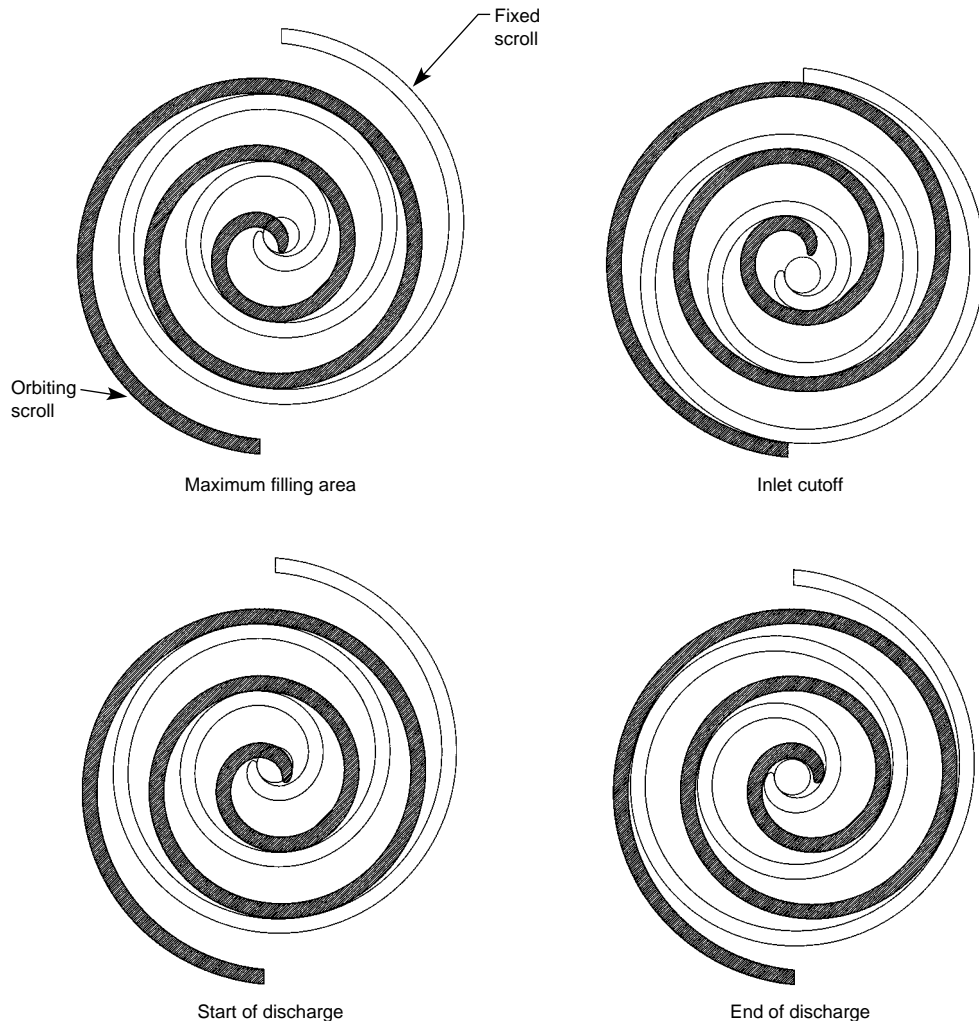
Dry Rotary Twin-Screw Compressors

A dry rotary, twin-screw compressor has timing gears to separate the rotors which may or may not be coated. The typical compressor is water-jacketed. Seals prevent gas leakage along the shafts and seal the oil to the bearings and timing gears. Sleeve bearings are most common, but some units have antifriction bearings. The displacement is the same as for the oil-flooded screw. Tip speeds vary from 80 to 120 m/s. Maximum compression ratio per stage is 4.5 based on $k = 1.4$. Sizes range from about 400 to 20,000 cfm (680 to 34,000 m³/h). Pressures range

from 15 to 180 psig (1,240 kPa). Two-stage compressors are almost always intercooled. With intercooling to 15°F they approach adiabatic efficiency (based on zero intercooling) including mechanical-loss ranges from 72 to 80 percent. While oil-flooded screw compressors can be controlled to very low capacities by inlet throttling, dry screw compressors cannot. The resulting pressure ratio causes excessive temperature increase across the compressor.

Orbiting Scroll Compressors

The primary elements of the orbiting scroll compressor are the fixed scroll, orbiting scroll, Oldham ring, crankshaft, and crankcase. Typically the two scrolls are involutes of circles phased 180° apart. The orbiting scroll is driven by the crankshaft and prevented from rotating via the Oldham ring (Fig. 14.3.21). As the scroll orbits, the pockets formed between the flanks of the scrolls move toward the center reducing in size until the built-in volume ratio is achieved. At this point two pockets start discharging simultaneously through the discharge port. The filling, compression, and discharging process usually takes 900 to 1,100 degrees of shaft rotation, depending on the number of wraps. The compressor displacement is a function of the involute generating radius, the starting angle, the thickness angle, and the scroll height. The crank



radius is equal to 0.5 times the involute pitch minus the scroll thickness. The scroll compressor is used primarily as an air-conditioning or heat-pump compressor in the 5 to 50 hp (3.7 to 37 kW) range.

Dynamic Compressors

A typical centrifugal compressor is shown in Fig. 14.3.22. The optimum range of operation can be found from Fig. 14.3.23. An estimate of the polytropic efficiency can be obtained from Fig. 14.3.24 or calculated from

$$\eta_p = 0.014 \ln(Q) + 0.600 \quad (14.3.34)$$

where Q is in cubic feet per minute.

For normal industrial compressors, $H_p = 10,000 \text{ ft} \cdot \text{lb/lb}$ (30 kJ/kg) per stage although much higher values can be obtained.

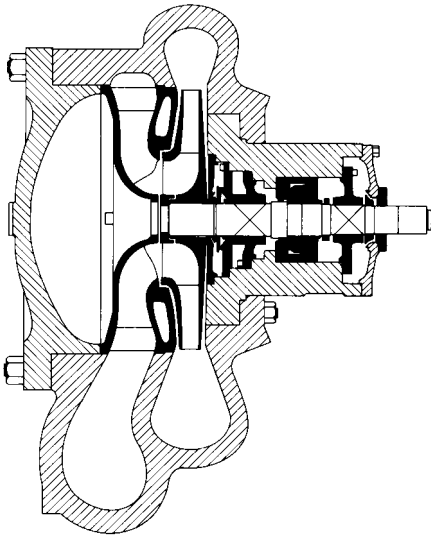


Fig. 14.3.22 Single-stage, cantilever-design centrifugal compressor.

The required speed can be estimated from:

$$N = \frac{(g_c H_p / \mu)^{0.5}}{\pi D} \quad (14.3.34)$$

The pressure coefficient ($\mu = g_c H_p / U^2$) varies from 0.4 to 0.7 (in the high-efficiency range). In the absence of specific data, use 0.55 for estimating.

The diameter is determined by the required flow and can be estimated from the flow coefficient $\phi = Q/ND^3 \propto V/U$, where V is the axial inlet velocity. The normal range of ϕ is 0.15 to 0.4 with the optimum at 0.3.

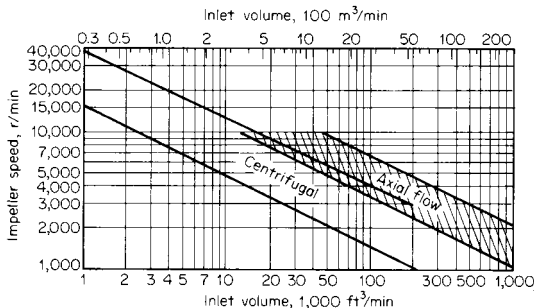


Fig. 14.3.23 Range of speed versus inlet capacity for near-optimum dynamic compressor efficiency. (Loomis, "Compressed Air and Gas Data," 3d ed., Ingersoll-Rand, pp. 4-28.)

The slope of the $\mu-\phi$ curve (and the $H_p \cdot Q$ curve) is strongly dependent on the backward lean of the impeller blades as shown in Fig. 14.3.25. $\beta_2 = 90^\circ$ is a radial blade. Normal closed industrial impeller blades will have an angle of 65 to 55°. It should be noted that an increase in density of the gas will flatten the slope.

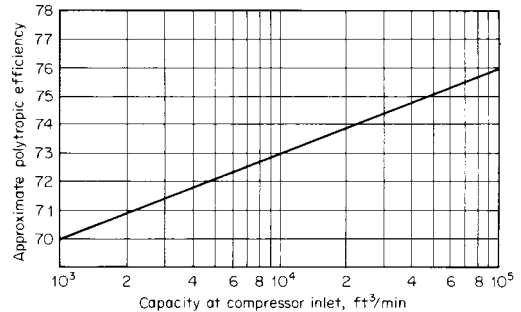


Fig. 14.3.24 Approximate polytropic efficiency versus compressor inlet capacity. (Rollins, "Compressed Air and Gas Handbook," 4th ed., Compressed Air and Gas Institute, pp. 3-71.)

A typical axial compressor is shown in Fig. 14.3.26. The range of application can be found from Fig. 14.3.18. Axial compressors are more efficient than centrifugals (as much as 5 percent) but have a much narrower range of operation. The surge limit will vary from 85 to 95 percent of design flow. For a given stage, the polytropic head will be about one-half that developed by a centrifugal stage, i.e., $\mu = 0.3$. Axial compressors are used only for air or clean gas. They find their major application in refineries, butadiene plants, and ethylene oxide plants. Jet-engine compressors are axial-flow compressors.

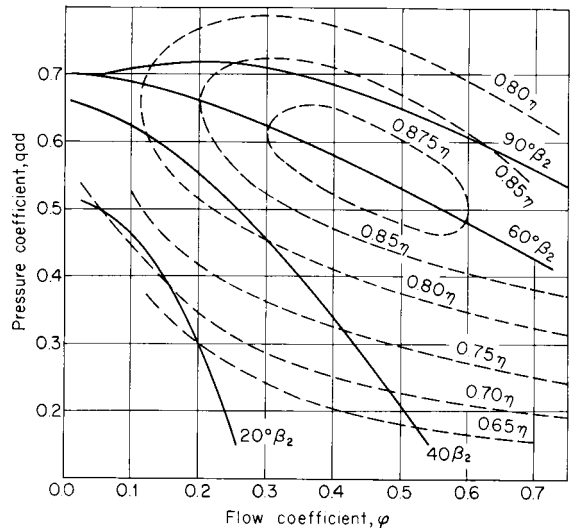


Fig. 14.3.25 Basic characteristics of centrifugal compressor, showing effects of backward-leaning impeller blades. (After Balje, ASME 51-F-12.)

Thrust Pressures

A double-inlet and an open impeller have no axial thrust, whereas a semienclosed impeller, with a frontal shroud, can impose substantial thrust, evaluated by:

$$T_A = Y(A_1 - A_2) \Delta \text{ (lb/in}^2\text{)} + 0.8(A_2 - A_3) \Delta \text{ (lb/in}^2\text{)} - P_1 A_3$$

where A_1 = full impeller area; A_2 = impeller area less the area of the inlet-eye seal; A_3 = shaft area through the seal, all in in²; and

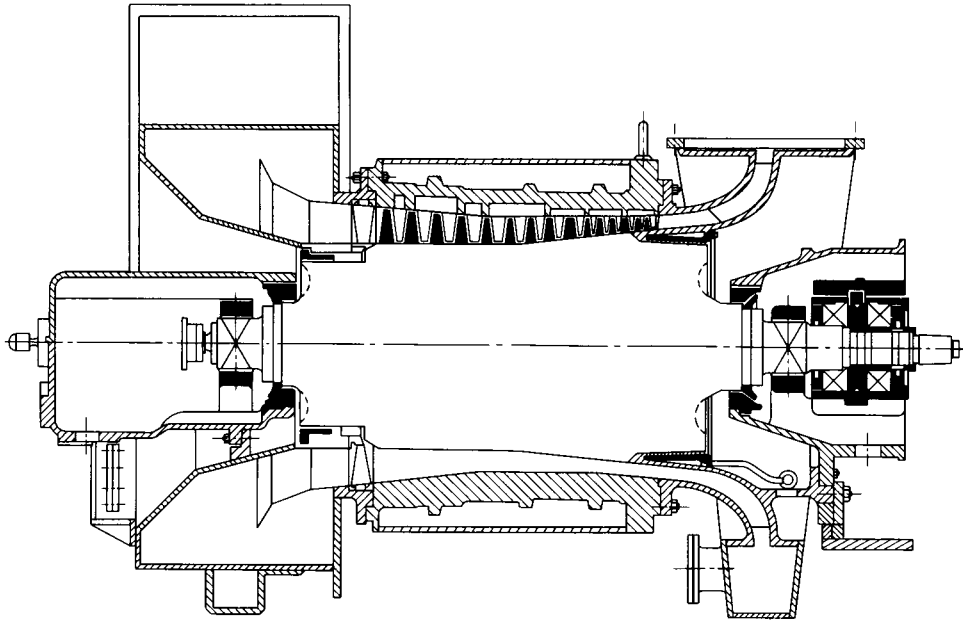


Fig. 14.3.26 Multistage axial compressor.

Δ (lb/in²) = pressure differential per stage. Y is the percentage of Δ (lb/in²) acting in back of a single-stage disk: (1) Without a frontal shroud and a plain back, $Y = 0.35$; (2) with 0.060-in back ribs, $Y = 0.28$; (3) with $\frac{1}{4}$ -in equalizer holes at half radius and about 2 in apart, $Y = 0.22$; and (4) with deep scallops, $Y = 0.18$. For fully shrouded impellers, as used in multiple stage, $Y = 1.00$. These thrusts are balanced by opposing impeller inlet ends or, more generally, by a balancing drum wherein an equal and opposite thrust is created, equal to the

product of the compressor Δ (lb/in²) and the area of the drum. Step-anoff ("Turboblowers," Wiley) shows that radial impellers create a radial thrust, caused by the uneven volute-pressure distribution, which is:

$$T_R = 0.36 \Delta \text{ (lb/in}^2\text{)} db$$

where d = the impeller diameter; and b = the width, including the shrouds, both in inches.

14.4 HIGH-VACUUM PUMPS

by Benjamin B. Dayton

REFERENCES: Dushman, "Scientific Foundations of High Vacuum Technique," 2d ed., Lafferty (ed.), Wiley. Roth, "Vacuum Technology," North-Holland. O'Hanlon, "A User's Guide to Vacuum Technology," Wiley. Power, "High Vacuum Pumping Equipment," Reinhold. Van Atta, "Vacuum Science and Engineering," McGraw-Hill. LaPelle, "Practical Vacuum Systems," McGraw-Hill. Holland, Steckelmacher, and Yarwood, "Vacuum Manual," E. & F. N. Spon. M. H. Hablanian, "High-Vacuum Technology," Marcel Dekker.

The pressure of gas in a chamber at a given temperature can be reduced by allowing the gas to escape through a port into a vacuum pump, or by a pumping system comprising two or more pumps in series, which compresses the gas and discharges it into the atmosphere, or by allowing the gas (or vapor) to condense on a sufficiently cold surface or to react with a chemically active surface exposed within the chamber or within an appendage to the chamber. The movement of the gas toward the pump or trapping surface can be accelerated by heating the walls of the chamber or by ionizing the gas and applying an electric field. **Rotary mechanical pumps and steam ejectors** which can compress the gas from about 10^{-3} torr up to atmospheric pressure are described earlier in Secs. 9 and 14. Pumps and trapping techniques required to produce pressures in the "high-vacuum region" (below 10^{-3} torr) are described below.

Units The term *pascal* (abbreviated Pa and equal to one newton per

square meter) is recommended as the SI unit for the pressure of the gas in vacuum systems, but the unit *torr* (equal to 133.322 Pa) is still widely used in equipment catalogs and published articles. The ISO also permits the use of bar (10^5 Pa) and millibar (mbar), which are convenient units for vacuum technology (one torr equals 1.333 millibar). **Pumping speed** is normally expressed in litres per second (L/s) or cubic feet per minute (ft³/min); one litre per second equals 2.12 cubic feet per minute.

SELECTION OF PUMPS

The type of vacuum pump selected depends primarily on the lowest pressure to be attained and the possible effects of vapor contamination. The lowest pressure that the pump itself can attain without vapor traps is called the **ultimate pressure** of the pump. The **size** of the pump depends either on the time to "pump-down" from atmospheric pressure or on the pressure to be maintained in the presence of a given gas load during the process cycle and the number of pumps which can be installed in parallel. The gas load is usually expressed in terms of **throughput** defined as the product of the static pressure and the volumetric flow rate across a given section at a given temperature. Size of pump is rated in terms of volumetric pumping speed (L/s or ft³/min) at the inlet pressure

for which the speed is a maximum. As shown in Fig. 14.4.1, for rotary-piston mechanical pumps this maximum speed occurs at atmospheric pressure, but for steam ejectors and oil vapor ejectors (similar in principle and design to steam ejectors but employing water cooling on the venturi section or diffuser) the maximum speed occurs at the top of a narrow peak on the speed-pressure curve. The **diffusion pump**, which employs one or more jets of vapor into which molecules from the chamber can diffuse and be carried forward into a region of higher gas pressure, has a broad plateau of maximum speed (see Fig. 14.4.1) from

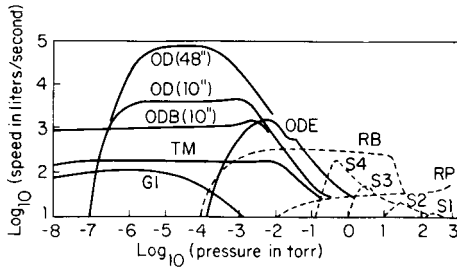


Fig. 14.4.1 Variation of pumping speed with inlet pressure of typical vacuum pumps. OD(48 in) = oil diffusion pump with nominal 48-in-diam inlet flange; OD(10 in) = 10-in oil-diffusion pump; ODB(10 in) = 10-in oil-diffusion pump with baffle; ODE = oil-diffusion-ejector pump; TM = turbomolecular pump; GI = getter-ion pump (sputter type); RB = Roots-type blower; RP = rotary-piston, oil-sealed mechanical pump; S1, S2, S3, and S4 = matching stages in series of four-stage steam ejector.

about 10^{-3} torr (1 mtorr) to pressures 20 times the ultimate pressure, the latter being approximately equal to the vapor pressure at room temperature of the fluid used to form the vapor jet. Placing a cooled baffle between the diffusion pump and the vacuum chamber reduces the net pumping speed but allows the attainment of lower pressures by condensing the pump-fluid vapor. The turbomolecular pump, employing many turbine blades rotating between slotted stator disks, also has a broad pumping range up to about 0.01 torr, as shown by curve TM in Fig. 14.4.1.

TYPES AND SIZES

The number, types, and sizes of vacuum pumps in series required to compress gas from the chamber up to atmospheric pressure usually depends on the compression ratio per pumping stage at the maximum throughput during the process cycle. For **ultrahigh vacuum** systems (pressures less than 10^{-9} torr) the number of pumps or stages required to reach the lowest pressures may depend on the rate of back leakage or back diffusion through the pumps at the minimum throughput. **Compression ratio** is defined as the ratio of the outlet (exhaust or discharge) pressure to the inlet (or intake) pressure at a given throughput when

pumping a gas or vapor that is not absorbed or condensed within the pump. This ratio is a variable quantity which depends in general on the ratio of the pumping speed of a given stage to the net speed of the fore pump, or next stage of compression. However, for a given throughput there is an upper limit to the compression ratio which can be maintained across a vapor jet pumping stage at a given power input. At maximum throughput under normal operation the limiting compression ratio for single-stage ejector pumps is about 10, and for the individual stages of multistage diffusion pumps the limiting ratio is about 4. Thus, a three-stage steam ejector is required to compress air from 0.8 to 760 torr, a ratio of about 10^3 . Oil diffusion pumps usually have from three to five stages, and a four-stage diffusion pump can compress air from 1.5 mtorr to 0.4 torr at maximum full-speed throughput, corresponding to a total compression ratio of about 4^4 or 256. The overall compression ratio can be much higher at very low throughputs, but the **forepressure** (pressure at the outlet) should not be allowed to exceed a value known as the **limiting forepressure** or **tolerable forepressure**, which is usually in the range of 0.2 to 0.6 torr for a four-stage diffusion pump.

Oil-sealed mechanical piston pumps can be operated with high compression ratios over a pressure range from atmospheric pressure (or above) to about 1 mtorr. Molecular drag, centrifugal, rotary blower, e.g., Roots-type, pumps, and turbomolecular pumps are limited by back flow or leakage of gas from the forepressure side through the clearances between the rotor and stator, and therefore require backing pumps to reduce the forepressure from atmospheric to a sufficiently low value to permit a high vacuum to be obtained on the inlet side. Oil vapor-jet pumps cannot discharge gas directly to the atmosphere because of limitations on the vapor pressure which can be generated without excessive thermal decomposition of the oil, and backing pumps must therefore be used to reduce their forepressures to values less than the vapor pressure of the oil at a safe temperature in the boiler. Some gas molecules (particularly hydrogen and helium) may succeed in diffusing back through a vapor jet from the forevacuum to the high vacuum, and for ultrahigh vacuum systems this back diffusion must be reduced by using a sufficient number of pumping stages and vapor jets with high vapor density and velocity.

Table 14.4.1 lists types, sizes, and operating characteristics of typical commercially available pumps. The maximum (standard measured) speed (in litres per second) of oil diffusion pumps is about $18D^2$ for three-stage models and $28D^2$ for four-stage models, where D is the true inlet diameter (in inches). The larger pumps are usually water-cooled, and about 80 percent of the heat generated must be carried away by the cooling water, the remainder being lost by convection, radiation, and conduction to the surroundings. The exit temperature of the cooling water should usually not exceed 40°C , and the required rate of flow in cubic centimeters per minute of water having an inlet temperature of 20°C and an exit temperature of 30°C is about numerically equal to the power input in watts. This rule of thumb applies to either diffusion pumps or water-cooled mechanical pumps where the horsepower of the motor is converted to watts by multiplying by 746.

Table 14.4.1 Types and Available Sizes of High-Vacuum Pumps

Type	Nominal inlet ID, in	Maximum air speed L/s	Starting pressure, torr	Ultimate pressure, torr	Power input, kW
Rotary piston	0.5–8	0.1–400	(atm)	$\sim 10^{-3}$	0.2–30
Blower (Roots)	2–24	40–13,000	30–100	$\sim 10^{-3}$	0.5–70
Turbodrag (hybrid)	2–12	60–2,000*	5–10	$\sim 10^{-8}$	0.1–2.6
Turbomolecular	2–26	50–10,000*	~ 0.1	$< 10^{-9}$	0.4–6
Diffusion (vapor)	2–48	100–90,000	~ 0.5	$< 10^{-9}$	0.3–30
Diffusion ejector	12–24	4,000–12,500	2–5	$\sim 10^{-4}$	6–23
Getter-ion (sputter)	0.3–36	1–50,000	~ 0.02	$< 10^{-10}$	0.1–50
Sorption	1–2	1–80	(atm)	$\sim 10^{-3}$	0.4–2
Cryopump	4–48	400–57,000*	$\sim 10^{-3}$	$< 10^{-10}$	1–8
Nonevaporable getter	3–8	150–1600†	$\sim 10^{-4}$	$< 10^{-10}$	0.2–1.8

* Nitrogen speed.

† Hydrogen speed.

1 torr = 133.32 Pa.

Diffusion Pumps Figure 14.4.2 shows in cross section a typical vacuum system with a diffusion pump A having three stages (annular nozzles forming vapor jets). The operating fluid, which is usually a low-vapor-pressure organic or silicone oil of molecular weight ranging from 350 to 500, is located in the bottom of the metal casing and is

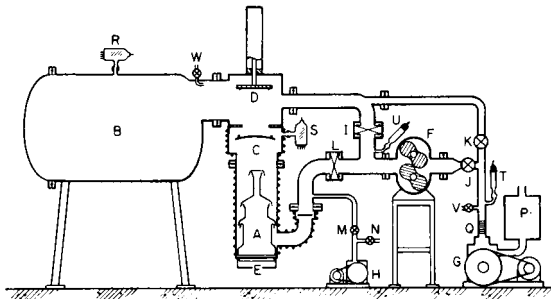


Fig. 14.4.2 Typical high-vacuum system. A = diffusion pump; B = chamber; C = baffle; D = valve; E = pump heater; F = Roots-type blower; G = roughing (mechanical) pump; H = holding (mechanical) pump; I to N = valves; P = oil reservoir and separator; Q = bellows; R, S = ion-gage tubes; T, U = Pirani-gage tubes; V, W = air-inlet valves.

heated by an external electrical heating unit E to about 200°C to generate the vapor. Mercury can be used as the pump fluid, but efficient refrigerated traps are required to keep the mercury vapor out of the vacuum chamber B. Oil diffusion pumps require some form of purging or purification of the oil during use to eliminate dissolved gases and volatile decomposition products. **Fractionating oil diffusion pumps** purify the oil by circulating it through a series of boilers, or boiler compartments, feeding vapor through separate chimneys to the various nozzles in a multistage pump. The volatile impurities are ejected with the vapor feeding the stages nearest the fore vacuum, and the purged oil of lowest vapor pressure supplies the top nozzle from which vapor molecules scattered back out of the inlet port create a partial pressure of oil vapor in the high vacuum which limits the ultimate pressure obtainable without cold traps [Hickman, *J. Appl. Phys.* **11**, 303 (1940)].

Getter-ion pumps or **sputter-ion pumps** employ chemically active metal layers which are continuously or intermittently deposited on the wall of the pump by either thermal evaporation or sputtering and which chemisorb oxygen, nitrogen, water vapor, and other active gases while the inert gases such as helium, neon, and argon are "cleaned up" by ionizing them in an electric discharge and drawing the positive ions to the wall where the neutralized ions are buried by fresh deposits of metal. These pumps require a roughing pump to reduce the pressure to less than about 20 mtorr at which point the active metal (usually titanium) can be evaporated or sputtered at the required rate, but after they begin operation no backing pump is required since all of the gas is trapped at the wall. When isolated by valves from the roughing pump, the getter-ion pumps form an enclosure sealed to the vacuum system so that a power failure cannot result in leakage of atmospheric air or vapors from a forepump into the system. Sputter-ion pumps can operate continuously for more than 1 year at pressures below 10^{-6} torr.

Cryopumps consist of one or more exposed surfaces refrigerated to a temperature usually below 100 K, at which certain gases will be condensed and form a layer having an equilibrium vapor pressure below a specified limit. Figure 14.4.3 shows a cryopump with plate arrays cooled to 15 K and 80 K directly by a mechanical cryocooler, usually based on the Gifford-McMahon cycle (Sec. 19.2) and charged with helium gas. Activated charcoal is fixed inside the 15 K array panels. Cooldown time is about 1 to 3 h. For vacuum systems to be cycled to atmospheric pressure, a high-vacuum valve should be placed over the cryopump inlet and closed when venting the system. The condition for again opening the valve is that the product of system volume and pressure in torr-litres should be slightly below the manufacturer's rated crossover limit (ranging from about 40 to 500 depending on the pump

size). When the cryopanel becomes saturated with thick condensed layers and the charcoal needs reactivation, the inlet valve is closed and the cryocooler switched off while the pump is allowed to warm up and purged with warm dry nitrogen. A pressure-relief valve is incorporated in the cryopump to protect against too-rapid evaporation of the condensed gases, and since hydrogen gas may be released first there should be no object present which might ignite the gas mixture. A plate cooled to 20 K, by circulating helium gas from a refrigeration unit through coils attached to the plate, will condense N_2 , CO_2 , CO , H_2O , O_2 , Ar , and Xe to maintain partial pressures of these gases less than 10^{-10} torr.

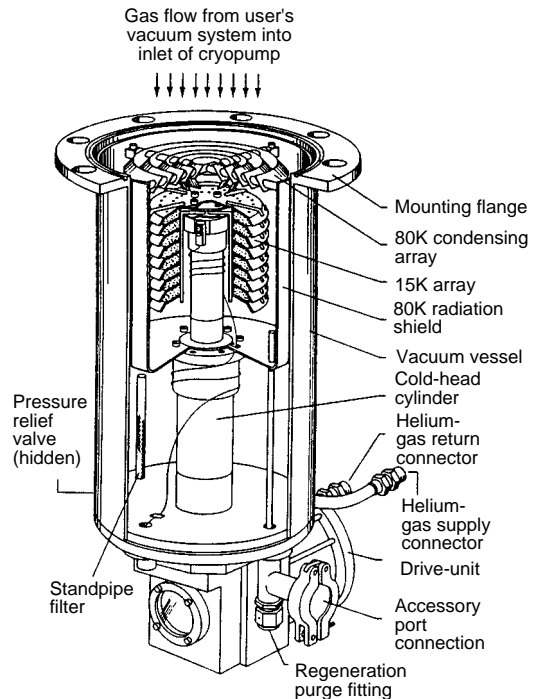


Fig. 14.4.3 Cryopump with two-stage mechanical refrigeration unit. (Courtesy CTI-CRYOGENICS.)

Hydrogen, helium, and neon are not adequately condensed at 20 K but may be "cryotrapped" in a deposit of H_2O and other gases condensing on the plate. Cryopumping is used in space simulation chambers to create the necessary low pressures and to act as a heat sink comparable to "cold black space." In this application the 20-K plates are shielded by liquid-nitrogen-cooled panels, and the whole array is shaped to cover the inside wall of the vacuum chamber.

Sorption pumps employ a sorbent such as activated charcoal or synthetic zeolite (Molecular Sieve) cooled by liquid nitrogen or other refrigerant. They can be used to rough a system down from atmospheric pressure to a few millitorr at which getter-ion pumps may begin operation, or an additional preconditioned sorption pump can be valved in to reduce the pressure to 10^{-5} torr or less.

Nonevaporable getter pumps employing cartridges or strips of zirconium-aluminum alloy or zirconium-vanadium-iron alloy mounted on flanges with central heating elements can be directly inserted in the wall of vacuum systems or enclosed in a separate housing attached to the system. These pump all active gases, especially hydrogen, which is the common residual gas in ultrahigh-vacuum systems. When very high pumping speed is required, the getter is heated above room temperature to promote diffusion of gas into the bulk material. The Zr-Al alloy must first be activated by heating for about 45 min to 700°C, but the Zr-V-Fe alloy needs to be heated to only about 450°C. Starting pressure is less than 10^{-4} torr and ultimate pressure for active gases can be less than

10^{-10} torr. Regeneration of the getter by heating is periodically required. The pumping of hydrogen is reversible by controlling the temperature, and an application is the pumping, storing, and releasing of hydrogen isotopes.

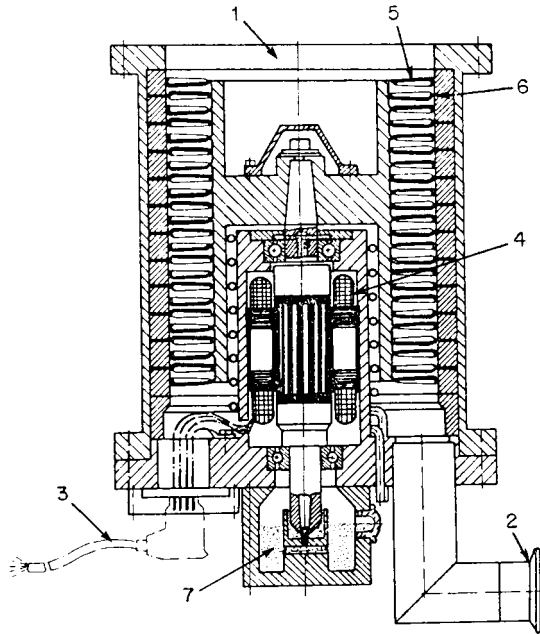


Fig. 14.4.4 Vertical turbomolecular pump. (1) Inlet flange; (2) outlet flange; (3) cable to frequency-converter power supply; (4) high-speed, low-voltage, three-phase motor; (5) rotor blades; (6) stator assembly; (7) lubricating oil reservoir. (Reprinted from "Vacuum Manual," copyright 1974 by E. & F. N. Spon, London.)

Turbomolecular pumps employ a rotor with multiple disks made with slots or formed into turbine blades revolving at 15,000 to 50,000 r/min (depending on diameter) in a cylindrical housing stacked with annular stator disks fitting between the rotor disks and also provided with slots or blades inclined at a reverse angle. In the horizontal-axis, dual-disk

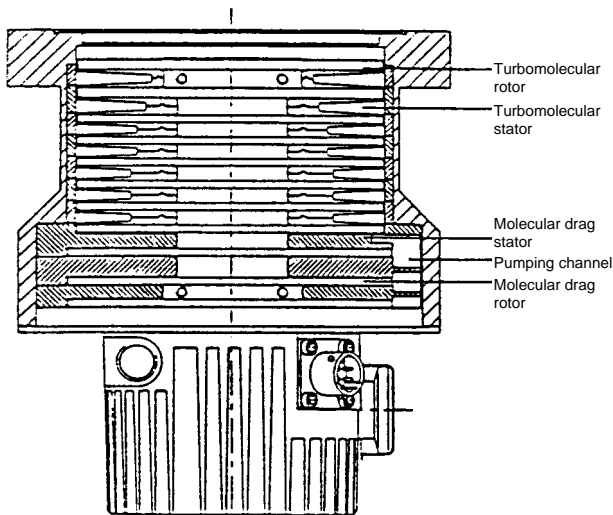


Fig. 14.4.5 Hybrid turbomolecular pump with molecular drag stages. (Reprinted with permission from the *Journal of Vacuum Science and Technology*, vol. 11, p. 1616, 1993.)

assembly turbopump (U.S. Patent 2,918,208), the gas enters through a large central port and then flows to the left or right through separate disk assemblies parallel to the axis. In the vertical single-disk assembly turbopump (Fig. 14.4.4), the gas enters through a protective screen in a large inlet port (1) and is impelled downward parallel to the axis by rotor blades (5) with tip speeds of about 300 m/s. Turbopumps with a magnetically suspended rotor can usually be cooled by ambient air, but pumps using oil- or grease-lubricated rotor bearings require water or forced-air cooling. Grease-lubricated pumps can be mounted in any orientation. The compression ratio increases rapidly with the molecular weight of the gas, and thus the vacuum produced is essentially free of vapors from lubricating oils or forepump fluid as long as the pump is operating. Shortly after switching a turbopump off, or after a power failure, dry gas should be flushed through the rotor assembly to prevent back migration of any oil vapor from the forepump. However, to ensure complete protection of the system from organic vapors, it is desirable to use turbopumps compounded with molecular drag stages or pumps with hybrid rotors employing a series of disk blades nested in grooved stators below the turbine blades, as shown in Fig. 14.4.5, which allow the operating exit pressure to be extended up to 10 torr so that oil-vapor-free "dry" mechanical backing pumps can be used. (M. H. Hablanian, *J. Vac. Sci. Technol. A*, **11**, p. 1614, 1993.)

INSTALLATION

In a typical assembly of pumps for a high vacuum system (Fig. 14.4.2) the diffusion pump A must be located close to the chamber B and should always have a water-cooled or refrigerated baffle C over the pump inlet so that vapor scattered back from the jets (backstreaming) will be condensed and returned as liquid to the pump. The pipe or manifold, which may include a valve D, connecting the chamber and the baffled pump should be of diameter equal to or larger than the inlet of the pump, and the length of the passage between the baffled pump and the chamber should preferably be not more than about 3 times the mean diameter of the passage. A space at least 6 in high should be allowed below the diffusion pump boiler E for easy servicing of the heaters, draining the pump fluid, or removal of the pump from the system.

Some mechanical pumps have appreciable vibration so that they should be firmly anchored or mounted on vibration damping pads, and flexible metal bellows Q should be installed in the forevacuum line. These bellows also aid in aligning the pumps during assembly. It is not necessary to have the mechanical pumps near the chamber because the resistance to gas flow of moderate lengths of pipe having the same diameter as the inlet of these pumps is not sufficiently large to create a serious pressure drop at mean pressures above 200 mtorr. Below 200 mtorr the pressure drop along the pipe is larger, but usually the diffusion pump operation is not affected unless the limiting forepressure (about 100 to 600 mtorr) is exceeded at the pump outlet.

VAPOR CONTAMINATION

The vapors evolved from vacuum systems may be condensed within mechanical pumps or the intercondensers of steam-ejector systems. If the condensed vapor is the same as the working fluid in the pump, such as water vapor condensed in "wet" vacuum pumps, no harm is done, provided that the fluid level is maintained at the optimum working value. If the condensed vapor can be separated from the working fluid by centrifuging, evaporation, filtering, or settling, suitable separating means can be installed. In certain types of rotary pumps the Gaede gas-ballast principle may be employed to avoid condensation by admitting air at a certain point in the compression cycle. If separation cannot be satisfactorily accomplished during operation of the pump, the vapor should be condensed in cold traps or liquid absorption columns before reaching the pump.

Baffles and traps are often required to prevent the backstreaming or migration of the pump operating fluid or sealing fluid into the high-vacuum chamber. Unless cooled baffles and traps or sorption traps for oil vapor are included in the connecting line, the ultimate pressure in the vacuum chamber will usually not be less than the vapor pressure of the

pump fluid at ambient temperature. While a water-cooled "optically tight" baffle installed above the inlet of a diffusion pump will condense the backstreaming pump-fluid vapor and return most of the fluid to the pump, some fluid will reevaporate from such a baffle and migrate back into the vacuum chamber. To reduce the partial pressure of pump-fluid vapor below the vapor pressure at the water cooling temperature it is necessary to add a cold trap or baffle refrigerated to temperatures usually below the pour point of the fluid. These cold traps must be periodically warmed and the condensate removed to avoid inefficient cooling of the exposed surfaces during operation.

FLOW OF GASES AT LOW PRESSURE

The pipe line between the high-vacuum pump and the vacuum chamber limits the volumetric flow so that the **net pumping speed** as measured by a vacuum gage located in the chamber is given by $S_n = S_0 U / (S_0 + U)$, where S_0 is the measured speed of the pump at its inlet and U is the **conductance** of the pipe defined by $U = Q / (P_n - P_0)$, where Q is the throughput, P_n is the pressure in the chamber, and P_0 is the pressure near the inlet of the pump as measured by a gage installed in a similar manner to that used to determine the pump speed S_0 . When there is no loss or gain of gas within the pipe line, $Q = S_n P_n = S_0 P_0 = U (P_n - P_0)$.

The conductance of a pipe depends on the geometry and the **Knudsen number** K (defined as the ratio of the mean free path of the gas molecules to the mean diameter of the cross section) as well as the direction and velocity of the molecules entering the pipe. For $K > 1$ the conductance for air in litres per second at 25°C of a circular pipe of length L (feet) and inside diameter D (inches) connecting a high-vacuum pump to a chamber of diameter greater than $3D$, including the "entrance correction" at the chamber, but neglecting the "exit correction" at the pump which depends on the inlet diameter and other factors, may be calculated from $U = 6.6D^3 / (L + 0.11D)$. The effect of right-angle bends in the pipe for "molecular flow" $K > 1$ and for $L > D/3$ may be approximated by adding $0.05D$ to L for each bend (where L is in feet and D in inches). A single right-angle bend in a short pipe ($L < D/3$) has practically no effect on the conductance as computed for a straight pipe of the same length along the centerline [Davis, *J. Appl. Phys.*, **5**, 358 (1954)].

When $K < 0.01$, the conductance in litres per second for air at 20°C of a long circular pipe length L (feet) and diameter D (inches) may be calculated from $U = 0.25D^4 \bar{P} / L$, where \bar{P} is the mean pressure in the pipe in millitorr (micrometers of Hg). For $0.01 < K < 1$ the conductance for air at 20°C of a long tube can be estimated from

$$U = \frac{0.25D^4 \bar{P}}{L} + \left(\frac{1 + 0.65D\bar{P}}{1 + 0.80D\bar{P}} \right) \left(\frac{6.6D^3}{L} \right)$$

The size of the primary pump, which acts as a "roughing pump" to pump the chamber down from atmospheric pressure to a pressure at which a diffusion pump or other high-vacuum pump can operate, may depend on the peak gas load during the process as well as the desired pump-down time. However, the size indicated by the peak load condition is frequently much smaller than the size required to meet the specified pump-down or roughing time. In this case, if the process cycle is much longer than the pump-down time, it is advisable to use two forepumps (primary pumps), a large one (G in Fig. 14.4.2) for roughing down and a smaller one (H in Fig. 14.4.2) for holding the vapor pumps during the roughing period and backing them during the processing period. Roots-type blowers are useful for shortening the roughing time for large chambers in the range below 20 torr and for handling unusually large bursts of gas which occur in some processes (F in Fig. 14.4.2).

An oil-sealed rotary mechanical pump is normally used as the primary pump, and the roughing time t_r (in minutes) required to evacuate a chamber of volume V (cubic feet) from atmospheric pressure to about 0.7 torr at which high-vacuum pumps begin to operate can be estimated

from $t_r = 10V/C$, where C is the rated speed (cubic feet per minute) at atmospheric pressure of the rotary pump. Since the speed of the high-vacuum pumps is usually of the order of 100 times that of the forepump, the pressure should drop quickly as soon as the high-vacuum pumps "take hold," but below 10^{-4} torr the pressure may begin to decrease more slowly because of the out-gassing of the materials exposed inside the vacuum system.

For most of the materials exposed in high-vacuum systems the **outgassing rate** can be assumed proportional to the exposed area A_m , although for some very porous materials the rate is more proportional to the bulk or mass. Except for evaporation from pure liquid or solid phases, the outgassing rate normally decreases with time when the temperature is constant. For many industrial-type vacuum systems it has been found that the pressure P in the chamber decreases approximately according to $P = P_u + K_1 A_m / S_n t^\alpha$ where P_u is the ultimate pressure, K_1 is a constant which may be considered equal to the average outgassing rate per unit area after 1 h of pumping, A_m is the exposed area, t is the total pumping time in hours, and α is an exponent which is usually nearly constant for the first few hours. For rough calculations in typical metal vacuum systems it may be assumed that $\alpha = 1$, but for systems containing large amounts of elastomers or plastics α may be closer to 0.5 [Dayton, *Trans. 6th Natl. Vacuum Symp.*, pp. 101–119, Pergamon Press, Oxford, 1960; Kraus, *ibid.*, pp. 204–205].

When the system is first evacuated after prolonged exposure to the atmosphere, most of the outgassing load for the first 10 to 100 h is usually water vapor, and for unbaked metal systems the numerical value of K_1 is of the order of 10^{-4} when A_m is in square feet and S_n in litres per second. For systems containing large amounts of elastomers or plastics K_1 is more of the order of 10^{-3} to 10^{-2} . Systems constructed entirely of metal and glass may be heated to temperatures as high as 500°C to accelerate the outgassing, the average rate increasing by approximately a factor of 10 for each 100°C increase in temperature. In order to reach 10^{-6} torr in 10 h in an unconditioned vacuum chamber, the net pumping speed (in litres per second) should be 10 to 100 times the exposed area (in square feet).

The time t_v (in minutes) to vent a vacuum chamber of volume V (cubic feet) from pressures less than 1 mtorr to atmospheric pressure through a standard globe valve of nominal diameter D (inches) is given approximately by $t_v = V/100D^2$. The oil in the boiler of oil diffusion pumps should be cooled to below 100°C before opening the pump to atmospheric pressure.

APPLICATIONS OF HIGH-VACUUM PUMPS

Steam ejectors, Roots-type blowers, diffusion-ejector oil vapor pumps, and rotary piston pumps are used to produce pressures in the 10^{-1} to 10^{-4} torr range for distillation of plasticizers, fat-soluble vitamins, and certain other organic chemicals; for dehydration of frozen foods, animal tissue, blood plasma, serum, and antibiotics; for refining, degassing, and casting metals in vacuum furnaces; for vacuum sintering of cermets and powder metallurgy parts; for vacuum annealing of special alloys; and for pumping down vacuum chambers to a pressure at which high-vacuum pumps can begin operation. Diffusion pumps, getter-ion pumps, and turbomolecular pumps are used to evacuate radio, television, and X-ray tubes and vacuum-insulated containers before sealing off; to produce pressures in the 10^{-3} to 10^{-6} torr range in chambers for coating injection-molded plastic parts, rolls of plastic sheet, glass plates, and other substrates with aluminum and other metal films; for coating optical parts with antireflection films or layers producing dichroic mirrors and interference filters; for depositing metal and semiconductor films to produce microelectronic circuits; to maintain pressures of 10^{-4} to 10^{-7} torr in mass spectrometers, electron microscopes, synchrotron-clotrons, betatrons, and other devices for accelerating or separating charged particles.

14.5 FANS

by Robert Jorgensen

REFERENCES: Jorgensen, "Fan Engineering," 8th ed., Buffalo Forge Company. "Laboratory Methods of Testing Fans for Rating," ANSI/ASHRAE Standard 51-85 and ANSI/AMCA Standard 210-85. "Fans and Systems," AMCA Publication 201. "Test Code for Sound Rating," AMCA Standard 300-86. "Standards Handbook," AMCA Publication 99-83. "ASME Performance Test Codes, Code on Fans," ANSI/ASME PTC 11-1984.

Symbols

- A = area, ft² (m²)
- b = barometric pressure, in Hg (Pa)
- D = diameter, ft (m)
- D_s = specific diameter, ft (m)
- H = fan power input, hp (W)
- H_0 = fan power output, hp (W)
- h = head, ft (m)
- K_p = compressibility factor, dimensionless
- L_p = sound pressure level, dB
- L_w = sound power level, dB
- L_{ws} = specific sound power level, dB
- log = logarithm to base 10
- M = molecular weight, dimensionless
- N = speed of rotation, r/min
- N_s = specific speed, r/min
- n = number or polytropic exponent
- p_s = fan static pressure, in wg (Pa)
- p_t = fan total pressure, in wg (Pa)
- p_v = fan velocity pressure, in wg (Pa)
- p_{sx} = static pressure at plane x , in wg (Pa)
- P_{tx} = total pressure at plane x , in wg (Pa)
- P_{vx} = velocity pressure at plane x , in wg (Pa)
- \dot{Q} = fan capacity, ft³/min (m³/s)
- \dot{Q}_x = volumetric flow rate at plane x , ft³/min (m³/s)
- T = absolute temperature, °R (K)
- W = power, W
- x = factor used to determine K_p , dimensionless
- z = factor used to determine K_p , dimensionless
- γ = isentropic exponent, dimensionless
- η_s = fan static efficiency, per unit
- η_t = fan total efficiency, per unit
- ρ = fan air density, lbm/ft³ (kg/m³)
- ρ_x = air density at plane x , lbm/ft³ (kg/m³)

Subscripts

- 1 = fan inlet plane
- 2 = fan outlet plane
- a = absolute
- b = basic known conditions
- c = calculated condition
- r = reading
- x = plane 1, 2, 3, or other

FAN TYPES AND NOMENCLATURE

Any device which produces a current of air may be called a fan. This discussion will be limited to fans which have a rotating impeller to produce the flow and a stationary casing to guide the flow into and out of the impeller (see Fig. 14.5.1). The form of the casing or the impeller may vary widely.

Fan Classification

One of the characteristics by which fans are classified is the nature of the flow through the blade passages of the impeller. Axial flow, radial flow, mixed flow, and cross flow are all possible in fan impellers. Certain fan names result from these classifications. Other fan names derive from other characteristics.

Propeller fans, tube-axial fans, and vane-axial fans all utilize **axial-flow impellers**, but their casings differ. Propeller fans may be mounted in a ring or panel. Tube-axial fans and vane-axial fans both use tubular casings, but for vane-axial fans they are equipped with stationary guide vanes. A great deal of the energy transferred to the air in axial-flow machines is in kinetic form. Some of this kinetic energy can be transformed into pressure energy by straightening the swirl, e.g., with vanes, or by reducing the exit velocity, e.g., with a diffuser. Propeller fans effect very little transformation and hence have very low pressure-producing capability. Vane-axial fans can be equipped for maximum transformation as well as high transfer of energy and hence have high pressure-producing potential depending on tip speed and blade angles. High hub ratios promote high energy transfer.

Centrifugal fans and tubular centrifugal fans both utilize **radial-flow impellers**. Centrifugal fans usually employ a volute or scroll-type casing, the flow entering the casing axially and leaving tangentially. Tubular centrifugals use tubular casings so that the flow both entering and leaving the casing is axial. A considerable portion of the energy transferred to the air in a radial-flow machine is due to centrifugal action; hence the name centrifugal fan. Since centrifugal action varies with blade depth, the pressure-producing capability of radial-flow fans will vary with this factor as well as tip speed and blade angles.

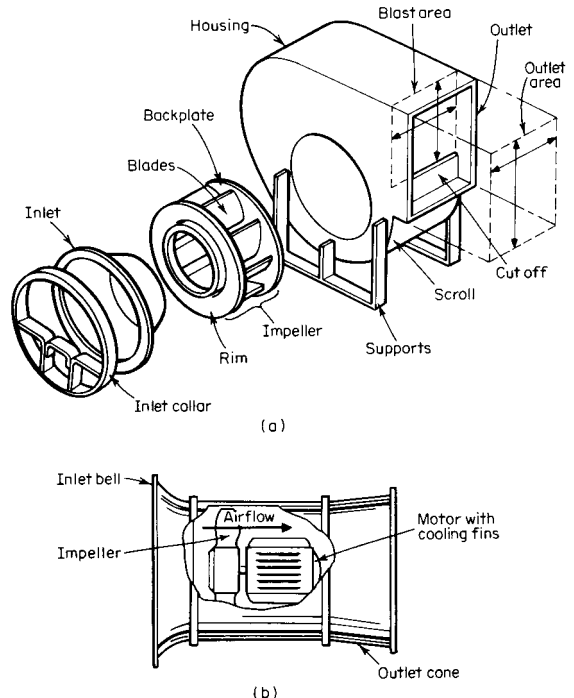


Fig. 14.5.1 Elements of fans and preferred terminology. (a) Centrifugal fans; (b) axial fans. (Based on ACMA Publ. 201.)

Mixed-flow impellers can be used in either axial or scroll-type casings. They are characterized as mixed flow because both axial and radial flow take place in the blading. Mixed-flow impellers used in axial-flow casings have a hub similar to a pure axial-flow impeller, but the inlet portion of the blading extends down over the face of the hub, thereby giving some radial guidance. Mixed-flow impellers used in scroll-cased fans have blades which give most of the axial guidance in the inlet portion and most of the radial guidance in the discharge portion.

In a **cross-flow impeller**, the air passes through the blading twice, entering more or less tangentially through the tip, passing across the impeller and out the other side. The casings are designed to provide this transverse flow. Cross-flow fans are also known as tangential fans or transverse-flow fans. Pressure-producing potential is low and depends on the formation of a vortex as the air leaves the impeller.

Fan Details

Propeller fans and other **axial fans** may use blades shaped to airfoil sections or blades of uniform thickness. Blading may be fixed, adjustable at standstill, or variable in operation. Propeller fans have very small hubs. Hub-to-tip diameter ratios ranging from 0.4 to 0.7 are common in vane-axial fans. The larger the hub, the more important it is to have an inner cylinder approximately the hub size located downstream of the impeller. The guide vanes of a vane-axial fan are located in the annular space between the tubular casing and the inner cylinder. Diffusers are generally used between the fan and the discharge ductwork.

Centrifugal fans use various types of blading. Forward-curved blades are shallow and curved so that both the tip and the heel point in the direction of rotation. Radial and radial-tip blades both are radial at the tip, but the latter are curved at the heel to point in the direction of rotation. Backward-curved and backward-inclined blades point in the direction opposite rotation at the tip and in the direction of rotation at the heel. All the above blades are of uniform thickness and are designed for radial flow. Airfoil blades have backward-curved chord lines so that the leading edge of the airfoil is at the heel pointing forward and the trailing edge at the tip pointing backward with respect to rotation. Impellers for all blade shapes are usually shrouded and may have single or double inlets. Blade widths are related to the inlet-to-tip-diameter ratio. Tip angles may vary widely, but heel angles should be set to minimize entrance losses. Scroll casings may be fitted with a streamlined inlet bell, an inlet cone, or simply a collar.

Tubular centrifugals may be designed for backward-curve, air-foil or mixed-flow impellers. An inlet bell and discharge guide vanes are required for good performance.

Cross-flow fans utilize impellers with blading similar to that of a forward-curved centrifugal, but the end shrouds have no inlet holes. Blade-length-to-tip-diameter ratios are limited only by structural considerations.

Power roof ventilators may use either axial- or radial-flow impellers. The casings will include either a propeller fan mounting ring or a tubular casing to guide the flow for an axial-flow impeller. If a radial-flow impeller is used, an inlet bell is required, but the scroll case may be replaced by the ventilator hood.

All fan types may have direct or indirect drive arrangements. Various standards including arrangement numbers have been adopted by AMCA. Arr. 1 signifies an overhung impeller on a fan shaft with two bearings on a base; Arr. 3 signifies an impeller on a fan shaft between bearings; Arr. 4 is for overhung impeller on a motor shaft; Arr. 7 is Arr. 3 with a motor base; Arr. 8 is Arr. 1 with an extended base for motor; Arr. 9 is Arr. 1 with motor mounted on side of unit; Arr. 10 is Arr. 1 with motor mounted inside of base. Rotation is specified as cw or ccw when viewed from the drive side. All fans can be equipped with variable-speed drives, variable inlet vanes, or dampers, but controllable-pitch axials do not need speed or vane control.

FAN PERFORMANCE AND TESTING

The conventional terms used to describe fan performance in the United States are defined blow.

Fan Air Density Air density is the mass per unit volume of the air. The density of a perfect gas is a function of its molecular weight, temperature, and pressure as indicated by

$$\rho_x = \frac{M}{386.7} \frac{529.7}{T_x} \frac{b_x}{29.92}$$

where the subscript *x* indicates the plane of the measurements. For dry air this reduces to

$$\rho_x = 1.325b_x/T_x$$

This expression will usually be accurate enough even when moist air is involved. Fan air density is the density of the air corresponding to the total pressure and total temperature at the fan inlet

$$\rho = \rho_1$$

Fan Capacity Volume flow rate is usually determined from pressure measurements, e.g., a velocity-pressure traverse taken with a Pitot static tube or a pressure drop across a flowmeter. The average velocity pressure for a Pitot traverse is

$$p_{vx} = (\sum \sqrt{p_{vx}}/n)^2$$

where subscript *x* indicates the plane of the measurements, subscript *r* indicates a reading at one station, *n* is the number of stations, and \sum is the summation sign. The corresponding capacity is:

$$\dot{Q}_x = 1.097A_x \sqrt{p_{vx}/\rho_x}$$

For a flow meter

$$\dot{Q}_x = 1.097CA_x Y \sqrt{\Delta p/\rho_x}/F$$

where *C* = coefficient of discharge of meter, *Y* = expansion factor for gas, Δp = measured pressure drop, and *F* = velocity-of-approach factor for the meter installation. Fan capacity is the volumetric flow rate at fan air density

$$\dot{Q} = \dot{Q}_x \rho_x / \rho$$

Fan Total Pressure Fan total pressure is the difference between the total pressure at the fan outlet and the total pressure at the fan inlet

$$p_{t2} = p_{t2} - p_{t1}$$

When the fan draws directly from the atmosphere.

$$p_{t1} = 0$$

When the fan discharges directly to the atmosphere,

$$p_{t2} = p_{v2}$$

If either side of the fan is connected to ductwork, etc., and the measuring plane is remote, the measured values should be corrected for the approximate pressure drop

$$p_{t2} = p_{tx} + \Delta p_{2-x} \quad p_{t1} = p_{tx} + \Delta p_{x-1}$$

Fan Velocity Pressure Fan velocity pressure is the pressure corresponding to the average velocity at the fan outlet

$$p_v = \left(\frac{\dot{Q}_2/A_2}{1.097} \right)^2 \rho_2$$

Fan Static Pressure Fan static pressure is the difference between the fan total pressure and the fan velocity pressure. Therefore, fan static pressure is the difference between the static pressure at the fan outlet and the total pressure at the fan inlet

$$p_s = p_t - p_v = p_{s2} - p_{t1}$$

Fan Speed Fan speed is the rotative speed of the impeller.

Compressibility Factor The compressibility factor is the ratio of the fan total pressure *p'*_t that would be developed with an incompressible fluid to the fan total pressure *p*_t that is developed with a compressible fluid, all other conditions being equal:

$$K_p = \frac{p'_t}{p_t} = \frac{[n/(n-1)] [(p_{t2a}/p_{t1a})^{(n-1)/n} - 1]}{(p_{t2a}/p_{t1a}) - 1}$$

Compressibility factor can be determined from test measurements using

$$\begin{aligned}x &= p_t/p_{1a} \\z &= [(\gamma - 1)/\gamma] 6356H/\dot{Q}p_{1a} \\K_p &= [z \log(1 + x)]/[x \log(1 + z)]\end{aligned}$$

Fan Power Output Fan power output is the product of fan capacity and fan total pressure and compressibility factor

$$H_o = \dot{Q}p_t K_p / 6,356$$

Fan Power Input Fan power input is the power required to drive the fan and any elements in the drive train which are considered a part of the fan. Power input can be calculated from appropriate measurements for a dynamometer, torque meter, or calibrated motor.

Fan Total Efficiency Fan total efficiency is the ratio of the fan power output to the fan power input

$$\eta_t = \dot{Q}p_t K_p / 6,356H$$

Fan Static Efficiency Fan static efficiency is the fan total efficiency multiplied by the ratio of fan static pressure to fan total pressure

$$\eta_s = \eta_t p_s / p_t$$

Fan Sound Power Level Fan sound power level is 10 times the logarithm (base 10) of the ratio of the actual sound power in watts to 10^{-12} watts,

$$L_w = 10 \log(W/10^{-12})$$

Fan sound power level can be calculated from sound pressure-level measurements in a known acoustical environment. The standard laboratory method of testing is to use a calibrated sound source to calibrate a semireverberant room. In-duct test methods are being developed. The total sound power level of a fan is usually assumed to be 3 dB higher than either the inlet or outlet component. The casing component varies with construction but will usually range from 15 to 30 dB less than the total.

Head The difference between head and pressure is important in fan engineering. Both are measures of the energy in the air. Head is energy per unit weight and can be expressed in ft·lb/lb, which is often abbreviated to ft (of fluid flowing). Pressure is energy per unit volume and can be expressed in ft·lb/ft³, which simplifies to lb/ft² or force per unit area. The use of the inch water gage (in wg) is a convenience in fan engineering reflecting the usual methods of measurement. It sounds like a head measurement but is actually a pressure measurement corresponding to 5.192 lb/ft², the pressure exerted by a column of water 1 inch high. Pressures can be converted into heads and vice versa:

$$p = ph/5.192 \quad h = 5.192p/p$$

For instance, for air at 0.075 lbm/ft³ density, 1 in wg corresponds to 69.4 ft of air. That is, a column of air at that density would exert a pressure of 5.192 lb/ft². Lighter air would exert less pressure for a given head. For a given pressure, the head would be higher with lighter air. Although it is not widely used in the United States, head is commonly used in a number of European countries.

Fans, like other turbomachines, can be considered constant-head-constant-capacity (volumetric) machines. This means that a fan will develop the same head at a given capacity regardless of the fluid handled, all other conditions being equal. Of course, this also means that a fan will develop a pressure proportional to the density at a given capacity, all other conditions being equal.

All the preceding equations are based on the U.S. Customary Units listed under Symbols. If SI units are to be used, certain numerical coefficients will have to be modified. For instance, substitute $\sqrt{2}$ for 1,097 in the flow equations when using Pa for pressure, kg/m³ for density, and m³/s for capacity. Similarly, substitute 1.0 for 6,356 in the power equations when using Pa for pressure, m³/s for capacity, and W for power.

Common metric practice in fan engineering leads to other numerical coefficients. When mm wg is used for pressure, m³/s for capacity, kW for power, and kg/m³ for density, substitute 4.424 for 1,097 in the flow equations and 102.2 for 6,356 in the power equations.

The preceding discussion utilized what may be termed the **volume-flow-rate-pressure** approach to expressing fan performance. An alternative, the **mass-flow-rate-specific-energy** approach is equally valid but not generally used in the United States. Refer to ASME PTC-11 for additional details.

FAN AND SYSTEM PERFORMANCE CHARACTERISTICS

The performance characteristics of a fan are best described by a graph. The conventional method of graphing fan performance is to plot a series of curves with capacity as abscissa and all other variables as ordinates. System characteristics can be plotted in a similar manner.

System Characteristics

Most systems served by a fan have characteristics which can be described by a parabola passing through the origin; i.e., the energy required to produce flow through the system (which can be expressed as pressure or head) varies approximately as the square of the flow. In some cases, the system characteristics will not pass through the origin because the energy required to produce flow through an element of the system may be controlled at a particular value, e.g., with venturi scrubbers. In some cases, the system characteristic will not be parabolic because the flow through an element of the system is laminar rather than turbulent, e.g., in some types of filters. Whatever the case, the system designer should establish the characteristics by determining the energy requirements at various flow rates. The energy requirement (pressure drops or head losses) for each element can be determined by reference to handbook or manufacturer's literature or by test.

The true measure of the energy requirement for a system element is the total pressure drop or the total head loss. Only if the entrance velocity for the element equals the exit velocity will the change in static pressure equal the total pressure drop. There are some advantages in using static pressure change, but the system designer is usually well advised to use total pressure drops to avoid errors in fan selection. The sum of the total pressure losses for elements on the inlet side of the fan will equal $-p_{11}$. This should include energy losses at the entrance to the system but not the energy to accelerate the air to the velocity at the fan inlet, which is chargeable to the fan. The sum of the total pressure losses for elements on the discharge side of the fan will equal p_{12} . This should include the kinetic energy of the stream issuing from the system. The total system requirement will be the arithmetic sum of all the appropriate losses or the algebraic difference $p_{12} - p_{11}$, which is also p_t .

A system characteristic curve based on total pressure is plotted in Fig. 14.5.2. A characteristic based on static pressure is also shown. The latter recognizes the definition of fan static pressure so that the only difference is the velocity pressure corresponding to the fan outlet velocity. This system could operate at any capacity provided a fan delivered the exact pressure to match the energy requirements shown on the system curve for that capacity. The advantages of plotting the system characteristics on the fan graph will become evident in the following discussions.

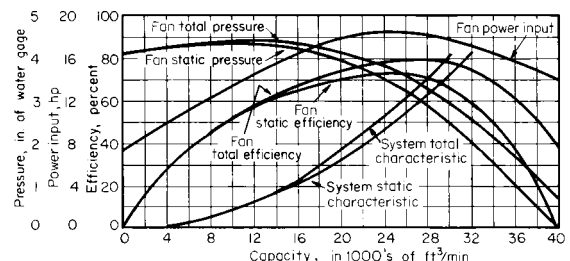


Fig. 14.5.2 Fan and system characteristics.

Fan Characteristics

The constant-speed performance characteristics of a fan are illustrated in Fig. 14.5.2. These characteristics are for a particular size and type of fan operating at a particular speed and handling air of a particular density. The fan can operate at any capacity from zero to the maximum shown, but when applied on a particular system the fan will operate only at the intersection of the system characteristics with the appropriate fan pressure characteristic. For the case illustrated, the fan will operate at $Q = 27,300 \text{ ft}^3/\text{min}$ and $p_i = 3.4 \text{ in wg}$ or $p_s = 3 \text{ in wg}$, requiring $H = 18.3 \text{ hp}$ at the speed and density for which the curves were drawn. The static efficiency at this point of operation is 73 percent, and the total efficiency is 80 percent. If the system characteristic had been lower, it would have intersected the fan characteristic at a higher capacity and the fan would have delivered more air. Contrariwise, if the system characteristic had been higher, the capacity of the fan would have been less. Capacity reduction can be accomplished, in fact, by creating additional resistance, as with an outlet damper.

Figure 14.5.3 illustrates the characteristics of a fan with damper control, variable inlet vane control, and variable speed control. These particular characteristics are for a backward-curved centrifugal fan, but the general principles apply to all fans. Outlet dampers do not affect the flow to the fan and therefore can alter fan performance only by adding resistance to the system and producing a new intersection. Point 1 is for wide-open dampers. Points 2 and 3 are for progressively closed dampers. Note that some power reduction accompanies the capacity reduction for this particular fan. Variable-inlet vanes produce inlet whirl, which reduces pressure-producing capability. Point 4 is for wide-open inlet vanes and corresponds to point 1. Points 5 and 6 are for progressively closed vanes. Note that the power reduction at reduced capacity is better for vanes than for dampers. Variable speed is the most efficient means of capacity control. Point 7 is for full speed and points 8 and 9 for progressively reduced speed. Note the improved power savings over other methods. Variable speed also has advantages in terms of lower noise and reduced erosion potential but is generally at a disadvantage regarding first cost.

Figure 14.5.4 illustrates the characteristics of a fan with variable pitch control. These characteristics are for an axial-flow fan, but comparisons to the centrifugal-fan ratings in Fig. 14.5.3 can be made. Point 10 corresponds to points 1, 4, or 7. Reduced ratings are shown at points 11 and 12 obtained by reducing the pitch. Power savings can be almost as good as with speed control. Notice that in all methods of capacity control,

operation is at the intersection of a system characteristic with a fan characteristic. With variable vanes, variable speed, and variable pitch, the fan characteristic is modified. With damper control, the effect could be considered a change in fan characteristics, but it generally makes more sense to consider it a change in system characteristics.

Fan-System Matching

It has already been observed in the discussions of both system and fan characteristics that the point of operation for **one fan** on a particular system will be at the intersection of their characteristics. Stated another way, the energy required by the system must be provided by the fan exactly. If the fan delivers too much or too little energy, the capacity will be more or less than desired. The effects of utilizing dampers, variable speed, variable-inlet vanes, and variable pitch were illustrated, but in all cases the fan was preselected. The more general case is the one involving the selection or design of a fan to do a particular job. The design of a fan from an aerodynamicist's point of view is beyond the scope of this discussion. Fortunately, most fan problems can be solved by selecting a fan from the many standard lines available commercially. Again, the crux of the matter is to match fan capability with system requirements.

Most fan manufacturers have several standard lines of fans. Each line may consist of various sizes all resembling each other. If the blade angles and proportions are the same, the line is said to be **homologous**. There is only one fan size in a line of homologous fans that will operate at the maximum efficiency point on a given system. If the fan is too big, operation will be at some point to the left of peak efficiency on a standard characteristic plot. If the fan is too small, operation will be to the right of peak. In either case it will be off peak. A slightly undersized fan is often preferred for reasons of cost and stability.

Selecting and rating a fan from a catalog is a matter of fan-system matching. When it is recognized that many of the catalog sizes may be able to provide the capacity and pressure, selection becomes a matter of trying various sizes and comparing speed, power, cost, etc. The choice is a matter of evaluation.

There are various reasons for using **more than one fan** on a system. Supply and exhaust fans are used in ventilation to avoid excessive pressure build-up in the space being served. Forced and induced-draft fans are used to maintain a specified draft over the fire. Two fans may fit the available space better than one larger fan. Capacity control by various fan combinations may be more economical than other control methods.

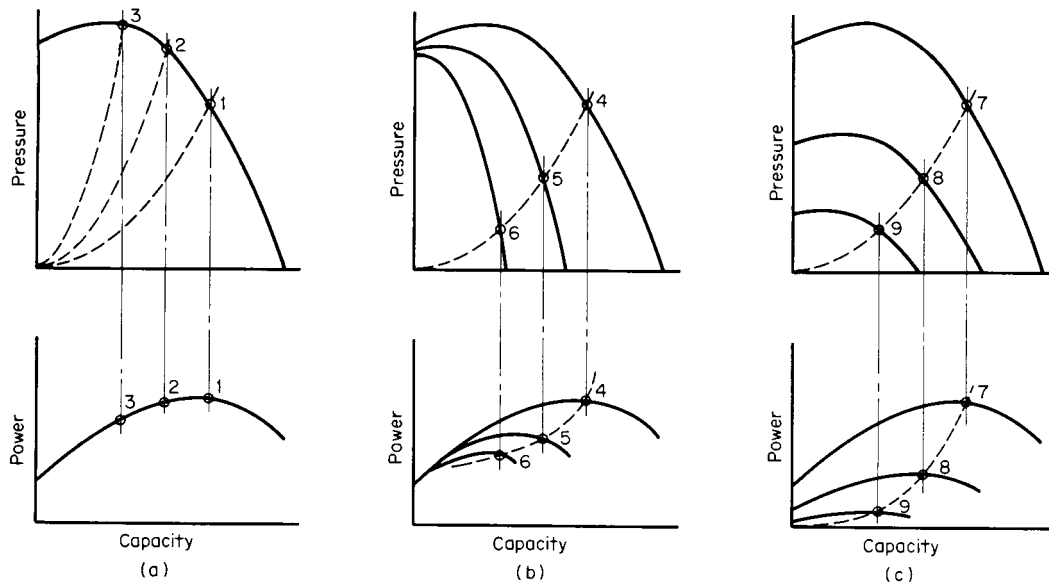


Fig. 14.5.3 Fan characteristics with (a) damper control, (b) variable-inlet vane control, (c) variable speed control.

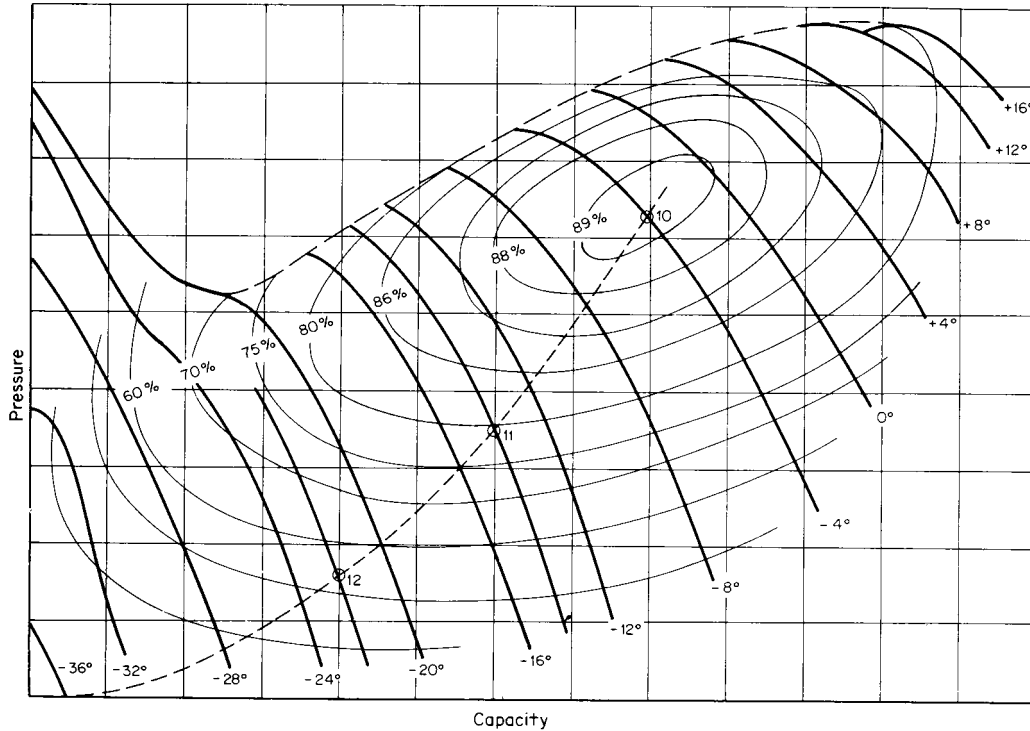


Fig. 14.5.4 Characteristics of an axial-flow fan with variable-pitch control.

Multistage arrangements may be necessary when pressure requirements exceed the capabilities of a single-stage fan. Standby fans are frequently required to ensure continuous operation.

When two fans are used, they may be located quite remote from each other or they may be close enough to share shaft and bearings or even casings. Double-width, double-inlet fans are essentially two fans in parallel in a common housing. Multistage blowers are, in effect, two or more fans in series in the same casing. Fans may also be in series but at opposite ends of the system. Parallel-arrangement fans may have almost any amount of their operating resistance in common. At one extreme, the fans may have common inlet and discharge plenums. At the other extreme, the fans may both have considerable individual ductwork of equal or unequal resistance.

Fans in **series** must all handle the same amount of gas by weight measurements, assuming no losses or gains between stages. The combined total pressure will be the sum of the individual fans, total pressures. The velocity pressure of the combination can be defined as the pressure corresponding to the velocity through the outlet of the last stage. The static pressure for the combination is the difference between its total and velocity pressures and is therefore not equal to the sum of the individual fan static pressures. The volumetric capacities will differ whenever the inlet densities vary from stage to stage. Compression in one stage will reduce the volume entering the next if there is no re-expansion between the two. As with any fan, the pressure capabilities are also influenced by density.

The combined total pressure-capacity characteristic for two fans in series can be drawn by using the volumetric capacities of the first stage for the abscissa and the sum of the appropriate total pressures for the ordinate. Because of compressibility, the volumetric capacities of the second stage will not equal the volumetric capacities of the first stage. The individual total pressures must be chosen accordingly before they are combined. If the gas can be considered incompressible, the pressures for the two stages may be read at the same capacity. In the area near free delivery, it may be necessary to estimate the negative pressure

characteristics of one of the fans in order to combine values at the appropriate capacity.

Fans in **parallel** must all develop sufficient pressure to overcome the losses in any individual ductwork, etc., as well as the losses in the common portions of the system. When such fans have no individual ductwork but discharge into a common plenum, their individual velocity pressures are lost and the fans should be selected to produce the same fan static pressures. If fan velocity pressures are equal, the fan total pressures will be equal in such cases. When the fans do have individual ducts but they are of equal resistance and joined together at equal velocities, the fans should be selected for the same fan total pressures. If fan velocity pressures are equal, fan static pressures will be equal in this case. If the two streams join together at unequal velocities, there will be a transfer of momentum from the higher-velocity stream to the lower-velocity stream. The fans serving the lower-velocity branch can be selected for a correspondingly lower total pressure. The other fan must be selected for a correspondingly higher total pressure than if velocities were equal.

The combined pressure-capacity curves for two fans in parallel can be plotted by using the appropriate pressures for ordinates and the sum of the corresponding capacities for abscissa. Such curves are meaningful only when a combined-system curve can be drawn. In the area near shutoff, it may be necessary to estimate the negative capacity characteristics of one of the fans in order to combine values at the appropriate pressure.

Figure 14.5.5 illustrates the combined characteristics of two fans with slightly different individual characteristics (*A-A* and *B-B*). The combined characteristics are shown for the two fans in series (*C-C*) and in parallel (*D-D*). Only total-pressure curves are shown. This is always correct for series arrangements but may introduce slight errors for parallel arrangements. An incompressible gas has been assumed. The questionable areas near shutoff or free delivery have been omitted. Two different system characteristics (*E-E* and *F-F*) have been drawn on the chart. With the two fans in series, operation will be at point *EC* or *FC* if

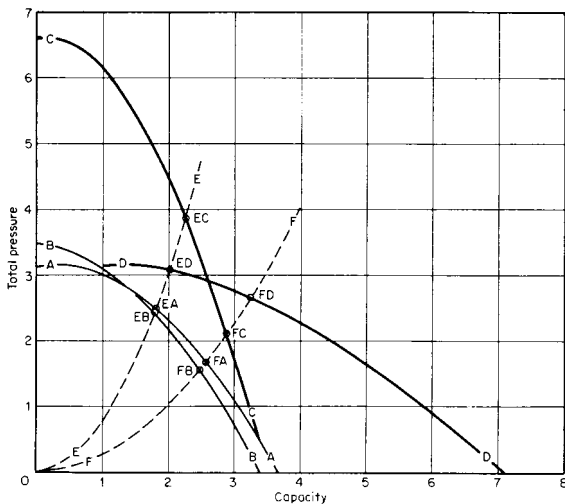


Fig. 14.5.5 Combined characteristics of fans in series and parallel.

the fan is on system *E* or *F*, respectively. Parallel arrangement will lead to operation at *ED* or *FD*. Single-fan operation would be at the point indicated by the intersection of the appropriate fan and system curves provided the effect of an inoperative second fan is negligible. Some sort of bypass is required around an inoperative fan in series whereas an inoperative fan in parallel need only be dampered shut. Parallel operation yields a higher capacity than series operation on system *F*, but the reverse is true on system *E*. For the type of fan and system characteristics drawn there is only one possible point of operation for any arrangement.

Fan Laws

The fan laws are based on the experimentally demonstrable fact that any two members of a homologous series of fans have performance curves which are homologous. At the same point of rating, i.e., at similarly situated points of operation on their characteristic curves, efficiencies are equal and other variables are interrelated according to the fan laws. If size and speed are considered independent variables and if compressibility effects are ignored, the fan laws can be written as follows:

$$\begin{aligned} \eta_{tc} &= \eta_{tb} \\ \dot{Q}_c &= \dot{Q}_b(D_c/D_b)^3(N_c/N_b) \\ p_{tc} &= p_{tb}(D_c/D_b)^2(N_c/N_b)^2(\rho_c/\rho_b) \\ H_c &= H_b(D_c/D_b)^5(N_c/N_b)^3(\rho_c/\rho_b) \\ L_{wc} &= L_{wb} + 70 \log(D_c/D_b) + 50 \log(N_c/N_b) \\ &\quad + 20 \log(\rho_c/\rho_b) \end{aligned}$$

The above laws are useful, but they are dangerous if misapplied. The calculated fan must have the same point of rating as the known fan. When in doubt, it is best to reselect the fan rather than attempt to use the fan laws.

The fan designer utilizes the fan laws in various ways. Some of the more useful relationships in addition to those above derive from considering fan capacity and fan total pressure as the independent variables. This leads to specific diameter, specific speed, and specific sound power level:

$$\begin{aligned} D_s &= D(p_t/\rho)^{1/4}\dot{Q}^{1/2} \\ N_s &= N\dot{Q}^{1/2}/(p_t/\rho)^{3/4} \\ L_{ws} &= L_w - 10 \log \dot{Q} - 20 \log p_t \end{aligned}$$

D_s , N_s , and L_{ws} are the diameter, speed, and sound power level of a homologous fan which will deliver 1 ft³/min at 1 in wg at the same point of rating as \dot{Q} and p_t for D and N . D_s and N_s can be used to advantage in fan selection. They can also be used by a designer to determine how

well a line will fit in with other lines. This is illustrated in Fig. 14.5.6. Each segment represents a particular fan line. Note the trends for each kind of fan. Incidentally, some fan engineers utilize different formulas for specific diameter and specific speed

$$D_{se} = Dp_{te}^{1/4}/\dot{Q}^{1/2} \quad N_{se} = N\dot{Q}^{1/2}/p_{te}^{3/4}$$

where p_{te} is the equivalent total pressure based on standard air. This makes $D_{se} = D_s \times 1.911$ and $N_{se} = N_s \times 6.978$.

Specific sound power level is useful in predicting noise levels as well as in comparing fan designs. There appears to be a lower limit of L_{ws} in the vicinity of forty-five dB for the more efficient types ranging to 70 dB or more for cruder designs. Actual sound power levels can be figured from

$$L_w = L_{ws} + 10 \log \dot{Q} + 20 \log p_t$$

Another useful parameter which derives from the fan laws is orifice ratio

$$R_o = \dot{Q}/D^2(p_t/\rho)^{1/2}$$

This ratio can be plotted on a characteristic curve for a known fan. If the ratio is determined for a calculated homologous fan, the point of rating can be established by inspection. Other ratios can be used in the same manner including p_v/p_t , p_t/\dot{Q}^2 , D_s , and N_s .

Stability Considerations

The flow through a system and its fan will normally be steady. If the fluctuations occasioned by a temporary disturbance are quickly damped out, the fan system may be described as having a **stable** operating characteristic. If the unsteady flow continues after the disturbance is removed, the operating characteristic is **unstable**.

To ensure stable operation the slopes of the pressure-capacity curves for the fan and system should be of opposite sign. Almost all systems have a positive slope; i.e., the pressure requirement or resistance increases with capacity. Therefore, for stable operation the fan curve should have a negative slope. Such is the case at or above the design capacity.

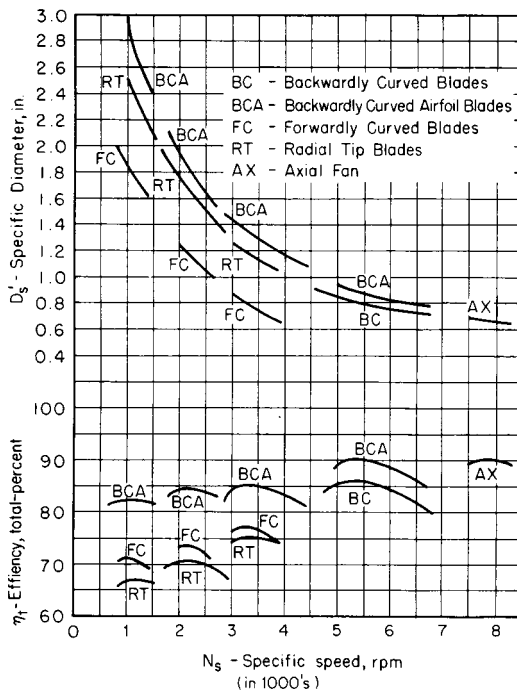


Fig. 14.5.6 Specific diameter and efficiency versus specific speed for single-inlet fan types.

When the slopes of the fan and system characteristics are of opposite sign, any system disturbance tending to produce a temporary decrease in flow is nullified by the increase in fan pressure. When the slopes are of the same sign, any tendency to decrease flow is strengthened by the resulting decrease in fan pressure. When fan and system curves coincide over a range of capacities, the operating characteristics are extremely unstable. Even if the curves exactly coincide at only one point, the flow may vary over a considerable range.

There may or may not be any obvious indication of unstable operation. The pressure and power fluctuations that accompany unsteady flow may be so small and rapid that they cannot be detected by any but the most sensitive instruments. Less rapid fluctuations may be detected on the ordinary instruments used in fan testing. The changes in noise which occur with each change in flow rate are easily detected by ear as individual beats if the beat frequency is below about 10 Hz. In any event, the overall noise level will be higher with unsteady flow than with steady flow.

The conditions which accompany unsteady flow are variously described as pulsations, hunting, surging, or pumping. Since these conditions occur only when the operating point is to the left of maximum pressure on the fan curve, this peak is frequently referred to as the **surge point** or **pumping limit**.

Pulsation can be prevented by rating the fan to the right of the surge point. Fans are usually selected on this basis, but it is sometimes necessary to control the volume delivered to the value below that at the surge point. This may lead to pulsation, particularly if the fan pressure exceeds 10 in wg.

If the required capacity is less than that at the pumping limit, pulsation can be prevented in various ways, all of which in effect provide a negatively sloping fan curve at the actual operating point. To accomplish this effect the required pressure must be less than the fan capabilities at the required capacity. One method is to bleed sufficient air for actual operation to be beyond the pumping limit. Other possible methods are the use of pitch, speed, or vane control for volume reduction. In any of these cases, the point of operation on the new fan curve must be to the right of the new surge point. Although in the section on Capacity Control dampers were considered a part of the system, they may also be considered a part of the fan if located in the right position. Accordingly, pulsations may be eliminated in a supply system if the damper is on the inlet of the blower. Similarly, dampening at the outlet of the exhauster may control pulsation in exhaust systems.

Another condition frequently referred to as instability is associated with flow separation in the blade passages of an impeller and is evidenced by slight discontinuities in the performance curve. There may be a small range of capacities at which two distinctly different pressures may be developed depending on which of the two flow patterns exists. Such a condition usually occurs at capacities just to the left of peak efficiency. Still another condition of unsteady flow may develop at extremely low capacities. This is known as **blowback** or **puffing** because air puffs in and out of a portion of the inlet. Operation in the blowback range should be avoided, particularly with high-energy fans.

A different type of unsteady flow may occur when two or more fans are used in parallel. If the individual fan characteristics exhibit a dip in pressure between shutoff and design, the combined characteristic will contain points where the point of operation of the individual fans may be widely separated even for identical fans. If the system characteristic intersects the combined-fan characteristic at such a point, the individual fans may suddenly exchange loads. That is, the fan operating at high capacity may become the one operating at low capacity and vice versa. This can produce undesirable shocks on motors and ducts. Careful matching of fan to system is required for either forward-curved centrifugals or most axials for this reason.

Fan Applications

The selection of a particular size and type of fan for a particular application involves considerations of aerodynamic, economic, and functional suitability. Many of the factors involved in aerodynamic suitability have been discussed above. Determination of economic suitability requires

an evaluation of first cost and operating costs. The functional suitability of various types of fans with respect to certain applications is discussed below.

Heating, ventilating, and air-conditioning systems may require supply and exhaust or return air fans. Historically, high-efficiency centrifugal fans, using either backward-curved or airfoil blades have been used for supply on duct systems. In low-pressure applications these types can be used without sound treatment. In high-pressure applications, sound treatment is almost always required. Axials have long been used for shipboard ventilation because they generally can be made smaller than centrifugals. Both adjustable axials and tubular centrifugals have proved popular on duct systems for exhaust service in building ventilation. Centrifugals with variable inlet vanes and axials with pitch control are being used for supply in variable-air-volume systems. Propeller fans and power roof ventilators are used for either supply or exhaust systems when there is little or no ductwork. Heating, ventilating, and air-conditioning applications are considered clean-air service. Various classes of construction are available in standard lines for different pressure ranges. Both direct and indirect drive are used, the latter being most common.

Industrial exhaust systems generally require fans that are less susceptible to the unbalance that may result from dirty-gas applications than the clean-air fans used for heating, ventilating, and air conditioning. Simple, rugged, industrial exhausters are favored for applications up to 200 hp. They have a few radial blades and relatively low efficiency. Most are V-belt-driven. Extra-heavy construction may be required where significant material passes through the fan.

Process air requirements can be met with either centrifugal or axial fans; the latter may be used in single or double stage. The higher pressure ratings are usually provided by a single-stage centrifugal fan with radial blades, known as a pressure blower. These units are generally direct connected to the driver. They not only compete with centrifugal compressors but resemble them.

Large industrial process and pollution-control systems involving more than about 200 hp are generally satisfied with a somewhat more sophisticated fan than an industrial exhauster or pressure blower. Centrifugal fans with radial-tip blades are frequently used on the more severe service. For the less severe requirements, backward-curved or airfoil blades may be used. Rugged fixed-pitch axials have also been used. Industrial fans are usually equipped with inlet boxes and independently mounted bearings and are usually direct-driven. Journal bearings are usually preferred. Inlet-box damper control can be used to approximate the power saving available from variable-inlet vane control. Variable-speed hydraulic couplings may be economically justified in some cases. Special methods or special construction may be required to provide protection against corrosion or erosion. These fans tend to take on the name of the application such as sintering fan, scrubber exhaust fan, etc.

Mechanical draft systems may utilize any of the fan types described in connection with the above applications. Ventilating fans, industrial exhausters, and pressure blowers have been used for forced draft, induced draft, and primary air service on small steam-generating units. The large generating units are generally equipped with the most efficient fans available consistent with the erosion-corrosion potential of the gas being handled.

Both centrifugals and axials are used for forced and induced draft. Axials have predominated in Europe, and there is a growing trend throughout the rest of the world toward axials. Centrifugals have been used almost exclusively in the United States until very recently.

Forced-draft centrifugals invariably have airfoil blade impellers. Induced-draft centrifugals may have airfoil-blade impellers, but for scrubber exhaust, radial-tip blades are more common. This is due, in part, to the high pressures required for scrubber operation and in part to the erosion-corrosion potential downstream of a scrubber. Gas recirculating fans are usually of the radial-tip design. Forced-draft control may be by variable speed but is more likely to be variable vanes on centrifugal fans. Variable inlet vanes or inlet-box dampers may be used to control induced-draft fans. All large centrifugals are direct-connected and have independent pedestal-mounted bearings. Journal bearings are almost always used.

Forced-draft axials are likely to be of the variable-pitch full-airfoil-section design. Hydraulic systems are almost always used for pitch control, but pneumatic and mechanical systems have been tried. Bearings are usually of the antifriction type. Fixed-pitch axials are usually used for induced-draft duty. Control is by variable-inlet vanes. Either journal or antifriction bearings may be used.

Other Systems

Fans are incorporated in many different kinds of machines. Electronic equipment may require cooling fans to prevent hot spots. Driers use fans to circulate air to carry heat to, and moisture away from, the product. Air-support structures require fans to inflate them and maintain the

supporting pressure. Ground-effect machines use fans to provide the lift pressure. Air conditioners and other heat exchangers incorporate fans. Aerodynamic, economic, and functional considerations will dictate the type and size of fan to be used.

Tunnel ventilation can be achieved by using either axial or centrifugal fans. Transverse ventilation utilizes supply and exhaust fans connected to the tunnel by ductwork. These fans are rated in the manner described above. Another method utilizes specially tested and rated axial fans called **jet fans**. The fans, which are placed in the tunnel, increase the momentum of some of the air flowing with the traffic. This air induces additional flow.