SECTION 3.8

VANE, GEAR, AND LOBE PUMPS

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DEFINITIONS AND NOMENCLATURE

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

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

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

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

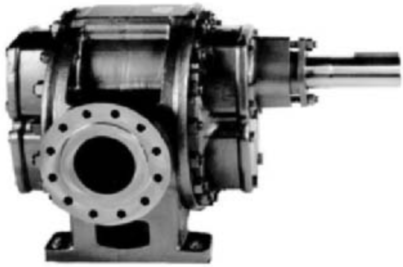

FIGURE 1 A typical rotary gear pump

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

With rotary pumps (see Figure 1), a driver turns one shaft and rotor assembly, which in turn physically meshes with another to form the cavities that move the fluid. This is known as an *untimed arrangement*. For some applications, however, there would be problems with the gears, lobes, or screws meshing this way. For instance, stainless steel gears will gall and seize if rubbed against each other. High wear rates will also occur if any dirt is trapped between the meshing lobes of a lobe pump, regardless of their material, or if a pump with meshing gears is run dry.

To circumvent this, the timed pump was developed. It uses timing gears physically located outside the pumping chamber to transmit torque between the pump shafts and synchronize the pumping elements relative to each other. By preventing them from contacting each other, they eliminate many of the problems of dirty fluids, material compatibility, and dry running. Most lobe pumps are built this way, and gear pumps can be timed or untimed as well.

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

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

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

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

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

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

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

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

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

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

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

The materials must also have compatible coefficients of thermal expansion. With the potential for deflection of the rotating parts always present, the materials selected must also have good bearing characteristics to resist galling up to the point of a compressive yield of the mating materials. This is especially important when pumping low-lubricity fluids.

Furthermore, materials used for corrosion resistance in non-contacting surfaces of centrifugal pumps may be unusable in rotary pumps where the continuous sliding contact between parts can wear away their passivating or protective layers. In general, rotary material restrictions become more severe when handling low-viscosity fluids at higher pressures and/or low lubricity fluids with abrasives. In addition, even where there is no load-bearing contact between the rotating and stationary parts under normal conditions, the high transient forces generated at startup, shutdown, or any other unusual operating conditions (such as cavitation) must be considered when selecting pump materials.

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

Vane Pumps Two basic types of vane pumps exist. The most common is the rigid sliding metal vane type, and the other is the flexible or elastomeric vane used for dirty or chemically aggressive fluids. Both are based around external sliding vanes rotating about a non-concentric cam.

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

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

When the cam surface is external to the radial surface of the rotor and the vane, or the vanes are mounted in the body or stator, the pump is called an *external vane pump* and is illustrated in Figure 2. The OTI, CTIO, and OTO volumes are defined the same as for internal vane pumps when multiple external vanes are used. In this case, the rotor surface, the body surface, and the fluid seal points between them define the CTIO volume.

In addition to rigid, sliding metal vane pumps, flexible or elastomeric vane type pumps also exist. This kind of pump, illustrated in Figure 3, has a pumping action similar to that of an internal vane pump with the OTI, CTIO, and OTO volumes defined by the rotor surfaces, the body surfaces, the fluid seal contacts between the rotor flexible vanes, and the body surfaces.

The flexible liner pump in Figure 4 is similar in pumping action to the external vane pump, and all three chamber volumes of it are defined by the inner surface of the body, the outer surface of the liner, and the liquid seal contact between the liner and body bore. Most flexible liner pumps, unlike other rotary pumps, have at least one position of the rotor in which no fluid seal exists between the OTI and OTO volumes. The pump depends only on fluid velocity and inertia to limit backflow during this phase of rotation.

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

Gear Pumps Evidenced by drawings dating back to the 16th century, the gear pump is one of the oldest pumps of any type. It is also the most common of all rotary pumps due to the wide variety of applications it can be used in.

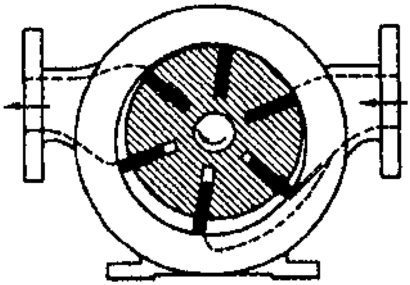

FIGURE 2 A typical external vane pump

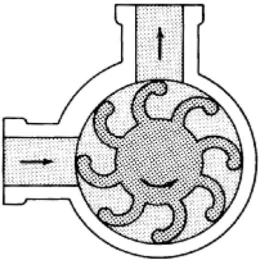

FIGURE 3 Flexible vane pump

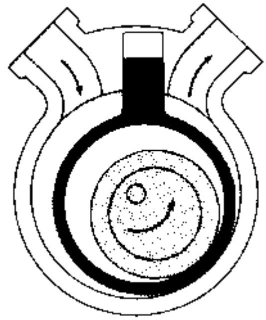

FIGURE 4 Flexible liner pump

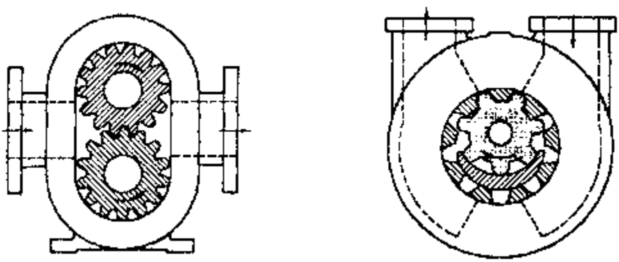

FIGURE 5 A typical external (left) and internal (right) gear pump

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

Gear pumps are classified as *external* or *internal* (see Figure 5), and external gear pumps can be either *timed* or *untimed*. External gear pumps have their gear teeth cut on their external or outside diameter and mesh about their outside diameters. Bearings support the shafts at both ends with the gears located between the bearings. This resists shaft deflection and contact between the gears and casing wall, enabling the pump to operate at higher pressures and with less overall wear over time than would otherwise be possible.

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

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

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

Both gears share pumping torque, and the proportional amount of the total torque experienced by each gear at any instant is determined by the locus of the fluid seal point between the gear teeth. As this fluid seal point moves toward the center of gear rotation, the pumping torque on that gear increases, and as the seal point moves away from the center, the torque decreases. When external timing gears are used, they transfer torque from one rotating assembly to another to safeguard against accelerated wear when dry running or handling low-lubricity or abrasive fluids.

A special form of gear pump illustrated in Figure 6 is known as a *screw-and-wheel pump*. The driving gear is helical, and the driven gear is a special form of a spur gear. The helical gear always is the driving, or power, rotor in this type of pump, and external timing gears are not used. The pumping torque in the screw-and-wheel pump is felt both by the screw and by the wheel, and the amount of torque felt by each is determined by the fluid seal contact locus points between the two rotors. As in other gear pumps, the running clearances between the rotors and between the rotors and the body walls determine leakage from the OTO volume to the OTI volume.

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

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

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

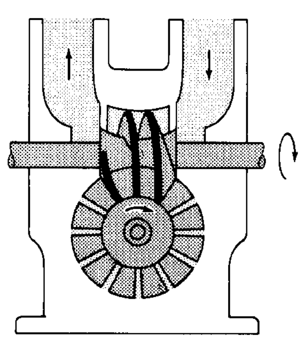

FIGURE 6 A typical screw-and-wheel pump

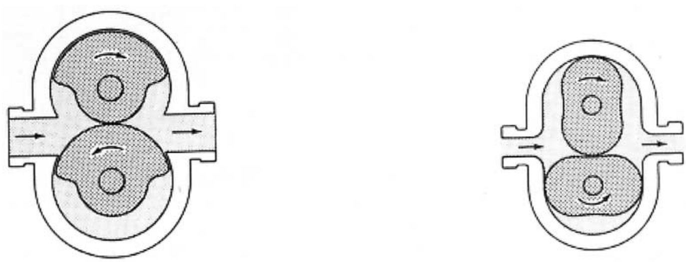

FIGURE 7 Typical single-lobe (left) and multiple-lobe (right) pumps

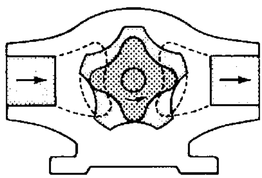

FIGURE 8 A typical internal gear, or internal lobe, pump

In the two-rotor lobe pumps shown in Figure 7, the torque is shared by both rotors with the proportional amount of torque dependent on the position of the rotor-to-rotor contact point on the rotor contact locus. When the contact point is at the major locus radius (maximum lobe radius of one rotor in contact with the minimum lobe radius of an adjoining rotor), one rotor sees the full pumping torque, while the other rotor feels a balanced torque. The transfer of a full pumping torque from one rotor to the other takes place as many times in each complete revolution of a rotor as there are lobes on the rotor.

An internal lobe, or *gerotor* pump, is shown in Figure 8 and has a single rotor with a lobe-like peripheral shape. It moves in a combination of rotations and gyrations about its center of rotation in a body with internal, lobe-shaped contours in such a way that the rotor always touches the body at two or more locations to preserve the fluid seal between OTI and OTO volumes. The outer rotor surface, inner body surface, and the fluid seal points between them define the OTI volume. The outer rotor surface and the inner body surface between two adjacent fluid seal points define the CTIO volume. The outer rotor surface, inner body surface, and the rotor-to-body fluid sealing points define the OTO volume.

Most pumps of this type have one fewer rotor lobe than an internal body lobe cavity and the term *progressing tooth gear pump* is sometimes used. The full pumping torque is seen by the single rotor, but the torque is cyclic. It is a function of the position of the rotor and its sealing arrangement with the pump body, while the number of torque cycles per rotor revolution is equal to the number of lobes on the rotor.

Lobe pumps are capable of flows up to about 1,000 gpm (3,785 l/min) and pressures up to 125 lb/in^2 (8.6 bar). They are commonly used to pump sludge in wastewater treatment plants and in stainless steel systems for handling foodstuffs in the food, beverage, dairy, and pharmaceutical industries.

A summary of the main application advantages of vane, gear, and lobe pumps is shown in Table 1.

COMPARISONS TO CENTRIFUGAL PUMPS

A rotary pump uses mechanical and hydraulic forces to create a flow against a system backpressure. Centrifugal pumps, on the other hand, create pressure by imparting a velocity to the fluid and converting the velocity energy to a pressure energy as the fluid flows

TABLE 1 Summary of vane, gear and lobe pump attributes

Application Advantages			
Vane	Gear (External)	Gear (Internal)	Lobe
• Can handle viscosities under 32 SSU • Self compensating for wear • Available in a variety of materials • Inexpensive	• Can handle high viscosities • High flows and pressures • Between-the-bearing design prolongs life. • Quiet running • Integral relief valve available	• Can handle high viscosities • Available in a wide variety of materials • Simple, inexpensive design • Integral relief valve available	• Can handle high viscosities • Low shear pumping • Available in a variety of materials • Can run dry if seals are flushed (due to timing gears)

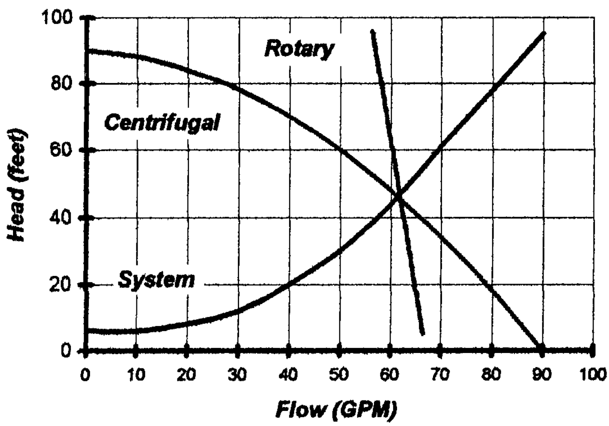

FIGURE 9 Typical performance curves (m = ft × 0.3048; m^3/h = gpm/4.403)

around the casing and out the discharge nozzle. A comparison of the resulting performance of these two different pump types is shown in Figure 9.

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

Other differences exist between centrifugal and rotary pumps as well. The performance curves, affinity laws, and terminology used to describe rotary pumps are all different. And since rotary pumps are primarily for viscous fluids, the applications and markets served by these two pumps are also different. One of the few direct comparisons that can be made between centrifugal and rotary pumps is with single- versus multi-stage pumps. Even here though, the analogy is not a perfect one, and certain rotary pumps, such as progressing cavity pumps, fit the description better than others. Other factors, such as vertical versus horizontal mounting, metal versus non-metallic materials, sealless and magnetically driven versus dual-containment mechanical seals and conventional drivers, are similar whether considering a centrifugal or rotary pump.

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

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

Other considerations, such as flat versus steep curves or matching a system curve to a pump curve, also cannot be applied to rotary pumps the way they can with a centrifugal

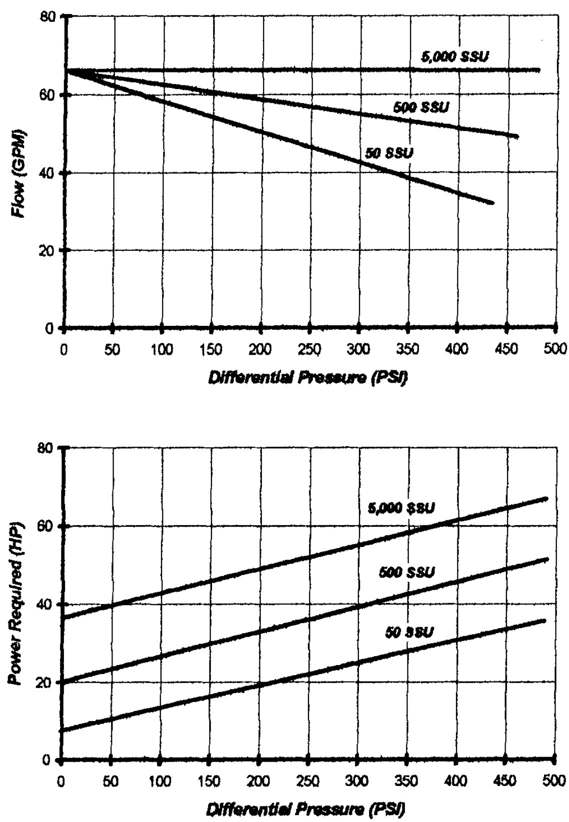

FIGURE 10 Typical rotary pump flow and power curves (bar = psi/14.5; kW = hp × 0.746; 50, 500, 5000 SSU = 7, 108, and 1080 centistokes respectively.)

pump. Instead, a rotary pump is selected for a given differential pressure and viscosity at the nearest commercially acceptable speed. For viscosities under 3,000 SSU (650 centistokes), this means at synchronous motor speeds with the flow rate falling at the design point or (ideally) slightly above it. For higher viscosities, rotary pumps will be run at reduced speeds, which in some cases can go well below 100 revolutions per minute (rpm).

Performance with Viscous Fluids Centrifugal pumps assume the absence of any appreciable viscous drag across the impeller shroud and vane surfaces when developing their pumping action. However, viscous drag, as the fluid passes across these surfaces, can be considerable with higher viscosities. With an increasing viscosity, an increasing amount of energy must be expended to overcome these forces and produce the same amount of hydraulic work. At some point, viscosity will simply overtake the centrifugal pump, and it can no longer overcome the inertia and viscous drag losses of the fluid. The effect of this can be seen in Figure 11.

Since rotary pumps do not work this way, they are better suited for high-viscosity fluids. Certain rotary pumps, such as twin screw pumps, can even go as high as 1,000,000 SSU (216,000 centistokes) without a significant deterioration of performance or efficiency. Compared to centrifugal pumps, rotary pumps are less efficient at lower viscosities but more efficient at higher viscosities. From an applications standpoint, this crossover point will vary from pump to pump and depends on many factors, but as a rule of thumb, it usually falls between 500 and 1,000 SSU (108 and 216 centistokes).

Affinity Laws Rotary pumps do not have a single *best efficiency point* (BEP) the way a centrifugal pump does, and no parallels for radial thrust loads exist as a function of the position on the curve relative to BEP the way there is with centrifugal pumps. Similarly, no rotary pump parameter is analogous to the specific speed (N_S) of centrifugal pumps; although, an approximate N_S-domain into which rotary pumps fall can be identified (see Chapter 1 and Section 2.1). Instead, rotary pumps simply use the flow per revolution to make general comparisons. A summary of the basic rotary pump affinity laws in shown in Table 2.

SYSTEM CONSIDERATIONS

The variety of applications and system conditions, including the vast number of fluids handled in rotary pump applications, precludes a comprehensive coverage in a handbook of this type. The best source for this information is the manufacturer of that particular pump type.

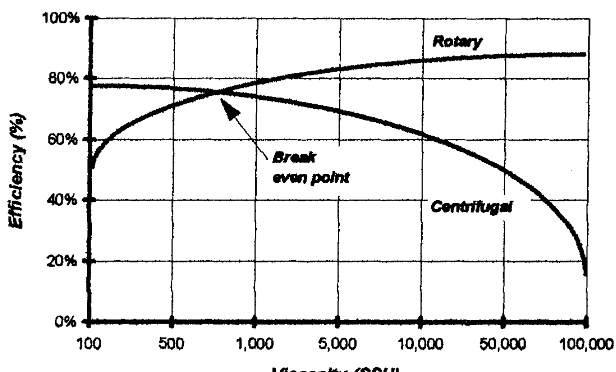

FIGURE 11 Typical efficiences as a function of viscosity. (In this range, centistokes ≈ SSU/5.)

TABLE 2 The basic affinity laws for rotary pumps

Rotary Pump Affinity Laws For Speed		
	Speed Change	Corresponding Effect
	$\{RPM_2 / RPM_1\}$	$\{GPM_2 / GPM_1\}$
	$\{RPM_2 / RPM_1\}$	no direct effect on differential pressure
	$\{RPM_2 / RPM_1\}$	$\{BHP_2 / BHP_1\}$
	$\{RPM_2 / RPM_1\}$	$\{NPSH_2 / NPSH_1\}^X$ where X varies from 1.5 to 2.5

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

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

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

The sealing arrangement on the pump must also be checked for any leakage of air or gas into the pump through the seal. If these conditions are satisfied, fluid will enter the pump soon after pump operation starts because of the vacuum generated by pump operation. If these conditions are not met, air or other gases will flow through leaks in the inlet system or seal to satisfy the pump flow rate requirements and the pump will run dry. One safeguard to ensure fluid at the pump inlet is to install a foot valve in the submerged portion of the inlet piping. Once primed, the foot valve will keep fluid in the pump and prevent it from running dry upon subsequent restarting.

Suspect Strainers

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

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

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

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

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

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

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

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

Inlet Pressure The absolute pressure above the vapor pressure *available* at the pump inlet must always exceed the absolute pressure above the vapor pressure *required* by the pump. For rotary pumps, this pressure is determined by Hydraulic Institute standards similar to those used for centrifugal pumps.

Another consideration is the effect of a net negative total differential pressure. This can occur when there is a variable positive static pressure on the inlet that exceeds the discharge or outlet pressure. In this case, the flow slip reverses direction and actually adds to the capacity of the pump, causing the total flow through the pump to be greater than the pump displacement capacity.

This can also occur with a stopped pump, and rotary pumps are not effective at stopping the flow through them. In applications where the flow cannot be permitted through an idle pump, or where inlet or outlet static pressure heads exist, valving in the system must be used to stop the flow. For example, in intermittent deliveries where the pump is "lifting" liquid from a source below its inlet without any valving in place to address this when the pump is stopped, the fluid will gradually drain backward through the pump and back to its source. This could create errors in measuring the amount of fluid transferred or cause the pump source (a holding tank) to overflow.

FLUID CONSIDERATIONS

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

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

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

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

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

Lowering the static suction lift of a system, increasing the suction pipe diameter, and simplifying the suction piping layout can reduce cavitation by raising the net inlet pressure

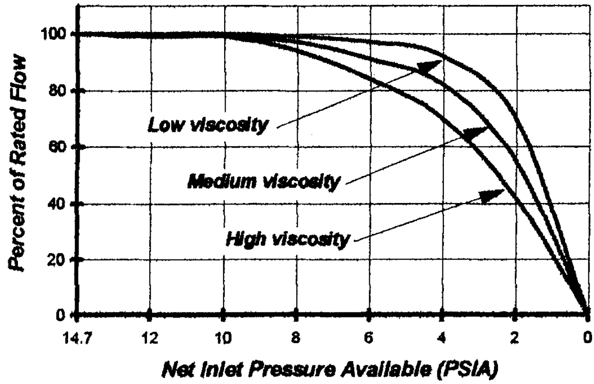

FIGURE 12 How viscosity and net inlet pressure available influence flow (bar = psi/14.5. Expressed as head of pumped liquid, the net inlet pressure is called *NPSH*.)

or *NPSH* available. In reality though, the net inlet pressure available is not something that can easily be altered. Instead, the pump vendor will usually be asked to select a pump with a lower net inlet pressure required. The best way to do this is to select a larger pump running slower. This will give the fluid more residence time to fill the void on the suction side, and with larger internal passages and ports, pump entry losses will be reduced as well.

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

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

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

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

Abrasiveness Abrasives present a problem for every pump type, and rotary pumps are no exception. With small operating clearances, the tendency of the abrasive is to wear down and open the tight clearances of the pump. Most rotary pumps' internal clearances are long, narrow, rectangular cross sections that can be modeled as two parallel flat plates, with one plate stationary and the other moving. These clearances range from essentially zero to a few thousandths of an inch. Thus, even minor variations in manufacturing tolerances or the effects of wear over time can cause considerable variations in the percentage change of the aperture volume. Also, the movement or deflection of movable elements in the pump, when exposed to pressure differences, can cause relatively large percentage changes in these clearances in different locations within the pump.

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

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

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

where Q_s = slip through the clearance, gpm (m³/hr)
k = a constant
P_d = differential pressure, lb/in² (bar)
w = width of clearance, in (mm)
d = clearance gap or depth, in (mm)
v = absolute viscosity, lb-sec/ft² (centipoises) [Note: (absolute viscosity in centipoises) = (kinemetic viscosity in centistokes) × sp. gr.]
l = length of clearance, in (mm)

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

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

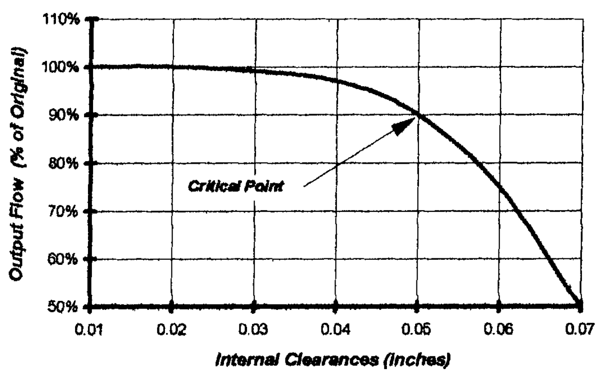

FIGURE 13 Output flow rate as a function of clearance size and wear

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

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

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

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

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

OPERATING CHARACTERISTICS

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

Displacement The *theoretical* or *geometric displacement* Q_d of a rotary pump is the total gross fluid volume transferred from the OTI volume to the OTO per unit time. A standard unit of displacement is gallons per minute (cubic meters per hour). For any given pump, the displacement depends only upon the physical dimensions of the pump elements and the pump geometry and is independent of other operating conditions. In those pumps designed for variable displacement, the pump usually is rated at its maximum displacement.

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

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

The average slip for any set of operating conditions can be found by measuring the flow rate from the outlet port (assuming an incompressible liquid) and subtracting that flow rate from the theoretical displacement flow rate Q_d that would otherwise be expected at those operating conditions. Most slip paths are constant in width but may vary in height with runout of the outside diameter of the rotors or wobble of the end faces of the rotors as they rotate. The paths can also vary considerably in length with the changing positions of the rotary and body-sealing surfaces during rotation.

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

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

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

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

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

The theoretical displacement D per revolution (where $Q_d = ND$ and N = revolutions per unit time) can be found by integrating the differential rate of a net volume transfer over one shaft revolution with respect to the angular displacement of the drive shaft through any complete planar segment taken through the pump chamber between the inlet and outlet ports. Most pump rotors have constant radial dimensions in the axial direction in the body cavity and sweep a right circular cylinder of volume while rotating. This means in single-rotor pumps or in multiple-rotor pumps where no sealing contact exists between rotors (all dynamic seals are formed between rotor elements and body surfaces), the volume transfer computation can be based on polar coordinates centered on each rotor axis and the contribution to the net volume transfer found for each rotor independently.

In general, the axial dimension of the rotor in the body cavity can be most simply expressed if the planar segment is taken through the rotor axis, or at least parallel to the rotor axis. Also, for most types of rotary pumps, the computation is simplified if the intersections of the plane with the body cavity occur in a CTIO region, usually midway between the inlet and outlet ports of the pump. This is particularly true for those rotary pumps that pump equally well in either direction of rotation and are generally symmetric. In many cases, the computation can be further simplified by separating the differential statement for volume transfer through the plane from the inlet to the outlet from that for volume transfer through the plane from the outlet to the inlet and expressing the results as a difference. Examples of this method of computation for some commonly used types of rotary pumps follow.

A section through a vane pump is shown in Figure 14. Let z be the axial distance toward the front endplate from the rotor end surface next to the rear endplate, and let Z be the total axial length of the rotor. Let r be the radial distance from the rotor axis. Let R_1 be the minimum radial dimension of the rotor elements at the intersection of the plane with the minor cam radius of the pump chamber in the CTIO zone. Let R_2 be the maximum radial dimension of the rotor elements at the intersection of the plane with the major cam radius of the pump chamber in the CTIO zone. Let ϕ be the angular displacement of the drive shaft (assumed to be direct-coupled to the rotor with no gear increase or decrease). Then, the general equation for D is

$$D = \int_{\phi=0}^{\phi=2\pi} \int_{r=R_1}^{r=R_2} \int_{z=0}^{z=Z} krd\phi drdz = k\pi Z(R_2^2 - R_1^2) \tag{3}$$

where k is a constant used to convert D to desired units ($k = 1$ if z and r are in feet and D is in cubic feet per revolution). For vanes with non-zero thickness, the actual value of D will be slightly smaller than this.

The equation describing the transition of the major radius cam surface to the minor radius cam surface is not used or needed in Equation 3, because the planar segment is entirely in the CTIO zone of the pump. Also, the integration limits for r were chosen by noting that the net volume transfer for all $r < R_1$ cancels and equals zero. The same result is obtained if the integral is expressed as the difference of the positive contribution of the integration limits 0 to R_1.

The same computations and formula apply to flexible vane pumps with the vanes on the rotor and to any vane-in rotor pump where the surface creating the pumping action is

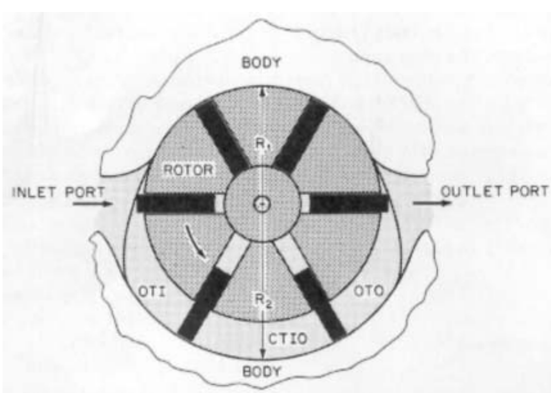

FIGURE 14 Displacement calculation dimensions for an internal vane pump

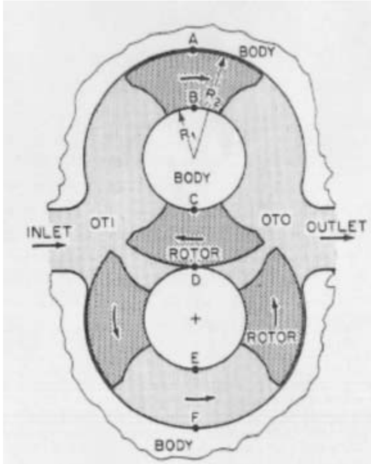

FIGURE 15 Displacement calculations for an external circumferential piston pump

formed on the interior surface of the body chamber. The equation can be used for most single-rotor positive displacement pumps with an appropriate selection of the numerical values for R_1 and R_2. For example, in the vane-in-body pump, R_1 is the minimum radius of the rotor cam and R_2 is the maximum radius of the rotor cam.

An example extending this method of computing displacement to multiple-rotor pumps that have no rotor-to-rotor sealing contact is the computation on the external circumferential piston pump shown in Figure 15. A derivation can be made in the same manner used for the vane pump, and the individual equations for the net volume flow for each rotor can be computed and summed.

The pump in Figure 15 is symmetrical, and the net contribution of each rotor occurs only during a total of half of the drive shaft revolution. In the derivation shown in Equation 3, this would mean a change in the integration limits of ϕ from 2π to π for the contribution of each rotor. However, when the contributions from each of the two rotors are added, the result is the same as Equation 3.

Another method could also be used. Because of the pump symmetry, an inspection shows that two net volume components are continually transferred from the inlet to the outlet chamber by the motion of the rotors in zones *A–B* and *E–F*. One equal volume element is also continually transferred from the outlet to the inlet chamber of the cooperative action of the rotors in zone *C–D*. Consequently, the net volume being transferred from the outlet to inlet in zone *C–D* is continually canceled by one of the volumes continually being transferred from the inlet to outlet at either zone *A–B* or zone *E–F*. The entire computation then can be made only for zone *A–B* or zone *E–F* over the entire revolution. The resulting formula for displacement would be that given in Equation 3, with R_1 being the minimum radial dimension of the piston element of the rotor and R_2 being the maximum radial dimension of the piston element of the rotor.

The direct computation of displacement for multiple-rotor, positive displacement pumps where a moving seal is formed by contact between rotors is much more complex. In such pumps, the equation of motion of the locus of the contact point between rotors as a function of angular displacement of the drive shaft is needed for a rigorous solution of the displacement. The differential rate of volume transfer is not constant with angular displacement but decreases as the contact locus moves toward the inlet chamber and increases as the contact locus moves toward the outlet chamber. In this case, a graph of pump displacement versus angular displacement would not be a straight line (or constant function of angular displacement), as in the prior two examples. In effect, the motion of the contact locus superimposes a ripple on the steady-state component of the differential rate of volume transfer.

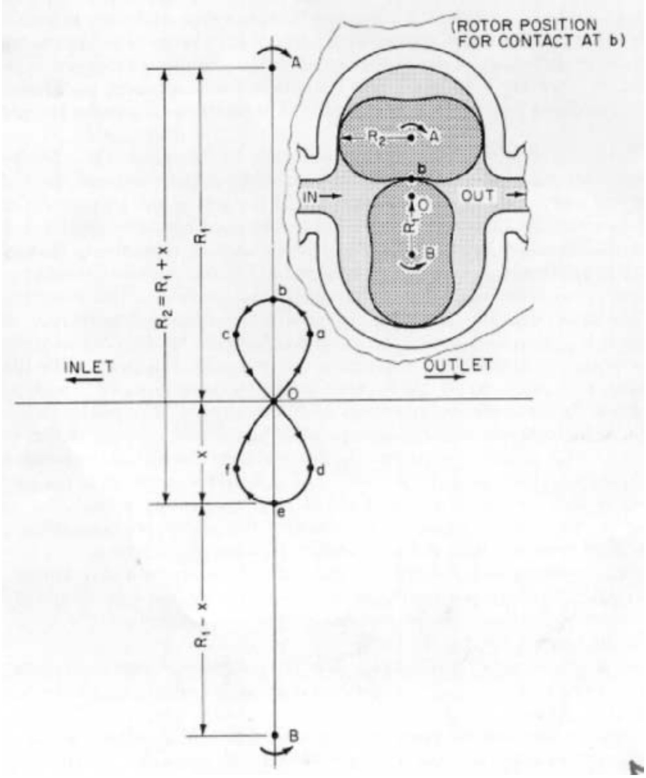

FIGURE 16 Rotor contact locus of lobe pump

Figure 16 shows the locus of the contact line between lobes in a two-lobe pump. The locus is shown on a plane taken perpendicular to the planar segment used for computing displacement, which is taken through the axis of both rotors with the center of rotation of the two rotors being points A and B. Point O is midway between the two rotor centers. The lower-case letters represent essential points in the differential rate of volume transfer.

Remembering that velocity of the movement of the locus point toward or away from the outlet port determines the amount of plus or minus deviation from the average differential transfer rate, the following observations can be made.

The maximum instantaneous differential volume transfer rate occurs as the locus passes through point O on its way from c to d or from f to a. The minimum instantaneous rate of differential transfer occurs when the locus passes through point b or point e. The average differential rate of transfer occurs at points a, c, d, and f. In the symmetric case shown, if R_1 is the distance from O to A or B, if x is the distance from O to b or e, and if R_2 is the maximum radial dimension of a lobe, then derivations similar to those given before will give the maximum and minimum differential rates of volume transfer:

$$\frac{dD}{d\phi_{\max}} = kZ(R_2^2 - R_1^2) \tag{4}$$

$$\frac{dD}{d\phi_{\min}} = (kZ(R_2^2 - R_1^2 - x^2) = kZ[2(R_2R_1 - R_1^2)]$$

since, for the lobe pump shown, $R_2 = R_1 + x$.

In most pumps of practical design, the peak-to-peak amplitude of the ripple is less than 10 percent of the steady-state component of displacement. Consequently, computing the peak displacement (where R_2 is the maximum radial dimension of the rotor and R_1 is half the distance between rotors of equal size) can approximate the displacement for multiple-rotor pumps with contacting rotors. The equation is given as

$$D_{av} = 2\pi kZ\left(\frac{R_2^2}{2} + R_2R_1 - \frac{3}{2}R_1^2\right) \tag{5}$$

$$D_{\min} = 4\pi kZ(R_2R_1 - R_1^2)$$

$$D_{\max} = 2\pi kZ(R_2^2 - R_1^2)$$

If the rotors are of unequal size, the transfer rate must be computed for each rotor with R_1 taken as the radius of the pitch circle of the rotor. This also applies to gear pumps where R_1 is the radius of the pitch circle of the gears. Displacement computed by this simplified method will usually be within five percent of the true average displacement for lobe pumps and within one percent for gear pumps. For a closer approximation, Equation 5 can be used. It is precise only when the ripple waveform has a zero average component and when it is symmetric.

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

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

The most common unit of speed is rpm, and its direction is usually described as either clockwise or counterclockwise when viewed from the front or drive end of the pump looking at the pump shaft. The economy of manufacturing, special pressure-balancing arrangements, relief valve orientations, and other such considerations have led to some pump designs requiring that the pump operate in one direction only. However, most rotary pumps will operate equally well in either direction.

Usually, no minimum speed exists for a rotary pump, but certain vane pumps that depend on a centrifugal force to draw the vanes out of their slots do have minimum speed requirements. Another exception is with low-viscosity fluids at high pressures where the slow speed enables slip to equal the theoretical displacement, resulting in no net output flow. To avoid this situation and assure the pump is running at a suitable volumetric efficiency, the pump should always run within its permitted speed range. Figure 17 shows a typical speed versus viscosity curve for a rotary pump.

Since flow is directly proportional to speed, it can be tempting to increase the pump speed above its maximum speed to obtain more flow. However, besides all the aforementioned concerns, one other item must be considered. With any rotary pump, as the cavities rotate, they present a void for the incoming fluid to fill. This void is available for a fixed amount of time, and the fluid in the suction chamber must accelerate to fill this void in the available time. The higher the fluid viscosity, the more energy is required to accelerate the fluid to fill the void. If the fluid cannot fill the void in the time available, a partial void and cavitation will occur.

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

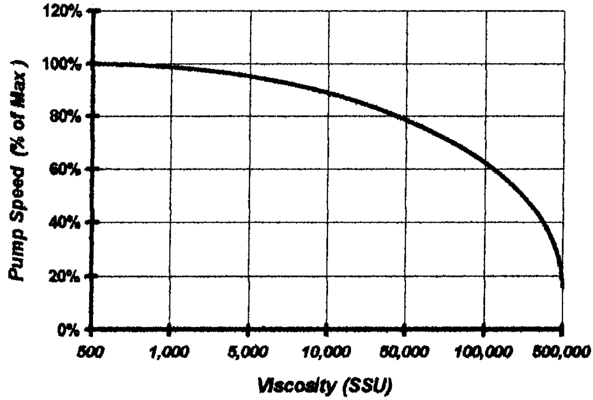

FIGURE 17 Typical speed reduction as a function of viscosity (In this range, centistokes = SSU/4.6)

In addition, where the fluid velocities (at the same pump locations used to determine the pressure differences) are sufficiently alike, they will cancel each other out. Should this not be so, velocity or dynamic pressure P_{vel} can be computed as

in USCS units
$$P_{vel} = \frac{wV^2}{288g} \tag{6}$$

in SI units
$$P_{vel} = \frac{wV^2}{20{,}387g}$$

where P_{vel} = dynamic pressure of the fluid being pumped
w = fluid specific weight (mass), lb/ft^3 (kg/m^3)
V = fluid velocity, ft/s (m/s)
g = acceleration due to gravity, ft/s^2 (m/s^2)

or can alternately be expressed as

in USCS units
$$P_{vel} = 0.000357\left(\frac{w}{g}\right) \times \left(\frac{Q}{a}\right)^2 \tag{7}$$

in SI units
$$P_{vel} = 13{,}600\left(\frac{w}{g}\right) \times \left(\frac{Q}{a}\right)^2$$

where Q = flow, gpm (m^3/min)
a = cross-sectional area perpendicular to flow, in^2 (mm^2)

Several pressure terms are of interest. The *outlet* or *discharge pressure* P_o is the total pressure at the outlet of the pump. In pumps with multiple outlets, this pressure is usually defined at the location in the outlet manifold where the pump is mated to the external piping system. Although composed of the sum of system and velocity pressures external to the pump, the outlet pressure is most commonly expressed as the gage pressure at the outlet port. The gage pressure (lb/in^2 gage or bar) is the difference between the absolute pressure and atmospheric pressure at the point of measurement.

The *inlet* or *suction pressure* P_{in} is the total pressure at the inlet to the pump or, for multiple-inlet pumps, at the manifold location where the pump connects to the external piping system. In common practice, the inlet pressure can be variously expressed as absolute pressure (lb/in^2 absolute or bar), as positive or negative gage pressure (lb/in^2 gage or bar), or as vacuum (inches or millimeters of mercury).

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

$$P_d = P_{out} - P_{in} \tag{8}$$

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

$$P_{net\ in} = P_{in} - P_{vapor} \tag{9}$$

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

The required net inlet pressure is established by the manufacturer for particular speeds, pressures, and fluid characteristics. In practice, the pump user is warned that the net inlet pressure is near or at the required net inlet pressure by noisy and rough operation of the pump caused by the incomplete filling of the CTIO volume with an accompanying reduction in pump flow rate.

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

Power Most of the pump *input shaft power* HP_{in} (kW_{in}) is the power imparted to the fluid delivered by the pump at given operating conditions and is frequently called *liquid* or *hydraulic power* HP_{hyd} (kW_{hyd}). A common unit used for expressing power ratings is horsepower (kilowatts) and can be computed by the equation:

In USCS units
$$HP_{hyd} = \frac{Q_d \times P_d}{1714} \tag{10}$$

In SI units
$$kW_{hyd} = \frac{Q_d \times P_d}{36}$$

where the constant 1,714 (36) gives the hydraulic power input in *hp* (*kW*) when Q_d is in gpm (m^3/h) and P_d is the differential pressure in lb/in^2 (bar). This is the theoretical power or the required power independent of mechanical friction power losses, and fluid friction power.

The *mechanical friction power* losses HP_{losses} (kW_{losses}) have two components. One is the mechanical friction power required by the elements outside the pumping chamber, such as the bearings and seals, and is usually independent of the fluid being pumped, the pump speed, and the differential pressure. The mechanical friction power inside the pumping chamber depends also on the pump speed and pump differential pressure, but it also depends on the viscosity and lubricity of the fluid.

The *fluid friction power* losses, or *viscous power* losses, are a function of the viscosity of the liquid being handled and on the shear rate in the fluid, which is a function of pump design and pump speed. When handling high-viscosity liquids, the viscous friction power grows with viscosity, even if speed and pressure remain constant. This power loss depends on a number of design features of the pump and usually will grow proportionally to the viscosity to the *nth* power, where n usually ranges between 0.3 and 1.0 for most rotary pumps. Increasing any clearances in a pump reduces the shear stress in the liquid in those clearances at any given speed and consequently reduces the torque to overcome the viscous friction.

The *useful output power* HP_{out} is less than HP_{hyd} by an amount $HP_{\text{slip loss}}$ due to leakage Q_{slip} through the clearances. Thus, *overall efficiency* $E_{\text{overall}} = HP_{out}/HP_{in}$, where (for SI units, substitute kW for HP)

$$HP_{in} = HP_{\text{slip loss}} + HP_{\text{mech loss}} + HP_{\text{friction losses}} + HP_{out} \tag{11}$$

Efficiency Rotary pumps are measured by their *volumetric*, *mechanical*, and (as just defined) *overall* efficiencies. This often requires certain trade-offs when selecting a pump for a given set of conditions since a rotary pump does not have a *single best efficiency point* (BEP) the way a centrifugal pump does.

Volumetric efficiency E_{vol} compares actual to theoretical output flows. It is an indication of both a pump's capability to handle a given differential pressure and viscosity and, in the case of older pumps, an indication of possible internal wear. The more flow a pump can deliver under these conditions, the higher its volumetric efficiency will be. The volumetric efficiency of a rotary pump is defined as

$$E_{vol} = \frac{Q_d - Q_s}{Q_d} \tag{12}$$

where Q_d = displacement or geometric flow delivered, in gpm (m³/hr)
Q_s = slip flow, in gpm (m³/hr)

The positive displacement nature of rotary pumps makes them suitable for many metering applications. The pump would be a perfect metering device with zero errors if there were no slip. It would be considered a dependable meter if the slip were low or maintained constant over the entire range of operating conditions. A useful ratio in a comparison of different types of rotary pumps for a given metering application is the ratio of minimum flow rate in any of the operating conditions to the maximum flow rate in any of the operating conditions at a given speed. In variable-speed applications, the ratio of minimum volumetric efficiency to maximum volumetric efficiency is substituted. The metering effectiveness is highest as this ratio approaches unity. The difference between this ratio and 1×100 is the percentage of change that can be expected either in the flow rate of the pump or in the total amount of fluid displaced for a given number of revolutions under extremes of operating conditions (such as pressures, temperatures, viscosities, and so on)

Volumetric efficiencies fall in the range of 65 to 98 percent for most rotary pumps. A changing volumetric efficiency over time usually indicates changing system conditions of service or pump wear.

Mechanical efficiency E_{mech} is the ratio of the pump input hydraulic power to input shaft power. It can also be looked at as the comparison of theoretical input power required for the actual power required. The less power a pump requires to produce a given amount of hydraulic work (output), the higher its mechanical efficiency. The total volume of fluid Q_d handled by the pump is larger than Q when slip is present. The amount of slip actually represents wasted power and affects pump efficiency. The difference between shaft power input and hydraulic power input consists of the power lost to mechanical friction and the power lost to fluid friction (a function of the viscosity of the fluid and pump shear stresses on the fluid). The mechanical efficiency of a rotary pump is found by the equation

$$E_{mech} = \left\{\frac{HP_{hyd}}{HP_{in}}\right\} = \left\{\frac{HP_{hyd}}{HP_{hyd} + HP_{\text{losses}}}\right\} \tag{13}$$

Overall efficiency $E_{overall}$ is the most important efficiency because it alone determines the overall effectiveness of a pump for an application. That is, knowing only the volumetric or mechanical efficiencies can result in misleading conclusions. Only with a satisfactory overall efficiency one can be reasonably assured of overall satisfactory performance. Overall efficiency is defined in terms of E_{mech} and E_{vol} as follows:

in USCS units
$$E_{\text{overall}} = \left\{\frac{Q_d - Q_{\text{slip}}}{Q_d}\right\} \times \left\{\frac{HP_{hyd}}{HP_{hyd} + HP_{\text{losses}}}\right\} \tag{14}$$

in SI units
$$= \left\{\frac{Q_c}{Q_d}\right\} \times \left\{\frac{kW_{hyd}}{kW_{in}}\right\}$$

$$= E_{vol} \times E_{mech}$$

Another way of looking at overall efficiency is

$$E_{\text{overall}} = \left\{\frac{Q_{\text{actual}}}{Q_{\text{theoretical}}}\right\} \times \left\{\frac{HP_{\text{theoretical}}}{HP_{\text{actual}}}\right\} \tag{15}$$

Pump Performance Figures 18, 19, and 20 show the change in displacement capacity Q_d, flow rate or capacity Q_c, and slip Q_s as the differential pressure across the pump P_d, the viscosity v, and the pump speed N are varied. Several assumptions are made in these graphs. It is assumed that inlet conditions are satisfactory and that there is no inlet effect on the pump capacity over the charted range. It is assumed that the fluid is Newtonian and the liquid is incompressible. In Figures 18 and 19, it is assumed that viscosity is constant and relatively low, and that the speed is within the normal range of the pump. In Figure 20, it is assumed that both pressure and speed are within the normal ratings of the pump.

The graph in Figure 18 is plotted with the flow rate, slip, and displacement flow rate on a linear scale as the ordinate (Y-axis) and pressure on a linear scale as the abscissa (X-axis). A further assumption is that the clearances and viscosity are such that the slip increases proportionately with pressure. The solid lines are the ideal characteristics when secondary effects are neglected. It may be noted that at zero pressure, slip Q_s is zero and Q_c equals Q_d. As the pressure increases, Q_s increases until it equals Q_d at pressure B. If the pressure imposed on the pump were to increase past this point, Q_s would exceed Q_d and the flow through the pump would be from outlet to inlet, causing a negative Q_c.

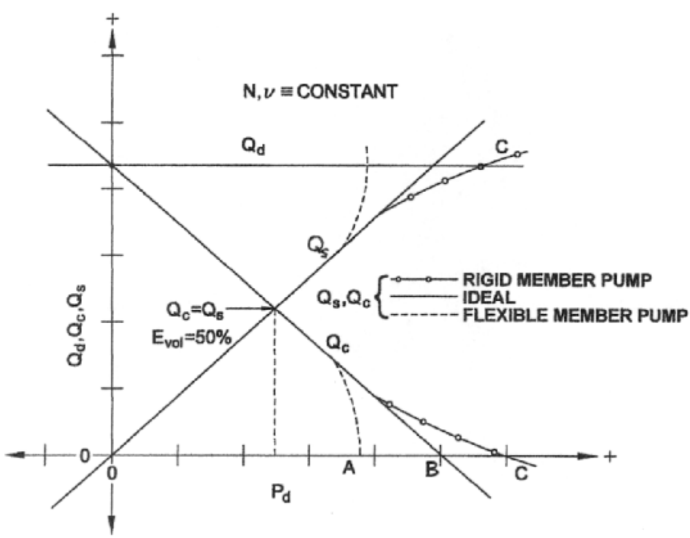

FIGURE 18 Varation of Q_d, Q_c, and Q_s with P_d, N and v constant

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

Pressure *B* data is usually not available from pump manufacturers since these values are usually far beyond the normal safe pressure rating of the pump. However, they can be estimated by extrapolating the data given. Cautions should be observed, however. A flexible member can quickly reach a pressure at which flexible member deflection becomes excessive, and the pressure at which Q_c equals zero is very quickly reached. This is illustrated in Figure 18 by the dashed line breaking away from the solid *Q* line and intersecting the abscissa at pressure point *A*. This characteristic of flexible member pumps provides a self-limited maximum pressure should the pump be dead-ended. In rigid rotor pumps, as the pressure increases beyond the normal operating range, deflections of the shafts and rotors bring the rotors into heavy bearing contact with the body chamber walls, reducing the dimensions of the slip clearance path. The dashed line leaving the Q_c and Q_s curves and intersecting the abscissa and Q_d line at pressure point *C* illustrates this. Consequently, the zero flow pressure, which is ideally at point *B*, may be considerably different from this extrapolated value, and it should be measured if it is important to the application.

Under the same assumptions given for Figure 18, Figure 19 shows the relative independence of the slip with speed when differential pressure is constant. Here the chart of *Q* intersects the abscissa at the point where the speed is low enough for the displacement flow rate to be equal to the slip at the pressure of operation. The speed at which Q_d equals $2 \times Q_s$ is the speed at which Q_s equals Q_c and the volumetric efficiency E_{vol} becomes 50 percent.

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

Figure 20 shows the effect of viscosity on the slip and flow rate in a rotary pump. The graphs are on a log-log scale. In this chart, it is assumed that the pressure, speed, and viscosity combine to keep the flow through clearances of the pump in the laminar or "viscous" flow region.

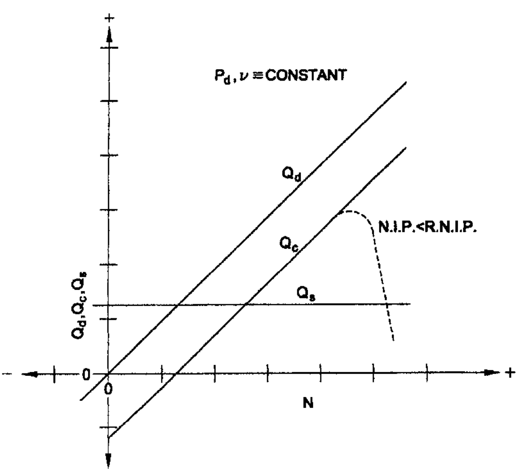

FIGURE 19 Varations of Q_d, Q_c, and Q_s with N, P_d, and v constant

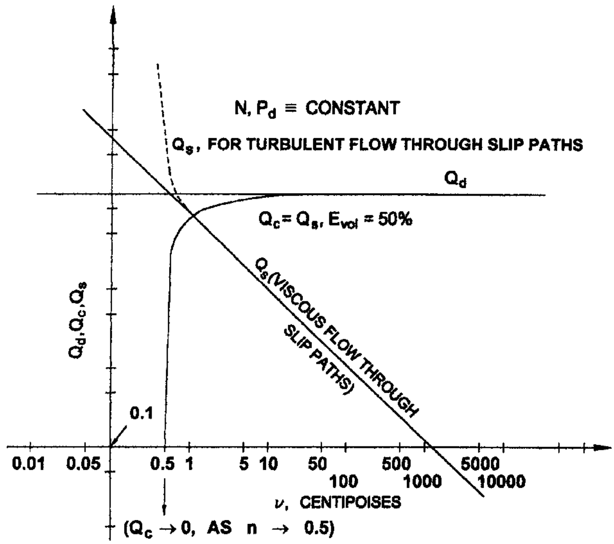

FIGURE 20 Varations of Q_d, Q_o, and Q_s with v, N and P_d constant

As viscosity increases, slip Q_s becomes very small and the flow rate of the pump Q_c approaches the displacement flow rate Q_d. As the viscosity decreases, the slip very rapidly approaches the displacement flow rate and the flow rate of the pump drops rapidly to zero or to a negative quantity. For any given pump with speed N and differential pressure P_d, there is a viscosity below which the slip flow, through clearances of the pump, will change to a turbulent flow. It is unlikely that this change will occur simultaneously through all slip paths. However, once it begins to occur, the slip will increase much more rapidly with further reductions in viscosity because of the turbulent flow relationship of the slip, pressure, and viscosity expressed in Equation 16:

$$Q_s = \frac{KP_d^{1/2}}{v^{1/x}} \tag{16}$$

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

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

The effect of inlet pressure on the capacity can be seen clearly if all secondary effects are eliminated. This assumes the pump is operating within its normal pressure limits and speed range and that viscosity is high enough to reduce the slip to a negligible value. A graph of the capacity as a function of inlet pressure is shown in Figure 21 with these assumptions.

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

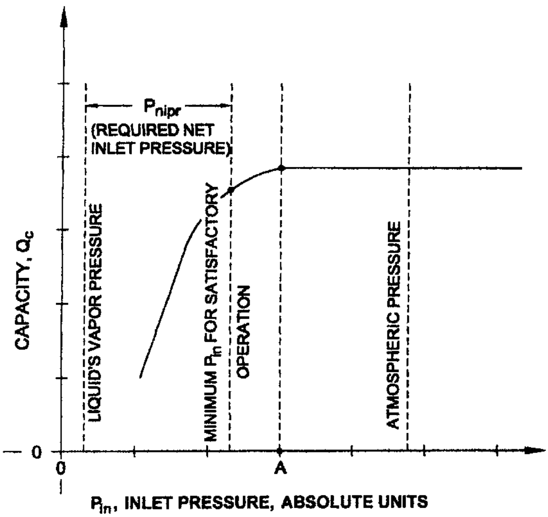

FIGURE 21 Varations of Q_c and P_{in} with other operating conditions constant

of liquid in this volume is reduced, leaving a deficiency of liquid volume. For example, if half the fluid volume swept from the inlet chamber is vapor, then only half the normal-capacity liquid volume is available at the outlet chamber, and the flow rate is reduced accordingly.

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

For lower viscosity liquids where the collapse of the vapor bubbles may be quite rapid, cavitation may cause a significant amount of damage to the body or rotor surfaces. It is important to understand that cavitation damage may occur even though the absolute inlet pressure is above pressure *A*. Locations may be found in the inlet chamber, particularly where the flow direction changes rapidly, as around sharp corners, where vapor formation in the form of very fine bubbles, and hence cavitation, may occur. However, these vapor cavities may be swept into higher pressure regions of the inlet chamber and collapse near body or rotor surfaces to cause cavitation damage, although the capacity of the pump is not affected. For any given viscosity then, there is an upper limit to the speed at which the pump can be operated.

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

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

These two limits on pump speed are illustrated in Figure 22. Three sets of operating conditions are shown. In operating condition 1, the outlet system friction resisting liquid flow (outlet system impedance) is relatively high and the outlet pressure developed by the flow is directly proportional to the pump flow rate (laminar flow). No static head exists in the system, and the outlet pressure is developed only when the pump causes liquid to flow through the outlet system. The liquid viscosity is assumed to be low enough to permit some slip in the pump. The net inlet pressure is assumed to be above pressure *A* over the range of operations shown.

Operating condition 2 is the same as operating condition 1, except that a static head (static outlet pressure) exists in the outlet system. This head is the pressure at which the chart of pump outlet pressure in condition 2 intercepts the zero speed line.

Operating condition 3 is one in which the impedance to the liquid flow in the outlet system is relatively low, but the liquid viscosity is relatively high. In this condition, Q_c, shown

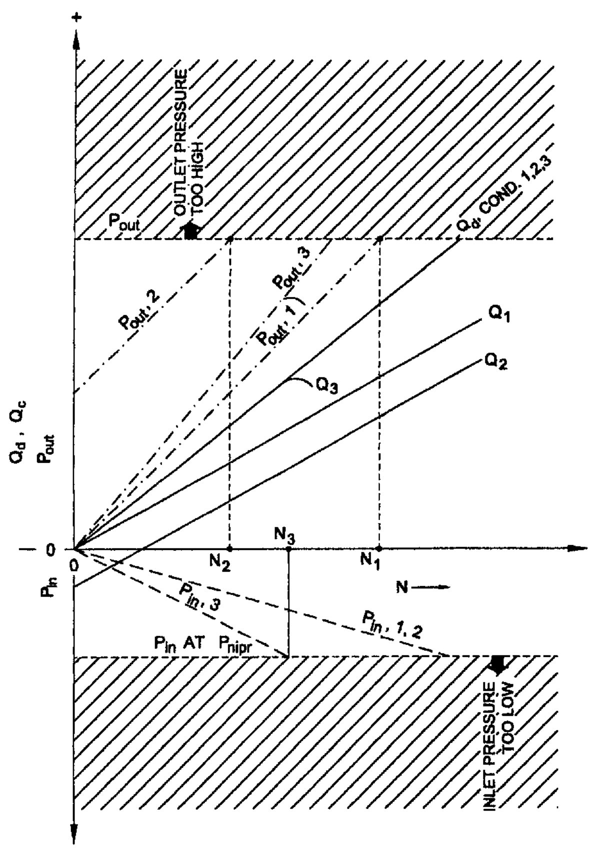

FIGURE 22 P_{in} and P_{out} limitations on pump speed

as Q_3, is equal to the displacement flow rate Q_d as speed is increased until a speed is reached at which the available net inlet pressure of the pump drops to the required net inlet pressure of the pump. If the speed is increased beyond this point, the capacity of the pump would drop rapidly as an ever-increasing part of the pump fluid becomes vapor instead of liquid.

The upper limit of speed for satisfactory operation of the pump is shown as N_1, N_2, and N_3 for the three conditions described. The locations of N_1, N_2, and N_3 on the speed axis are independent of each other because they depend primarily on the operating conditions of the pump, the system in which it operates, and on the conditions of the pumped fluid. For example, a negative head in the outlet system could cause N_3 to move to a higher value than N_2, and N_2 to move to a higher value than N_1. Operation with a liquid with lower viscosity (but still sufficient to reduce the slip to zero) could cause N_3 to be higher than N_1.

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

SPECIFICATIONS, INDUSTRIES, AND APPLICATIONS

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

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

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

Rotary pumps are used for both metering and transfer applications; so, most vane, gear, and lobe pumps are found in these services. Metering pumps tend to be smaller and operate with varying flow rates. Transfer pumps typically have a fixed speed, although variable speed units are becoming more common as the cost for this equipment falls.

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

1. **Liquid handling** In this class, well over 2,000 fluids are pumped and performance is judged on the pump's capability to handle the specific liquids, and the hydraulic power generated by the pump is a secondary consideration.

2. **Hydraulic fluid power** In this class, the hydraulic power generated by the pump is used to actuate valves, pistons, cylinders, rotary actuators, and similar devices. Pumps in this class are usually vane, gear, piston, or screw types and are designed for pressures up to 9,000 lb/in^2 (620 bar) on selected hydraulic power fluids.
3. **Commercial and retail** In this class are the pumps designed for a single specific use, application, or even fluid. Pumps in this miscellaneous class include everything from miniature pumps for home aquariums to well water pumps for agricultural use to fuel injection pumps for automobiles.

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

Markets and Applications The markets and applications served by vane, gear, and lobe pumps are among the broadest of all rotary pumps. Although it would not be possible to list every application served by these pumps, a broad sampling of the more common ones is shown in Table 3.

SUMMARY

In this section, we have discussed vane, gear, and lobe pumps in a general sense and covered many of the technical attributes that give them their unique characteristics. From an applications standpoint, the advantages these pumps have over other types can be summarized in Table 4.

TABLE 3 Markets and applications

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

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

Chemical Industry
Any process or distribution system handling
- acids
- adhesives
- bleaches
- caustics
- cellulose acetate
- glycerin
- nitrates

Cosmetics and Soap Industry
Non-emulsifying supply to packaging machinery of:
- detergent
- lanolin
- liquefied soap
- shampoo

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

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

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

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

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

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

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

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

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

TABLE 4 The basic features and applications benefits of the main rotary pumps*

Vane, Gear and Lobe Pumps	
Feature	Benefit
Flow largely independent of pressure	Predictable pump performance over varying system conditions
Wide hydraulic coverage- Flows to 10,000 GPM, differential pressures to 5,000 PSI	Few applications a rotary pump can't handle
Efficiently handles high viscosity fluids- over 100,000 SSU in some cases	Lower operating cost as efficiencies actually increase with viscosity
Smooth, pulse-free flow	Less system cost no need for vibration isolators or vibration dampeners
Self priming and will not vapor lock	Less system complexity - no need to prime or re-prime
Non-shearing pump action	Will not degrade shear sensitive polymers and petrochemicals

*10,000 gpm = 2,300 m^3/h; 5000 psi = 345 bar; 100,000 SSU =21,600 centistokes.

Many other types of rotary pumps were included in this discussion. The three types discussed here, and the numerous design variations of each type, constitute the bulk of rotary positive displacement pumps in use today.