SECTION 3.7
SCREW PUMPS

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

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

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

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

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

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

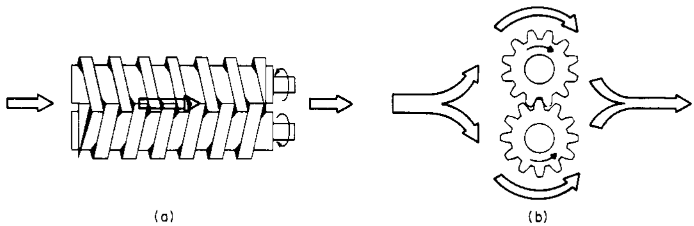

FIGURE 1 Diagrams of screw and gear elements, showing (a) axial and (b) circumferential flow.

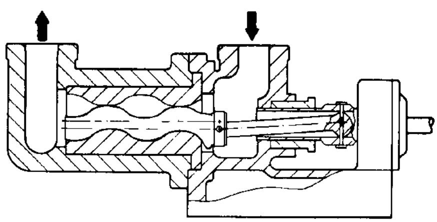

FIGURE 2 The single-screw or progressive cavity pump

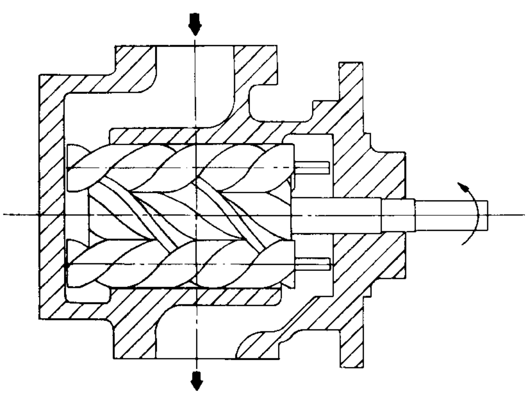

FIGURE 3 Multiple-screw single-end arrangement.

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

- A wide range of flows and pressures
- A wide range of liquids and viscosities
- High speed capability, allowing the freedom of driver selection
- Low internal velocities
- Self-priming, with good suction characteristics
- A high tolerance for entrained air and other gases

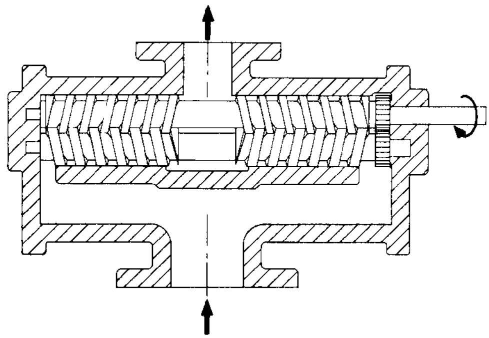

FIGURE 4 Mutliple-screw double-end arrangement.

- Low velocities for minimum churning or foaming
- Low mechanical vibration, pulsation-free flow, and quiet operation
- A rugged, compact design that is easy to install and maintain
- High tolerance to contamination in comparison with other rotary pumps

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

- A relatively high cost because of close tolerances and running clearances
- Performance characteristics sensitive to viscosity changes
- High pressure capability requires long pumping elements

THEORY

In screw pumps, it is the intermeshing of the threads on the rotors and the close fit of the surrounding housing that creates one or more sets of moving seals in a series between the pump inlet and outlet. These sets of seals or locks, as they are sometimes referred to, act as a labyrinth and provide the screw pump with its positive pressure capability. The successive sets of seals form fully enclosed cavities (see Figure 5) that move continuously from inlet to outlet. These cavities trap liquid at the inlet and carry it along to the outlet, providing a smooth flow.

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

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

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

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

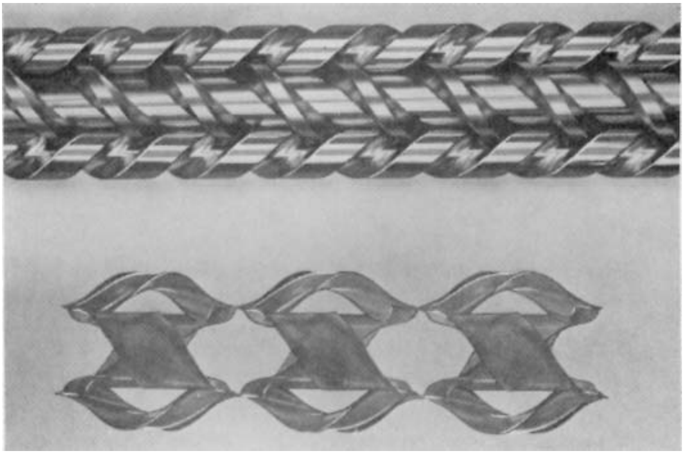

FIGURE 5 Axially moving seals and cavities. Alternate cavities filled with oil shown below. (Imo Pump)

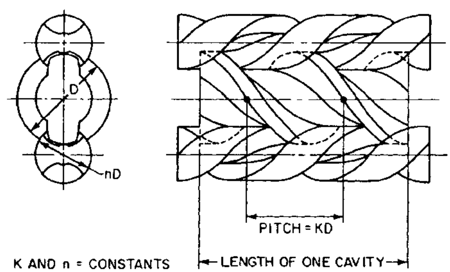

FIGURE 6 Screw-thread proportions, showing lead, diameters, and cavity length.

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

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

$$Q_t = kD^3N$$

or, bringing the pitch into evidence,

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

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

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

Slip can also be calculated,[1,2] but usually it depends upon empirical values developed by extensive testing. These test data are the basis of the design parameters used by every pump manufacturer. Slip generally varies approximately as the square of the nominal diameter. The net flow rate therefore is

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

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

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

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

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

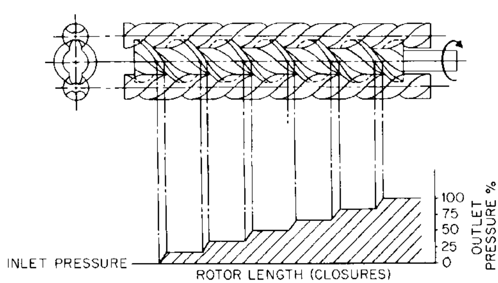

FIGURE 7 Pressure gradient along a screw set.

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

CONSTRUCTION

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

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

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

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

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

Single-End Screw Pumps All three types of screw pumps are offered in the single-end construction. As pressure requirements in many applications have been raised, the sin-

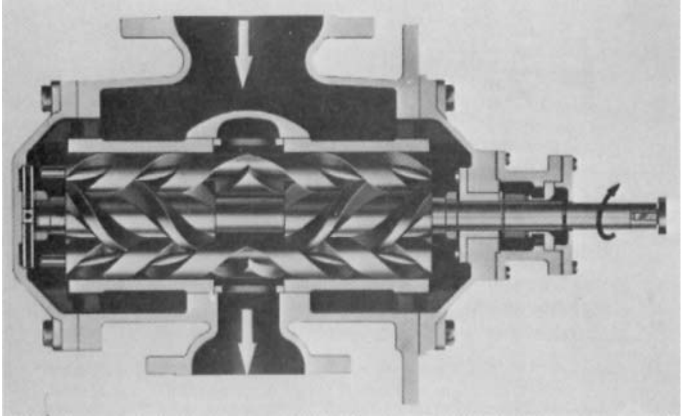

FIGURE 8 Double-end pump. Flow path provides axial balance. (Imo Pump)

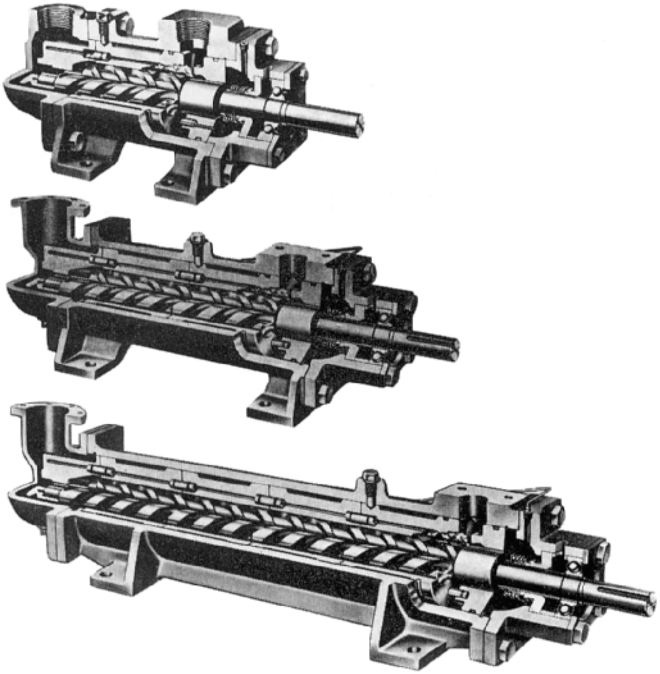

FIGURE 9 Increasing pump pressure capability by modular design (Imo Pump).

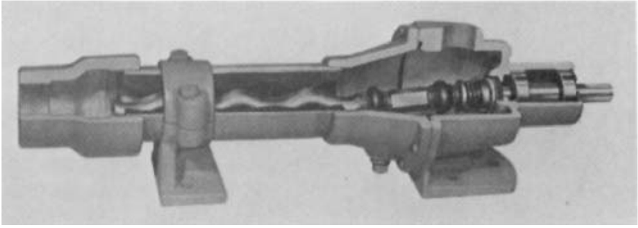

FIGURE 10 Single-rotor pump. (Robbins-Meyers)

gle-end design has come into much wider use because it provides the only practical means for obtaining the greatest number of moving seals necessary for high-pressure capabilities. The main penalties of the single-end pump are the requirement and complexity of balancing the axial loads.

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

Special mention must be made of the single-end, single-rotor design (see Figure 10). The pump elements of this design consist of only a stator and one rotor. The stator has a

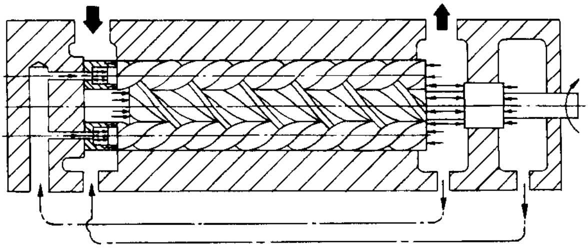

FIGURE 11 Axial balancing of power and idler rotors.

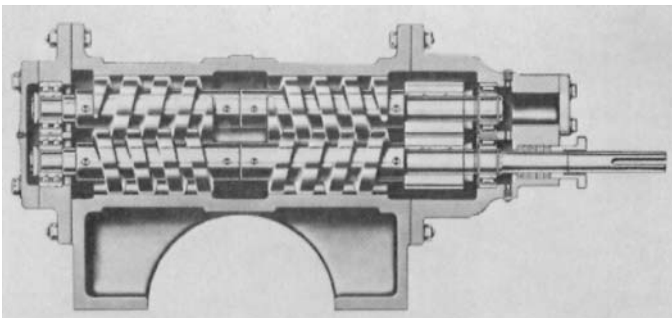

FIGURE 12 Double-end internal gear design. (Flowserve Corporation)

double-helix internal thread and is constructed of hard chrome stainless or tool steel material. One version of this design uses internally developed pressure in the pump to compress the elastomeric stator on the rotor, thus maintaining minimum running clearances.

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

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

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

The external timing arrangement is the most popular and is extensively used. It has both the timing gears and the rotor support bearings located outside the pumping chamber (see Figure 13). This type can handle a complete range of fluids, both lubricating and non-lubricating, and, with proper materials, has good abrasion resistance. The timing gears and bearings are oil-bath-lubricated from an external source. This arrangement

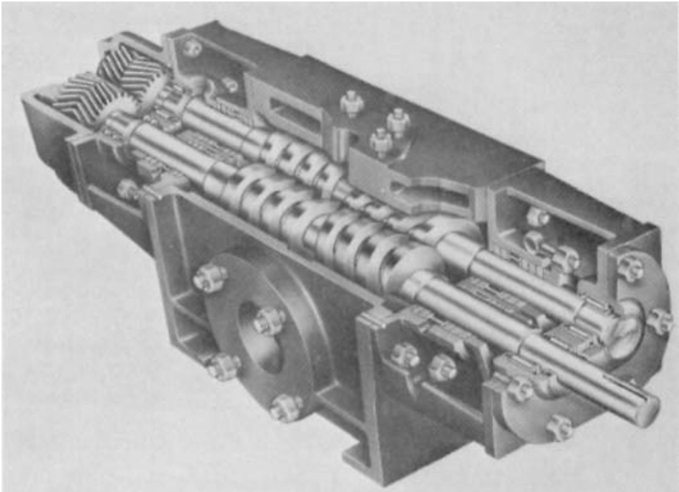

FIGURE 13 Double-end external gear design. (Warren)

requires the use of four stuffing boxes or mechanical seals, as opposed to the internal type, which employs only one shaft seal.

The main advantage of the timed screw pump is that the timing gears transmit power to the rotors with no contact between the screw threads, thus promoting long pump life. The gears and rotors are timed at the factory to maintain the proper clearance between the screws. With certain designs and loading characteristics, the bearings at each end of the rotating elements can support the rotors so that they do not come in contact with the housing; hence, no liner is required. One set of these bearings also positions and supports the timing gears. Under more heavily loaded conditions, the body bores act as sleeve bearings and provide additional support for the rotors.

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

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

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

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

The use of the rotor housing as the only means for supporting idler rotors is a unique feature of the untimed screw pump. No outboard support bearings are required on these rotors. The idler rotors in their related housing bores are, in effect, partial journal bearings

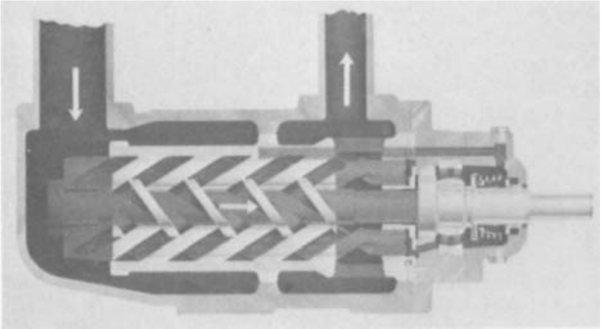

FIGURE 14 Single-end design. (Roper).

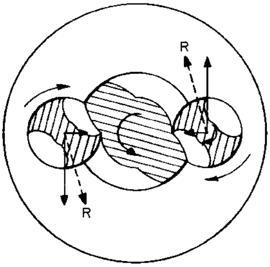

FIGURE 15 Force diagram on rotor set.

that generate a hydrodynamic film. The key parameters of rotor size, clearance, surface finish, speed, fluid viscosity, and bearing pressure are related as in a journal bearing. Because, in this design, the screws transmit the driving torque to the idlers and because the casing bores support the rotors, the pumped fluid must have some lubricating properties. This prevents this pump design from being used in applications with any solids or abrasives in the pumpage.

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

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

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

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

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

In many untimed designs, an antifriction bearing is employed on the shaft end of the driven rotor (refer to Figure 9) to provide precise shaft positioning for mechanical seal and coupling alignment. This bearing can be either an external grease-sealed bearing or an internal type with the pumped fluid providing the lubrication. The bearing also supports overhung loads with belt or gear drives.

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

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

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

PERFORMANCE

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

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

In any screw pump application—regardless of design—suction lift, viscosity, and speed are inseparable. The speed of operation is dependent upon viscosity and suction lift. If a true picture of these latter two items can be obtained, the problem of making a proper pump selection becomes simpler and the selection will result in a more efficient unit.

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

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

Naturally, the more viscous the liquid, the greater the resistance to the flow, and therefore the slower the rate of filling the moving cavities of the threads in the inlet. Conversely,

TABLE 1 Internal axial velocity limits

Liquid	Viscosity, SSU	Velocity, ft/s (m/s)
Diesel oil	32	30 (9)
Lubricating oil	1,000	12 (3.7)
#6 fuel oil	7,000	7 (2.1)
Cellulose	60,000	$\frac{1}{2}$ (0.15)

low-viscosity liquids flow more readily and will quickly fill the rotor cavities. It is obvious that if the rotor elements are moving too fast, the filling will be incomplete and a reduction in output will result. To obtain complete filling, the rate of liquid flow into the pumping elements should always be greater than the rate of cavity travel. Table 1 lists examples of safe internal axial velocity limits found from experience by one screw pump manufacturer for various liquids and pumping viscosities with only atmospheric pressure available at the pump inlet.

It is quite apparent from Table 1 that the pump speed must be selected to satisfy the viscosity of the liquid being pumped. The internal axial velocity is directly related to the pump speed of rotation and to the screw thread lead. The lead is the advancement made along one thread during a complete revolution of the driven rotor, as measured along the axis. In other words, it is the distance traveled by the moving cavity in one complete revolution of the driven rotor. This is also referred to as *pitch* on single start screws.

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

Vapor pressure is an important fluid property, which must always be recognized and considered. This is particularly true of volatile petroleum products such as gasoline. Crude oil is an example of a volatile fluid where the vapor pressure has been overlooked in the past when applying screw pumps. The vapor pressure of a liquid is the absolute pressure at which the liquid will change to a vapor at a given temperature. A common example is that the vapor pressure of water at 212°F (100°C) is 14.7 lb/in^2 (1 bar). For petroleum products, as will be discussed in Subsection 9.19.1, the *true vapor pressure* (TVP) at any temperature is a function of the *Reid vapor pressure* (absolute) (RVP). RVP is determined by the ASTM Standard D-323 procedure and is quoted at 100°F (38°C). TVP at this temperature is slightly higher than the RVP.

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

Net Positive Suction Head The suction conditions of screw pumps are normally defined by the *Net Positive Suction Head* (*NPSH*) available at the pump inlet. In some cases with systems open to the atmosphere, the inlet conditions can also be defined as suc-

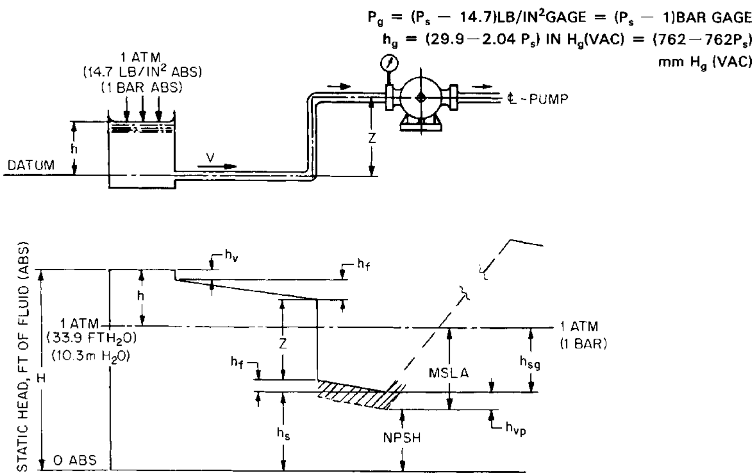

FIGURE 16 Relationship of hydraulic gradient, *NPSH,* and *MSLA.*

tion lift. Suction lift occurs when the total available pressure at the pump inlet is below the atmospheric pressure and is normally the result of a change in elevation and pipe friction. Since atmospheric pressure at sea level corresponds to 14.7 lb/in² (1 bar) absolute, or 30 in Hg (762 mm Hg), in a system with the suction open to the atmosphere, this is the maximum amount of pressure available for moving the fluid, and suction lift cannot exceed these figures. In practice, the available pressure is lower because some of it is used up in overcoming friction in the inlet lines, valves, and fittings. It is considered the best practice to keep the suction lift as low as possible to ensure that an adequate *NPSH* is available to fill the suction cavities of the pumping screw. In all cases, the system *NPSHA* (available) should be greater than the pump *NPSHR* (required). Figure 16 provides an example of *NPSH* with the following values:

NPSHA = ATM + reservoir liquid level − elevation head − frictional head loss + velocity head, where ATM is the atmospheric pressure expressed in the height of a column of the liquid being pumped.

$$NPSHA = \frac{C_1 h_b}{\text{sp. gr.}} + h - Z - h_f + \frac{V^2}{2g} - h_{vp}$$

where: $C_1 = \begin{cases} 33.9/29.92 & = 1.133 \text{ for USCS units} \\ 10.3/760 & = 0.136 \text{ for SI units} \end{cases}$

If the pressure at the inlet of the pump (P_g) is known, the *NPSHA* is calculated as

$$NPSHA = \frac{C_1 h_b}{\text{sp. gr.}} + \frac{C_2 P_g}{\text{sp. gr.}} + \frac{V^2}{2g} - h_{vp}$$

where: $C_2 = \begin{cases} 2.31 & \text{for USCS units} \\ 10.20 & \text{for SI units} \end{cases}$

If the suction pressure is a vacuum or negative gage reading, the *NPSHA* is calculated as

$$NPSHA = \frac{C_1 h_b}{\text{sp. gr.}} - \frac{C_3 h_g}{\text{sp. gr.}} + \frac{V^2}{2g} - h_{vp}$$

where: $C_2 = \begin{cases} 1.133 & \text{for USCS units} \\ 0.01360 & \text{for SI units} \end{cases}$

The maximum suction lift available is calculated as

$$MSLA = \frac{C_1 h_b}{\text{sp. gr.}} - NPSH$$

The definitions are as follows:

V = velocity, ft/s (m/s)

P_g, h_g = pressure gage readings at the pump inlet flange lb/in^2 (bar) gage and in Hg (mm Hg) (vac)

P_s = absolute static pressure at pump inlet, lb/in^2 (bar) abs

h_g, h_{sg} = static head at pump inlet, ft (m) of liquid abs or gage

Z = elevation head, ft (m) in reference to datum

h = reservoir liquid level, ft (m) in reference to datum

h_b = barometric pressure, in Hg (mm Hg) absolute

h_v, h_f = velocity head and friction head loss, ft (m)

P_{vp}, h_{vp} = liquid vapor pressure, lb/in^2 (bar) abs

P_{sv} = net positive inlet pressure, lb/in^2 (bar) abs

$NPSH$ = net positive suction head, ft (m) of liquid abs

P_f = frictional pressure loss, lb/in^2 (bar)

$MSLA$ = maximum suction lift available from pump, ft (m) of pumped liquid in the above equation—or in Hg (mm Hg) (vac)

sp. gr. = specific gravity of pumped liquid

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

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

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

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

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

Entrained and Dissolved Air As mentioned previously, a factor that must be given careful consideration is the possibility of entrained air or other gases in the liquid to be pumped. This is particularly true of installations where recirculation occurs and the fluid is exposed to air through either mechanical agitation, leaks, or improperly located drain lines.

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

One of the apparent effects of handling liquids containing entrained or dissolved gas is noisy pump operation. When such a condition occurs, it is usually dismissed as cavitation. Then too, many operators never expect anything but noisy operation from rotary pumps. This should not be the case, particularly with screw pumps. With properly designed systems and pumps, quiet, vibration-free operation can be produced and should be expected. Noisy operation is inefficient; steps should be taken to make corrections until the objectionable conditions are overcome. Correct system inlet designs and optimized pump designs with a proper speed selection can go a long way toward overcoming the problem.

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

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

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

Although it is the maximum viscosity and the expected *NPSHA* that are used to determine the size of the pump and to set the speed, it is the minimum viscosity that affects the capacity. Screw pumps must always be selected to give the specified capacity when handling the expected minimum viscosity since this is the point at which the maximum slip, and hence minimum flow rate, occurs (see Figure 17). It should also be noted that the minimum viscosity often determines the selection of the pump model because most manufacturers have special lower-pressure ratings for handling liquids having a viscosity of less than 100 SSU.

Non-Newtonian Liquids The viscosity of most liquids is unaffected by any agitation or shear to which they may be subjected as long as the temperature remains constant. These

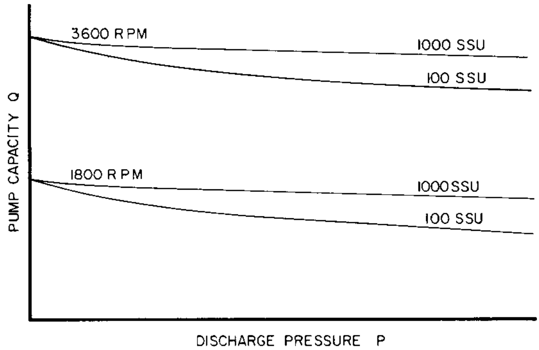

FIGURE 17 Head-capacity (flow rate) performance curve with viscosity a parameter—for two speeds.

liquids are accordingly known as *true* or *Newtonian* liquids, for which the viscosity is constant. Another class of liquids, however, such as cellulose compounds, glues, greases, paints, starches, slurries, and candy compounds, displays changes in viscosity as agitation is varied at constant temperature. The viscosity of these substances depends upon the shear rate at which it is measured, and these fluids are termed *non-Newtonian.*

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

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

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

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

Speed It was previously stated that viscosity and speed are closely tied together and that it is impossible to consider one without the other. Although rotative speed is the ultimate outcome, the basic speed that the manufacturer must consider is the internal axial velocity of the liquid going through the rotors. This is a function of pump type, design, and size.

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

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

The delivered capacity of any specific rotary pump is reduced by

- Decreasing speed
- Decreased viscosity
- Increased differential pressure

The actual speed must always be known. Most often, it differs somewhat from the rated or nameplate specification. This is the first item to be checked and verified in analyzing any pump performance. It is surprising how often the speed is incorrectly assumed and later found to be in error.

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

Pump volumetric efficiency E_v is calculated as

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

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

On the other hand, viscosity changes affect the slip inversely to a certain power, which has been determined empirically. An acceptable approximation for the range of 100 to 10,000 SSU is obtained by using the 0.5 power index. Slip varies approximately with the differential pressure, and a change from 400 SSU to 100 SSU will double the slip in the same way as will a differential pressure change of 100 to 200 lb/in^2 (7 to 14 bar):

$$S = K\sqrt{\frac{P}{\text{viscosity}}}$$

Figure 18 shows the flow rate and volumetric efficiencies as functions of pump size.

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

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

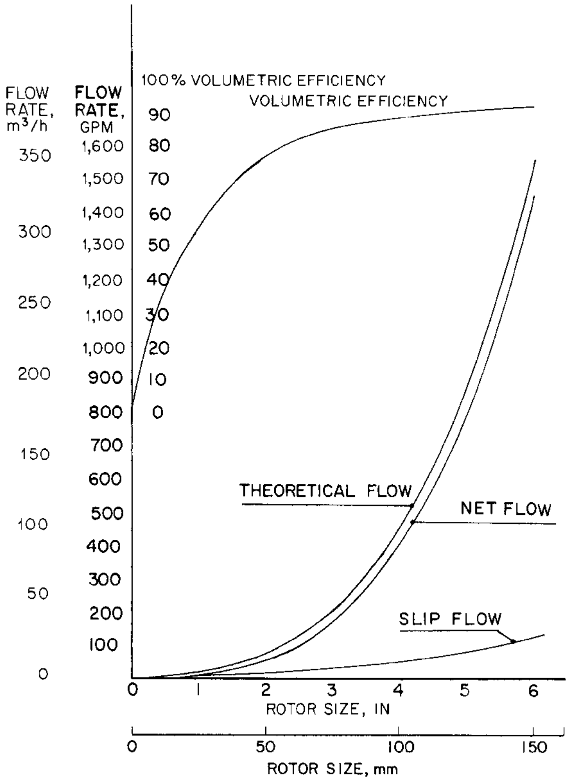

FIGURE 18 Flow rate and volumetric efficiency as functions of pump size.

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

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

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

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

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

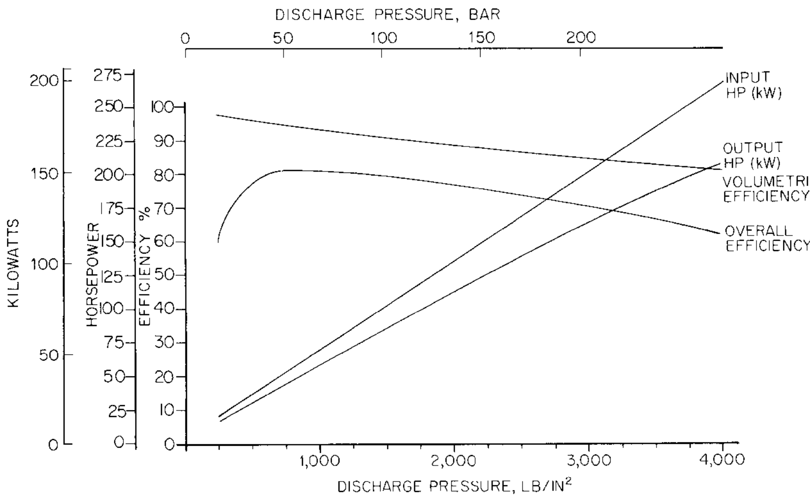

FIGURE 19 Typical overall efficiency curves.

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

The actual pump power output, *whp* (*wkW*), or the delivered liquid horsepower (kilowatts), is the power imparted to the liquid by the pump at the outlet. It is computed similarly to theoretical liquid horsepower (kilowatts) using Q in place of Q_t. Hence, the value will always be less. The pump efficiency E_p is the ratio of the pump power output to the brake horsepower (see Figure 19).

SPECIAL MULTIPHASE APPLICATIONS

Screw pumps have been used with gas-entrained applications for many years, but recent process changes in oil field technologies have created requirements for pumping multiphase fluids, containing more than just nominal amounts of gases. In many oil well applications, the liquid oil flow eventually degenerates into all sorts of difficult multiphase mixtures of oil, gas, water, and sand. In the past, it was common for the gas to be separated and flared off at the well head with only the liquid product to be retained for further processing. If the gas is to be processed as well, separators, compressors, and dual pipelines are required to handle the gas phase. Therefore, a pump, which can handle these difficult liquids with high gas content, can save significant equipment costs as well as operating costs. Under various conditions, the well output can vary from 100 percent liquid to 100 percent gas and all possible combinations. The applications also require the pumping equipment to be able to switch rapidly between the extremes or to handle slugs of liquid or gas, while maintaining the full discharge pressure. The timed two-screw type of pump has proven capable of pumping these multiphase products.[1]

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

When pumping multiphase products with high *gas void fractions* (GVF), the pump must be designed with a small pitch to provide a sufficient number of locks. The key to

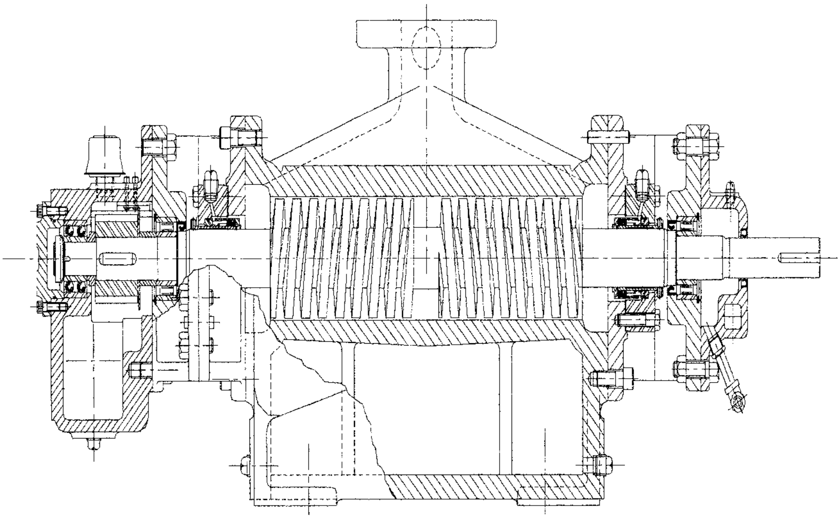

FIGURE 20 Special screw pump for multiphase applications. (Flowserve Corporation)

pumping multiphase products is to ensure that some liquid is always available to seal the screw clearances and reduce the slip. Even a small amount of recirculated liquid is sufficient to provide this seal and enable the screw pump to operate with GVFs approaching 100 percent. Depending on a number of factors, the volume of liquid required to seal and cool the screws can be three to six percent of the total inlet volume flow rate. In order to ensure that sufficient liquid is available at conditions of high GVFs, a separate liquid flush can be provided or a separator type of pump body can be used. This type of body includes a special chamber that can separate some liquid from the multiphase mixture being pumped. This liquid can be recirculated back to the screws and mechanical seals to provide sealing and cooling liquid at times when the product is almost all gas. Figure 20 shows a special screw pump designed for multiphase applications with a separating chamber built into the body.[2]

When pumping liquids, the slip through the internal clearances is proportional to the differential pressure and inversely proportional to the viscosity. However, in multiphase applications, as the GVF increases, the slip decreases until the inlet volume flow rate is equal to the pump displacement. This results in an almost constant inlet volume flow rate, regardless of the differential pressure. This performance can be explained by examining the pressure drop of the multiphase product across the finite clearances. This theoretical analysis confirms that a small amount of liquid in the clearances will effectively seal these clearances and reduce the slip to near zero. Figure 21 shows the typical performance characteristics for a screw pump in a multiphase application when pumping mixtures of air and water at various GVF values.[3]

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

The newest area of application for multiphase pumps is subsea. With special modifications, the rotary screw multiphase pumps can be coupled to submersible motors and mounted on the sea bed, instead of on surface platforms. The idea of pumping multiphase products directly from a subsea well head to shore facilities by means of submersible multiphase pumps has significant potential savings in separating equipment and platforms.[4] Figure 22

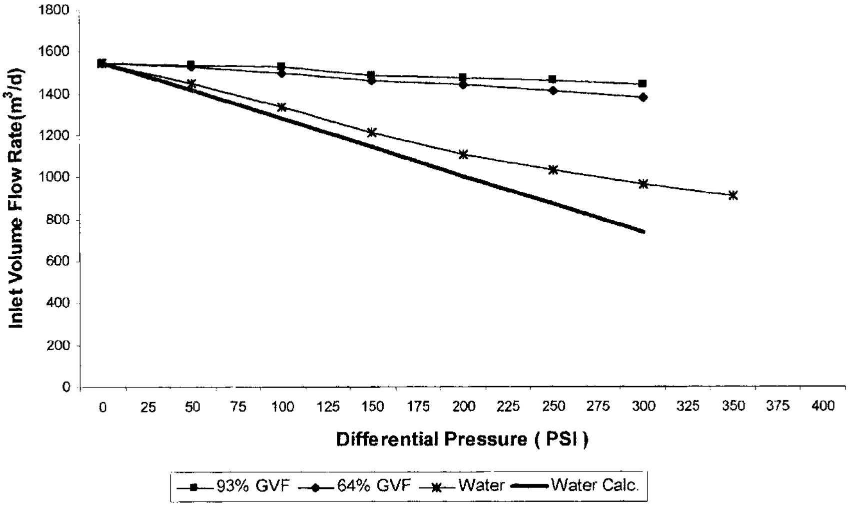

FIGURE 21 Typical performance of a screw pump on multiphase products (bar = psi/14.504).

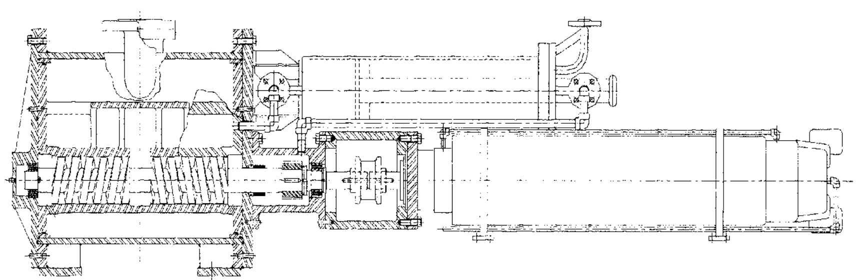

FIGURE 22 Multiphase screw pump for subsea application.

shows a special, timed, two-screw pump, configured for subsea multiphase applications, with special, product-lubricated outboard bearings and pressure-compensated, lube oil chambers.

INSTALLATION AND OPERATION

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

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

Hydraulic Institute Engineering Data Book (Reference 5) along with other similar hydraulic engineering references.

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

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

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

Start-Up A priming connection should be provided on the suction side, and a relief valve should be set from five to ten percent above the maximum working pressure on the discharge side. Under normal operating conditions with completely tight inlet lines and wetted pumping elements, a screw pump is self-priming. Starting the unit may involve simply opening the pump suction and discharge valves and starting the motor. It is always advisable to prime the unit before the initial startup to wet the screws. In new installations, the system may be full of air, which must be removed. If this air is not removed, the performance of the unit will be erratic, and, in certain cases, air in the system can prevent the unit from pumping. Priming the pump should preferably consist of filling not only the pump with fluid but as much of the suction line as possible.

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

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

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

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

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

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

CONCLUDING REMARKS

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

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