SCREW PUMPS OF ONE, TWO AND THREE SCREW DESIGN

by

Joachim Wegener

Department Head, Screw Pump Development

and

Walter E. Cozza

Manager, Projects Department

Radolfzell Works

Allweiler AG

Radolfzell, West Germany

Joachim Wegener is Head of the Screw Pump Design Department of Radolfzell Works, Allweiler AG, in Radolfzell, West Germany. He has been in this position for 20 years.

Mr. Wegener spent three and one-half years in study at the Dresden Technical College. He graduated in mechanical engineering after one year at the Wuppertal Technical College.

He spent one year as a Technical Designer of turbomachinery for Turbo-Manufacturing in Dresden. He spent eight years as a Screw-compressor Designer for Gutehoffnungschutte (GHH) in Sterkrade, West Germany.

Mr. Wegener has many technical publications, including two reports done at the European Pump Symposium at Karlsruhe. He is a member of and serves as chairman of the Technical Designers Workshop of VDI (Society of German Engineers). He is also a member of the Research Society of a German mechanical institution.

Walter E. Cozza has been Head of the Project Department at Allweiler AG in Radolfzell for two and one-half years. He spent six years as USA Application and Sales Engineer for Mahle Pistons in Stuttgart, West Germany. He was at J. Wisemann GmbH in Stuttgart for 14 years as a Production Engineer, Chief of Engineer Design and Assistant Foundry Manager.

After a three and one-half year tool and die making apprenticeship in Germany, Mr. Cozza spent six years studying and working in the USA. He went to vocational school as well as engineering school, and worked for Ace Tool and Manufacturing Company and E. D. Coddington, all in Milwaukee, Wisconsin.

Mr. Cozza is affiliated with VDI (Society of German Engineers).

ABSTRACT

Some fluid solutions utilized throughout the world have proven to be difficult to move, using "conventional" pumps. These included gas-liquid, Newtonian and non-Newtonian fluids. The screw pump, with its inherent slow pressure build-up and low output pulsation, has successfully been used for pumping many of these mixtures. Explanation and discussion of some of the important mechanical features, including wear, noise and performance, are presented. Design and selection criteria for the screw pump are also covered.

OPERATING PRINCIPLES EXPLAINED

The basic principle of all screw pumps is the screw conveyor with its typical axial direction of discharge. Among the advantages of this delivery principle are the relative insensitivity to dirt, insensitivity to viscosity, low-turbulence delivery of the fluid, and largely pulsation-free and thus low-noise delivery. The single-screw pump has gained worldwide importance for the transfer of plastic melts in what is commonly called an extruder.

Sealing Conditions

When fluid media internal friction forces are insufficient for producing any significant pressure, sealing in a pump must be created between the delivery pressure and suction chambers by adding one or more sealing screws. For this reason, the following sealing conditions must be observed:

$$Z_A - n_D\ (Z_D) + n_D = 0$$

where

Z_A = Number of threads of the driving screw
Z_D = Number of threads of each sealing screw
n_D = Number of sealing screws

Application

For material transfer at low pressures, it is predominantly the externally supported twin-screw pump which has gained ground, whereas internally supported twin-screw and three-screw pumps are used for both material and energy transfer.

DESIGN CRITERIA OF THREE-SCREW PUMPS

The structural design of these pumps is best illustrated by an internally supported three-screw pump.

Profiling

Three-screw pumps, with cross-sectional profiles as shown in Figure 1, would typically incorporate the following diameter ratios:

$$d_i{:}d_a{:}D_1{:}D_a = 1{:}3{:}3{:}5$$

This profile is characterized by a favorable proportion of the delivery cross-sectional area to the material cross-sectional area. Of course, sufficient strength and rigidity of the sealing screws must be ensured. The sealing screws, also called working screws, rotate virtually torque-free. The driving screw is not exposed to radial forces, which gives the pump a favorable overall efficiency. The flanks of the working screws are formed by elongated epicycloids and the flanks of the driving screw constitute shortened epicycloids. As can be seen from Figure 1, point "A" of the driving screw constitutes the flank of the working screw, whereas point "B" of the working screw covers the flank of the driving screw. Double-line sealing is achieved by axially extending the driving screw which, in turn, imparts high volumetric efficiency to the pump. The profile is designed so that only the driving screw delivers pressure, whereas the two working screws are driven by the liquid pressure. Except for friction, there is no power transmitted from the driving screws onto the working screws. Thus, wear of the screw flanks is almost non-existent.

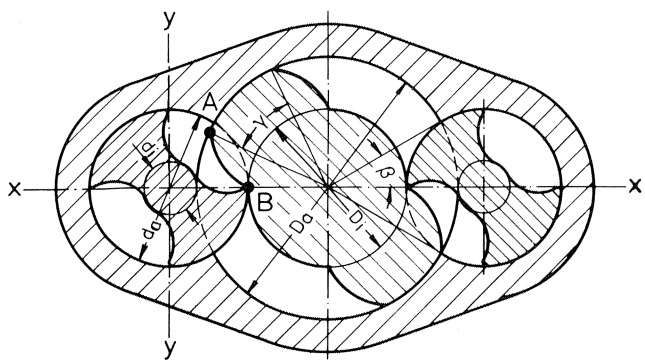

Figure 1. Cross-Sectional Profile of a Three-Screw Pump.

Speed Considerations

The allowable circumferential speed or rpm of a screw pump is governed by both mechanical and hydraulic considerations. Mechanically, the maximum allowable surface speed between the spindle flanks is, in case of the three-screw pump, 12 m/s (38 fps). The maximum permissible mechanical rpm is calculated from this value and tabulated in Table 1.

The maximum permissible rpm from a hydraulic viewpoint is governed by the net positive suction head (NPSH) value which, in relation to the rpm, depends on the viscosity and the axial speed of the fluid. This permissible speed is always lower than the maximum permissible mechanical speed. The maximum permissible mechanical speed can only be utilized if the pump is operating with positive suction pressure. The permissible range of viscosity for three types of screw pumps is shown in Table 2. The friction produced is a function of the viscosity and the rpm. The amount of friction generated can be determined by utilizing the following relation:

$$P_{R2} \approx P_{R1}\sqrt{\nu_2/\nu_1}\,(n_2/n_1)^{1.5\sim2.0} \quad (KW)$$

where

P_R = Friction
ν = Viscosity
n = rotating speed
Subscript 1 = known value
Subscript 2 = new value or value to be calculated.

One can see from this equation that the friction value changes with the square root of the quotient of the viscosity, and with the quotient of the rpm raised to a power between 1.5 and 2.

Table 1. Typical Maximum Speeds of Three-Screw Pumps Without Consideration of Suction Conditions.

Driving Screw Diameter (mm)	Maximum Rotating Speed (rpm)
30	12300
32	11600
40	9260
45	8240
55	6740
60	6180
70	5300
80	4640
90	4120
100	3710
110	3370
120	3100
140	2650
140	2650

Design and Mode of Operation

The screws are supported in an exchangeable liner with a clearance ranging from 0.8 to approximately 2.6 percent of the screw diameter. Modern designs avoid additional bearings or bushings by using the profile section of the screws as a bearing component. Screw deflection is thus virtually eliminated and clearances can be kept correspondingly small. Both are prerequisites for high delivery pressures, which can be realized with these pumps up to approximately 250 bar (3625 psi). As a result of the pressure build-up along the screw axially, the working screws are subjected to balanced radial forces which are transmitted, by way of the ridge surfaces of the screws, to the liner. During this process, certain specific surface pressures, which are a function of viscosity and speed, must not be exceeded. The specific surface pressure is influenced by the profile length. With a pressure difference of up to approximately 100 bar (1450 psi), a profile length of $4(D_a)$ is sufficient. With pressures up to approximately 160 bar (2320 psi), a profile length of $6(D_a)$ should be selected, and over 160 bar, a profile length of $8(D_a)$ is required. The above values relate to hydraulic oil with a viscosity of 37 mm^2/s (centistokes), or 172 SSU.

Axial loads of the screws generated by the influence of profile and pressure on the profile faces are compensated hydraulically. Screw pumps of the type shown in Figure 2 offer hydraulic compensation of the driving screw by incorporating a compensating piston. The compensating piston is designed with an end face which is of equal area as the face of the screw, thus creating offsetting forces (Figure 3). The hydraulically generated axial forces of the driving screw are thus compensating each other. Therefore, the only task of the ball bearing is to axially position the driving screw and absorb small axial residual forces generated by the transmission of friction torque into the working screws and the lead angle, as well as any radial forces which might be caused by coupling imbalance and improper alignment. Hydraulically created axial forces are isolated from the two working screws by the fact that the working screw is machined with journals on the delivery side, which run in bushings and thus are not exposed to the discharge pressure. This is illustrated in Figure 4. Sealing of the driving screw is normally by a mechanical seal; however, sealing by soft packing or radial seal ring is occasionally used. The cartridge with the screw set is mounted in a pump housing having a tubular

Table 2. Viscosity Limits for Screw Pumps.

	Screw Pumps		
	Single-Screw Type	Twin-Screw Type Externally Supported	Three-Screw Type
Permissible Viscosity (SSU)	1-4.62 $\times 10^6$	1-4.62 $\times 10^5$	1-4.62 $\times 10^4$

connection between the suction and the delivery sides. Depending upon the installation, the pump housings are designed for foot, flange or vertical installation.

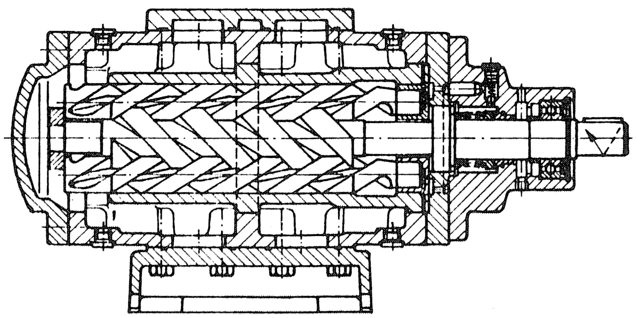

Figure 2. Sectional View of a Three-Screw Pump.

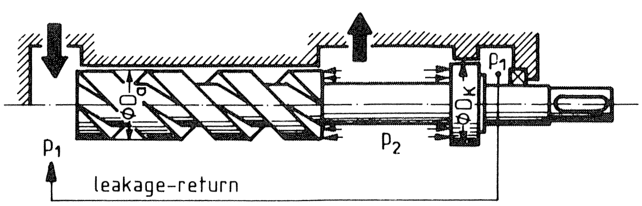

Figure 3. Hydraulic Balancing of Main Screw (Principle).

Materials

At present, nitride-hardenable steels are most frequently used as materials for the screws. The finish-milled screws are nitrided to a hardness of HRC 56 to HRC 60 in gaseous

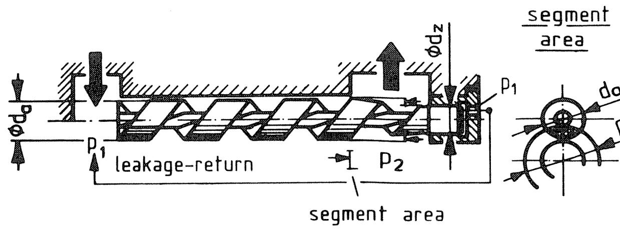

Figure 4. Hydraulic Balancing of Idler Screw (Principle).

ammonia, and thereafter, the outside diameter and flanks are precision-ground. The liner material must meet all the requirements or demands made on typical sliding bearings. These materials must, therefore, incorporate good running properties, sufficient emergency running properties, capability of quick running-in, wear-resistance, pressure-resistance and corrosion-resistance. Grey cast iron of GGL-25 quality covers a wide pressure and viscosity range. The hardness should be between HB 160 and HB 200. This leaves a ferritic matrix which, in the authors' experience, exhibits better running-in behavior and also better emergency running properties than other compositions.

For higher pressure requirements, the liner is made of a proven aluminum friction bearing alloy.

Plastic coating of liners has seen successful application in services with marginal lubricating properties. Using a thickness of 0.2 mm, these coatings have excellent adhesion to base materials such as GGL-25. The coatings also provide desirable sliding and emergency running properties due to the use of Teflon® and graphite filler materials. The principal materials of construction are shown in Table 3.

Table 3. Materials of Construction for Screw Pumps.

Part	Single-Screw Type	Twin-Screw Type (Externally Supported)	Three-Screw Type
Screw	Stainless Steel	Carbon Steel Chrome Steel Stainless Steel	Nitriding Steel
Casing	Cast Iron Stainless Steel	Cast Iron Nodular Cast Iron Bronze Aluminum Bronze	Cast Iron Nodular Cast Iron Steel, welded
Liner/ Stator	Soft Natural Rubber Perbunan Neoprene Hypalon Viton Butyl Rubber Thiokol Vulkollan Polyamide Cast Meehanite	Cast Iron Nodular Cast Iron Niresist 26 Cast Steel Stainless Steel Bronze Aluminum Bronze	Cast Iron Aluminum Alloy Cast Iron, Coated

EXTERNALLY SUPPORTED TWIN-SCREW PUMP

In externally supported twin-screw pumps (Figure 5), the delivery chamber and bearing chamber are separate. This design principle and an appropriate flank clearance within the profiled area of the screw allow non-lubricating fluids to be pumped. Timing gears without any flank clearance simultaneously transmit half of the torque to the secondary screw. Like all screw pumps, the externally supported twin-screw pump is self-priming. Due essentially to the double-flow arrangement of the delivery elements, twin-screw pumps have low NPSH requirements even with high delivery flowrates. The gap required between the screws, as well as between the screws and the pump liner, causes the volumetric efficiency to decrease rapidly with increasing differential pressure. However, the necessary gap provides for so-called "internal ventilation" of the pump so that the pump is capable of transporting gas-liquid mixtures. These characteristics make it the preferred choice for material transfer in the chemical, oil and ship-building industries. Externally supported twin-screw pumps can also be made of rust and acid-resistant materials. Moreover, fluids with temperatures of up to 400°C (720°F) can be handled. Heating of the pump is possible. These pumps are constructed for delivery flowrates of up to approximately 1600 m^3/h (7000 gpm).

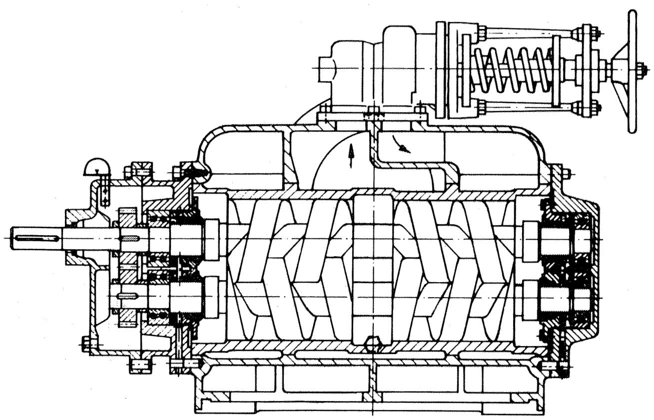

Figure 5. Sectional View of an Externally Supported Twin-Screw Pump.

CHARACTERISTICS OF SCREW PUMPS

Screw pumps have some important characteristics which make them ideally suited for energy transfer applications. These include low pulsation, low noise, and slow pressure build-up. Theoretically, screw pumps are pulsation-free since they deliver the same volumetric flow for any angle of rotation. In practice, however, slight pulsations ranging in magnitude from 1 to 2.5 percent of the discharge pressure will be produced. Two occurrences are responsible for this: the changing quality or effectiveness of sealing between the delivery and suction chambers as the screws rotate, and the compressibility of the fluid pumped. Fluids, as is known, can be compressed by an amount

$$\Delta V = V\beta P$$

where the compressibility factor, β for air-free hydraulic oil is $6.7(10^{-5})$ cm^2/kp, for water $4.5(10^{-5})$ cm^2/kp, and for synthetic, not easily inflammable, fluids such as phosphate ester, $3.7(10^{-5})$ cm^2/kp. If, as is common with any pumping process, the volume at lower pressure enclosed in the delivery elements is now connected with the volume at higher pressure which is in the pipeline, pressure compensation and pressure pulsation will result. From the compressibility factors given above, it can be seen that hydraulic fluids with lower compressibility factors, such as phosphate ester, are less able to reduce the shock pressures caused by the switching processes than fluids with higher compressibility factors. Therefore, some plants operate rougher than others.

Pressure build-up time must be understood as the time it takes to bring an individual liquid particle from suction pressure to operating pressure. In the case of screw pumps, due to the length of the screws, there are always several chambers between the delivery and suction sides. Therefore, the pressure is gradually built up from suction pressure to operating pressure so that there will never be a sudden pressure change. As a result, there are small pressure pulsations and low sound pressure levels. An additional advantage of this slow pressure build-up is the protection of the fluid pumped, since a rapid pressure build-up, as has been established, contributes to the rapid aging of the fluid being pumped. Pressure pulsations incite not only the surrounding plant, but also the oil column fixed between the pump and throttling elements. This column can vibrate and thus generate sound radiation. Should this be the case, a plant can contribute materially to noise pollution. Here, the plant designer must verify that not only the plant components, but also the fixed oil column, exhibit vibration frequencies far enough from the pulsation frequency so as to avoid resonant conditions. The pulsation frequency of screw pumps is calculated as:

$$f = nz/60$$

where n represents the rpm of the driving screw and z the number of leads or threads of the driving screw. The inherent frequency of the oil column is calculated from

$$f = \frac{1}{2\ell}\left[\frac{98.1}{\rho\beta}\right]^{1/2} \quad (S^{-1})$$

where

ℓ = length of the fixed oil column, in meters

ρ = density, in kg/dm^3, and

β = compressibility factor, cm^2/kp

The sound pressure level of a three-screw type of pump is remarkably low. During testing, a medium size screw pump exhibited a sound level of only 57 dB(A) at a speed of 2900 rpm and a delivery pressure of 100 bar (1450 psi).

In addition to its special suitability for energy transfer, the three-screw pump is also well-suited for the various material transfer tasks in which higher pressures may be required for a number of reasons. Thus, screw pumps are used almost exclusively as transfer and burner pumps for light and heavy fuel oils, provided they do not have to handle extremely small throughputs. Due to their axial flow design, clogging susceptibility is minimal and wear is extremely low. Viscosities from 1 cSt to 10,000 cSt are easily accommodated. With higher viscosities, the clearances may have to be increased so as to reduce friction.

A new and important application has opened up in the machine tool industry, in the field of pumping oil in water emulsion. As the pressures of the cooling fluids had to be increased in grinding, cutting and deephole-boring equipment, these services became the domain of three-screw pumps with internal bearings. Cutting fluid consists of up to 97 percent water, the viscosity is one centistoke, and pressures range up to 80 bar (1100 psi). The delivery flowrates of these pumps reach approximately 300 m^3/h (1300 gpm), and suction capabilities to approximately 8 mWC (23.2 in. Hg). The release of air from the fluid to be pumped prevents any increase of the suction lift which could otherwise, theoretically, reach 10 meters. Mineral oils contain approximately 9 percent by volume of air. Initially dissolved, the air later emerges from the solution as the pressure drops and air bubbles are formed.

Any description of screw pumps would be incomplete if additional advantageous properties were not mentioned as well. These include the reversibility of the sense of rotation not only of the one-screw, but also the twin- and three-screw types. For example, it is possible, upon completion of any delivery task, to pump out the delivery line by reversing rotation. Also, the low flywheel effect, due to the small screw diameters, is worth mentioning, along with almost perfect balance, i.e., low vibration. Shorter startup and fewer down-times can result. It is exactly these characteristics which are the prerequisites for application as a hydraulic motor, as thus the full nominal torque is immediately available at startup, and a quick reversal of rotation is possible. If one wanted to make use of the advantages offered by hydraulic drives without having to put up with such disadvantages as the tendency to be noisy, the three-screw pump would serve ideally as a hydraulic motor.

The torque of the pumps must not be confused with the flywheel effect. With positive displacement pumps, starting torque increases linearly with speed. Only the initial break-away torque deserves to be observed and must be especially considered with single-screw pumps, since this torque may reach high values with this pump type. Appropriate data can be provided by qualified manufacturers on the basis of their experience with regard to the kind of fluid to be pumped and the stator material selected for the application. An overview of common characteristics found in screw pumps can be found in Table 4.

Table 4. Common Characteristics of Single-, Twin-, and Three-Screw Pumps.

Pump type	Preferred application	Common properties
Single-screw	Solid-containing fluids	Self-priming Good suction capacity
Twin-screw, externally supported	Non-lubricating fluids Air- or gas-containing fluids	Insensitive to viscosity Little pulsation Near-silent
Three-screw	High pressures at a low sound level	Safe to operate Low sensitivity to dirt

SELECTION OF SINGLE-, TWIN- OR THREE-SCREW PUMPS

Three criteria should be considered in determining whether a single-, twin- or three-screw pump should be used for handling a given delivery task. The particular characteristics of the pump, the properties of the pumpage, and, of course, the required operating conditions must be considered.

An overview of the preferred fields of application for single-, twin- and three-screw pumps with regard to the three criteria mentioned is presented in Table 5. The general application limits of single-, twin- and three-screw pumps are illustrated in Table 6. With a view to avoiding misinterpretation, it must be emphasized once again that not only the single-, but also the twin- and three-screw pumps are appropriate for handling many liquid and non-liquid fluids. Hence, the decision as to whether a single-, twin-, or three-screw pump should be used must take supplementary data into account. Typical of these supplementary data would be operating cost calculations for a minimum period of two years, inclusive of necessary spare parts and repair costs. However, a survey of existing experience will also prove helpful. Clearly, the investment alone is not a sufficient criterion for decision making. Recall, again, that only twin-screw internally supported and three-screw pump types are serious contenders for energy transfer and hydraulic applications.

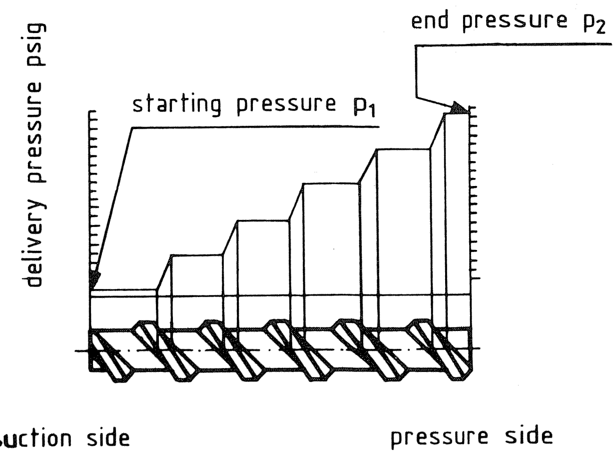

Figure 6. Delivery Capacity as a Function of Suction Lift.

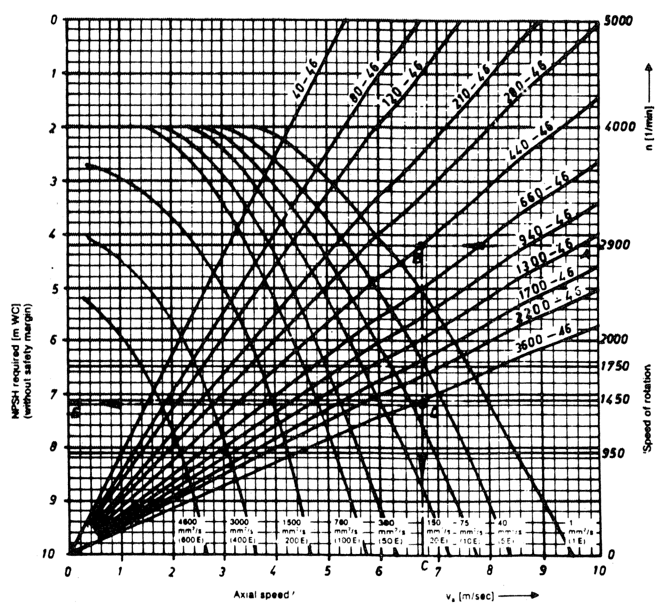

Figure 7. NPSH Curves for Various Screw Pumps.

Table 5. Preferred Fields of Application for Single-, Twin-, or Three-Screw Material Transfer Pumps.

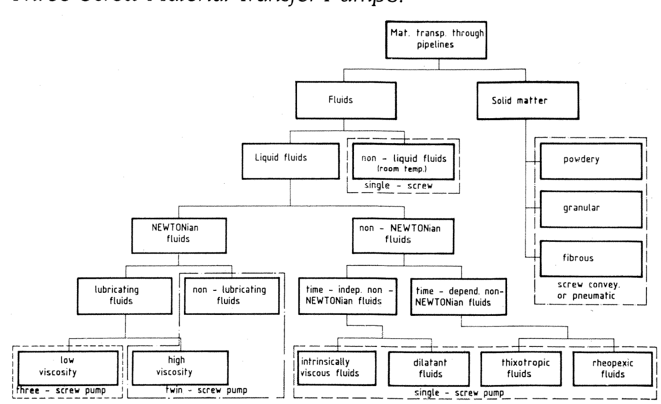

Table 6. General Application Limits of Screw Pumps.

Operating conditions		Screw pumps		
		single-screw type	twin-screw type externally supported	three-screw type
Delivery	(gpm)	0 to 880	2,2 to 4840	0,9 to 1320
Delivery pressure	(psi)	87 (348)	232 (435)	3625
Suction head	in Hg	24,6	24,6	23,2
Max. temperature	°F	194 (302)	752	572
Admissible viscosity	(SSU)	1 to $4{,}62 \cdot 10^6$	1 to $4{,}62 \cdot 10^5$	1 to $4{,}62 \cdot 10^4$
Non-lubricating media		yes	yes	no
Solid-matter contents in fluid to be pumped		yes	no	no
Corrosive media		yes	yes	no

Values in brackets for special designs .

NPSH Considerations

The criterion for determining the $NPSH_R$ value for screw pumps is the actual start of hydraulic noise at a given suction lift. At the time this noise begins, there will be a loss of flow of approximately two percent. The flow reduction results from air changing from its dissolved (molecular) state to an undissolved state (air bubbles). Oil actually contains nine percent by volume of dissolved air. At the pressure side of the pump the air bubbles are again compressed. This change in volume will change the pressure and cause hydraulic noise to be generated.

For a pictorial representation of the delivery capacity as a function of pump suction lift refer to Figure 6. Finally, a series of NPSH curves, which refer to a liquid without any air enclosed, are introduced and the beginning of bubble formation is shown in Figure 7. These curves must be used in conjunction with a "margin-of-safety add-on value," which is generally supplied by the screw pump manufacturer and may vary with pump model and medium pumped. This example illustrates the selection process for a pump of given capacity, say, "Type 440-46," operating at a speed of 2900 rpm. The pumpage has a viscosity of 55 mm^2/sec, and it is desired to determine the $NPSH_R$, in meters of water. To find this value, enter the graph at the given speed, 2900 rpm, and proceed horizontally to the intersection with the size/model line 440-46. This is point "B." From "B," drop a line to point "C," which allows the determination of the axial speed of the screw or screws incorporated in this pump model. Interpolation between viscosity lines locates point "D" on line "B-C." From point "D," proceed horizontally to point "E," which establishes the requires NPSH as 7.2 meters of water.

SUMMARY

Screw pumps have successfully been used in many adverse services, including pumping gas-liquid, Newtonian and non-Newtonian mixtures. With either internally or externally supported screw designs, these pumps exhibit a favorable overall efficiency, low wear on working components, low pulsation (low noise) and slow pressure build-up performance. Pumpage rates over 70,000 gpm and discharge pressures in excess of 3000 psi are obtainable with suction heads of about 25 in Hg. Selection criteria have been presented and show the versatility of the screw pumps, the choice for many chemical and oil transfer systems.