Kirk-Othmer Encyclopedia of Chemical Technology.

PUMPS

Pumps are used in a wide range of industrial and residential applications. Pumping equipment is extremely diverse, varying in type, size, and materials of construction. There have been significant developments in the area of pumping equipment since the early 1980s. There are now materials for corrosive applications (1, 2); modern sealing techniques (3, 4); improved dry-running capabilities (5, 6, 64) of sealless pumps, which are magnetically driven; and applications of magnetic bearings in multistage high energy pumps (7, 8). The passage of the Clean Air Act of 1990 by the U.S. Congress, a heightened attention to a safe workplace environment, and users' demand for better equipment reliability have all led to improved mean time between failures (MTBF) and scheduled maintenance (MTBSM). Also, the new higher-efficiency motor designs have contributed to the selection and application of the most efficient and effective equipment, from both pump and a driver standpoint.

1. Classification

One general source of pump terminology, definitions, rules, and standards is the Hydraulic Institute (HI) Standards (9), approved by the American National Standards Institute (ANSI) as national standards. A classification of pumps by type, as defined by the HI, is shown in Figure 1.

Pumps are divided into two fundamental types based on the manner in which pumps transmit energy to the pumped media: kinetic (of which centrifugal pumps are the most popular) or positive displacement. In the first type, a centrifugal force of the rotating element, called an impeller, impels kinetic energy to the fluid, moving the fluid from pump suction to the discharge. The second type uses the reciprocating action of one or several pistons, or a squeezing action of meshing gears, lobes, or other moving bodies, to displace the media from one area into another, ie, moving the material from suction to discharge. Sometimes the terms inlet, for suction, and exit or outlet, for discharge, are used. The pumped medium is usually liquid. However, many designs can handle solids in suspension, entrained or dissolved gas, paper pulp, mud, slurries, tars, and other exotic substances, which, at least by appearance, do not resemble a liquid. Nevertheless, an overall liquid behavior must be exhibited by the medium in order to be pumped. In other words, the medium must have negligible resistance to tensile stresses.

The Hydraulic Institute classifies pumps by type, not by application. The user, however, must ultimately deal with specific applications (65). Often, based on personal experience, preference for a particular type of pump develops. This preference is passed on in the particular industry. For example, boiler feed pumps are usually of a multistage diffusor barrel type, especially for the medium and high energy [over 750 kW (1000 hp)] applications, although volute pumps in single- or multistage configurations, having radially or axially split casings, have also been applied successfully. Examples of pump types and applications and the reasons behind applicational preferences follow.

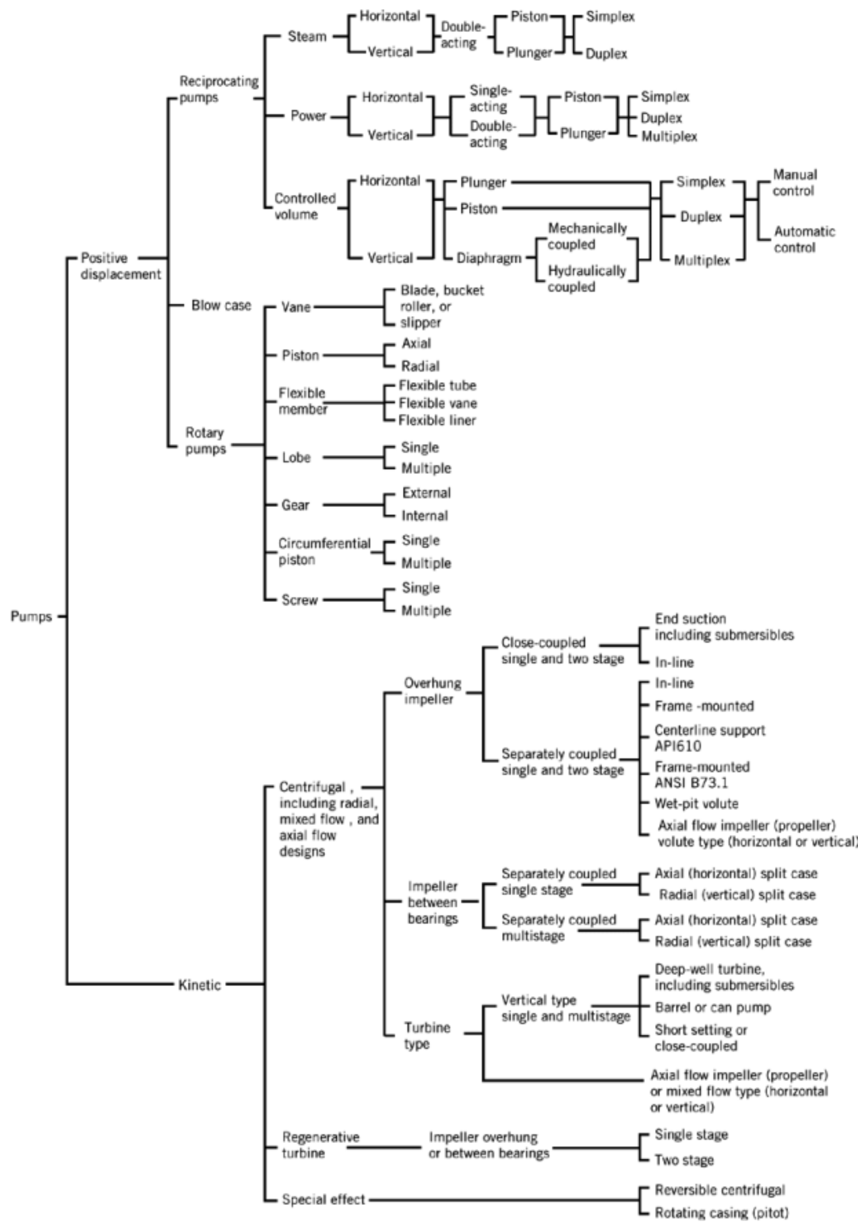

Fig. 1. Types of pumps.(Courtesy of the Hydraulic Institute.)

2. Operating Conditions

Before a pump selection can be made, the duty conditions must be specified. These include type of fluid, density or specific gravity, temperature, viscosity, flow, inlet and outlet pressures, possibility of dry-running, fixed or variable operating conditions, presence of solids, corrosive/erosive material in the liquid. For a typical process installation, an accurate estimate of the pumping system is required, starting with the source of the liquid

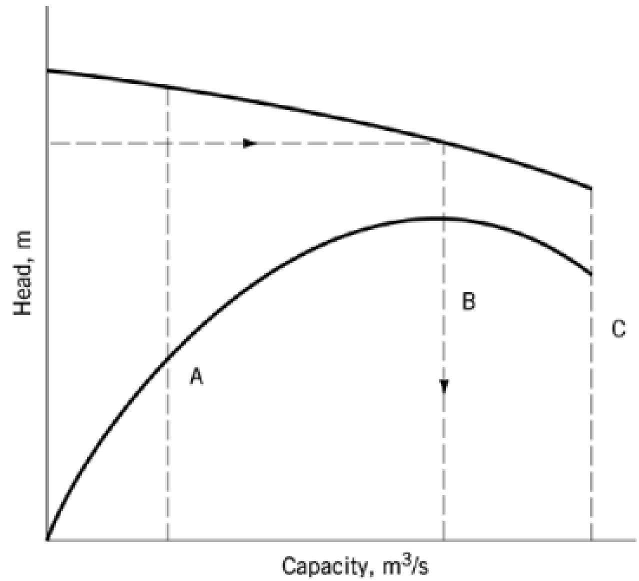

Fig. 2. Head–capacity curve, where A represents the minimum allowable capacity; B, the best efficiency point (BEP); and C, the maximum allowable capacity.

(tank, vessel, pipeline, basin, etc) through the planned system layout to the terminal point (see Pipelines; Piping systems). Piping sizes must be determined, based on suitable flowing velocities for the fluid (66) or for the fluid mixture in the case of slurries. Pressure drops through the entire piping system must be estimated, including all valves, fittings, process equipment such as heat exchangers, heaters, and boilers, and any losses through orifices or control valves. Difference in pressures between the suction and discharge location, together with changes in elevation between these two, completes the various elements of the system and makes it possible to estimate or design a complete pumping system.

Most process plant engineers utilize some form of preprinted pump calculation worksheet or a computer to aid in the collection of information. This helps to ensure that, for a given system, all possible pumping situations, alternative routes, alternative pressures or temperatures, varying flowing rates, varying fluid properties, etc, are considered. Good examples of pumping systems evaluation are available (10). The most severe or limiting case is then chosen as the pump-rated condition. At the same time, however, care must be taken not to demand the pump to operate at such a variety of conditions that it is forced to perform either below the minimum allowable flow or too far out in the flow. For best reliability, extra pumps (spares) should be included whenever needed to limit the operation of each pump to within the allowable flow limits.

2.1. Capacity

Pumps deliver a certain capacity, Q, sometimes referred to as flow, which can be measured directly by venturi, orifice plate (11), or magnetic meters (12) (see Flow measurement). The indirect way to determine capacity is often used. Whereas this method is less accurate than applying a flow meter, it often is the only method available in the field: The total head is measured and the capacity found from the pump head–capacity (H–Q) curve (Fig. 2). More recently, sonic flow meters (13) have been used, which can be installed on the piping without the need for pipe disassembly. These meters are simple to use, but require relatively clean single-phase liquid for reliable measurements.

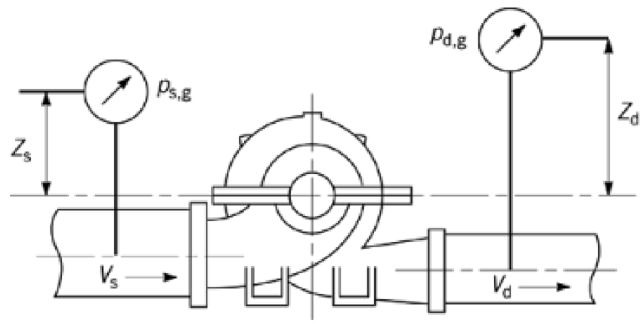

Fig. 3. Definition of the variables used to determine pump head where (_ _ _) represents the common reference plane for measurement. See text (eqs. 1–3).

2.2. Head

The true meaning of the total developed pump head, H, is the amount of energy received by the unit of mass per unit of time (14). This concept is traceable to compressors and fans, where engineers operate with enthalpy, a close relation to the concept of total energy. However, because of the almost incompressible nature of liquids, a simplification is possible to reduce enthalpy to a simpler form, a Bernoulli equation, as shown in equations 1–3, where g is the gravitational constant, SG is specific gravity, γ is the density equivalent, H_s is suction head, H_d is discharge head, and H is the pump head, ie, the difference between H_d and H_s.

$$H_s = \frac{p_{s,g}}{\gamma} + \frac{V_s^2}{2g} + Z_s = \frac{p_{s,g} \times 2.31}{\mathrm{SG}} + \frac{V_s^2}{2g} + Z_s \qquad (1)$$

$$H_d = \frac{p_{d,g}}{\gamma} + \frac{V_d^2}{2g} + Z_d = \frac{p_{d,g} \times 2.31}{\mathrm{SG}} + \frac{V_d^2}{2g} + Z_d \qquad (2)$$

$$H = H_d - H_s = \frac{p_{d,g} - p_{s,g}}{\gamma} + \frac{V_d^2}{2g} - \frac{V_s^2}{2g} + (Z_d - Z_s) \qquad (3)$$

As can be seen from these equations and Figure 3, the head ingredients are static (pressure, p), dynamic (velocity, V), and elevation, Z, components. A good explanation of this subject is available (15).

2.3. Power

There are two main ways to measure the power delivered by the driver to the pump. The first method is to install a torque meter between the pump and the driver. A torque meter is a rotating bar having a strain gauge to measure shear deformation of a torqued shaft. Discussion of the principle of torque meter operation is available (16). The benefit of this method is direct and accurate measurements. The power delivered to the pump from the driver is calculated from torque, T, and speed (rpm) in units of brake horsepower, ie, BHP (eq. 4a) when T is in $\mathrm{lbs{\cdot}ft}$, and kW (eq. 4b) when T is $\mathrm{N{\cdot}m}$.

$$\mathrm{BHP} = \frac{T \times \mathrm{rpm}}{5252} \qquad (4a)$$

$$\mathrm{kW} = \frac{T \times \mathrm{rpm}}{9545} \qquad (4b)$$

The disadvantages of this method are the need for a torque meter, longer total length of the pumping unit, and greater susceptibility to misalignment and vibrations. This method is used only at a manufacturer's test facilities or research laboratories. It is not used in the field.

In the second method, the pump and the motor are coupled directly, and either power (in kilowatts) or the current, *I,* and voltage, *U,* are measured at the motor terminals. To determine the power actually transmitted into a pump, the motor power factor (PF) and efficiency ($\mathrm{Eff}_{\mathrm{motor}}$) must be known. These values are usually taken from the motor manufacturer's calibration curves (17).

For a three-phase electric motor, the horsepower is calculated as

$$\mathrm{kW} = \frac{I \times U \times (3)^{1/2} \times \mathrm{PF} \times \mathrm{Eff}_{\mathrm{motor}}}{1000} \qquad (5)$$

$$\mathrm{BHP} = \frac{\mathrm{kW}}{0.746} \qquad (6)$$

If the pump manufacturer uses motors for pump power measurement, these motors are calibrated to determine the horsepower from the electric power (or current and voltage) reading and calibration curves. Such test motors are recalibrated periodically, ensuring the same degree of accuracy as shown by the torque meters.

2.4. Efficiency

A portion of the power delivered by the driver to the pump is spent to overcome hydraulic losses, ie, fluid friction, separation, and mixing; volumetric losses, ie, leakage across wear rings of centrifugal pumps (or a "slip" through clearances of rotary pumps); and mechanical losses, ie, bearings friction, internal rubbing, and mechanical seal or packing friction losses. The disk friction (18) of centrifugal impeller (or gears, for the gear-type pumps, etc) rotating inside the casing is hydraulic in nature, but is customarily grouped together with mechanical power losses because these latter losses are caused by the fluid trapped between the rotating shrouds and casing walls. The difference between the driver power and the pump losses equals the power delivered to the fluid, ie, the pumped medium.

$$\mathrm{kW}_{\mathrm{fluid}} = \mathrm{kW} - \mathrm{kW}_{\mathrm{losses}}\ (\text{metric}) \qquad (7a)$$

$$\mathrm{HP}_{\mathrm{fluid}} = \mathrm{BHP} - \mathrm{HP}_{\mathrm{losses}}\ (\text{U.S.}) \qquad (7b)$$

The pump overall efficiency is then defined as

$$\mathrm{Eff} = \frac{\mathrm{kW}_{\mathrm{fluid}}}{\mathrm{kW}} = \frac{Q \times H \times \mathrm{SG}}{\mathrm{kW} \times 0.102} \qquad (8a)$$

when Q is in m^3/s and H in m; and

$$\mathrm{Eff} = \frac{\mathrm{HP}_{\mathrm{fluid}}}{\mathrm{BHP}} = \frac{Q \times H \times \mathrm{SG}}{\mathrm{BHP} \times 3960} (\text{for centritugal pumps}) \qquad (8b)$$

$$\mathrm{Eff} = \frac{\mathrm{GPM} \times \mathrm{PSI}}{\mathrm{BHP} \times 3960}\ (\text{for positive displacement pumps}) \qquad (8c)$$

if Q is in gal/min and H in ft. More details on overall pump efficiency, as well as the various components, ie, hydraulic, volumetric, and mechanical, can be found in the literature (14).

2.5. Specific Speed

A review of the dimensionless analysis as related to pumps can be found in Reference 14. One of these nondimensional quantities is called the specific speed. This historical term is actually misleading, since although *specific speed* has a pump rotational speed component within its definition, it has, however, very little to do with idea of mechanical rotation, or speed, as its name seems to imply. The universal dimensionless specific speed, Ω_s, is defined as in equation 9:

$$\Omega_s = \frac{\Omega\,(Q)^{0.5}}{(gH)^{0.75}} \qquad (9)$$

Ω, Q, g, and H may be in any consistent set of units. In equation 9, Ω, the rotational speed, is in rad/s; Q, in m^3/s; H, in m; and g, the gravitational acceleration, is 9.81 m/s^2. The specific speed is calculated at the best efficiency point for the maximum impeller diameter. In the United States, it is customary to drop the gravitation constant and represent specific speed, NS, in the following form:

$$NS = \frac{N\,(Q)^{0.5}}{H^{0.75}} \qquad (10)$$

where N is the rotational speed in rpm, Q is in gal/min, and H in ft.

In Europe and elsewhere, specific speed is calculated as

$$nS = \frac{N\,(Q)^{0.5}}{H^{0.75}} \qquad (11)$$

where n, the rotational speed, is in rpm; Q and H are in m^3/s and m, respectively. The various types of specific speed are all interconvertible:

$$NS = \Omega_s \times 2733 \quad nS = \Omega_s \times 53.0 \quad NS = nS \times 51.6 \qquad (12)$$

For double suction pumps, using the HI convention, Q is taken as the total pump flow, although some publications use half-flow, ie, flow per impeller eye. A double-suction impeller, even having identical profiles (of each of its halves) with a similar single-suction impeller, would thus have an artificially high value ($\sqrt{2}$ higher) of specific speed, since a total pump flow is used in calculations. If half of the flow is used in calculations of a specific speed of a double-suction pump, the result would be more consistent in comparing the profile shapes of the impellers. Therefore, it is important to know whether full, or half, flow was used in calculations of a specific speed of a double-suction pump, in a given publication, or a reference. For multistage pumps, the developed head must be taken per stage for the NS calculation. By definition (eq. 9), high head (per stage), low flow pumps have low specific speed; low head, high flow pumps, such as turbine and propeller pumps, have high specific speed.

Centrifugal pump efficiency is related to specific speed, flow, and surface roughness (19). Numerous attempts have been made to relate efficiency of any pump to these parameters in a single formula or chart. In practice, however, too many variations exist in design, casting quality, and other factors to allow for a single correlation. Nevertheless, approximate charts have been developed for average achieved efficiencies of commercial pumps, as shown in Figure 4 (20). Although this chart was originally developed using single-stage pumps, it is a useful tool for making quick efficiency estimates and comparisons, even for a variety of centrifugal pump types.

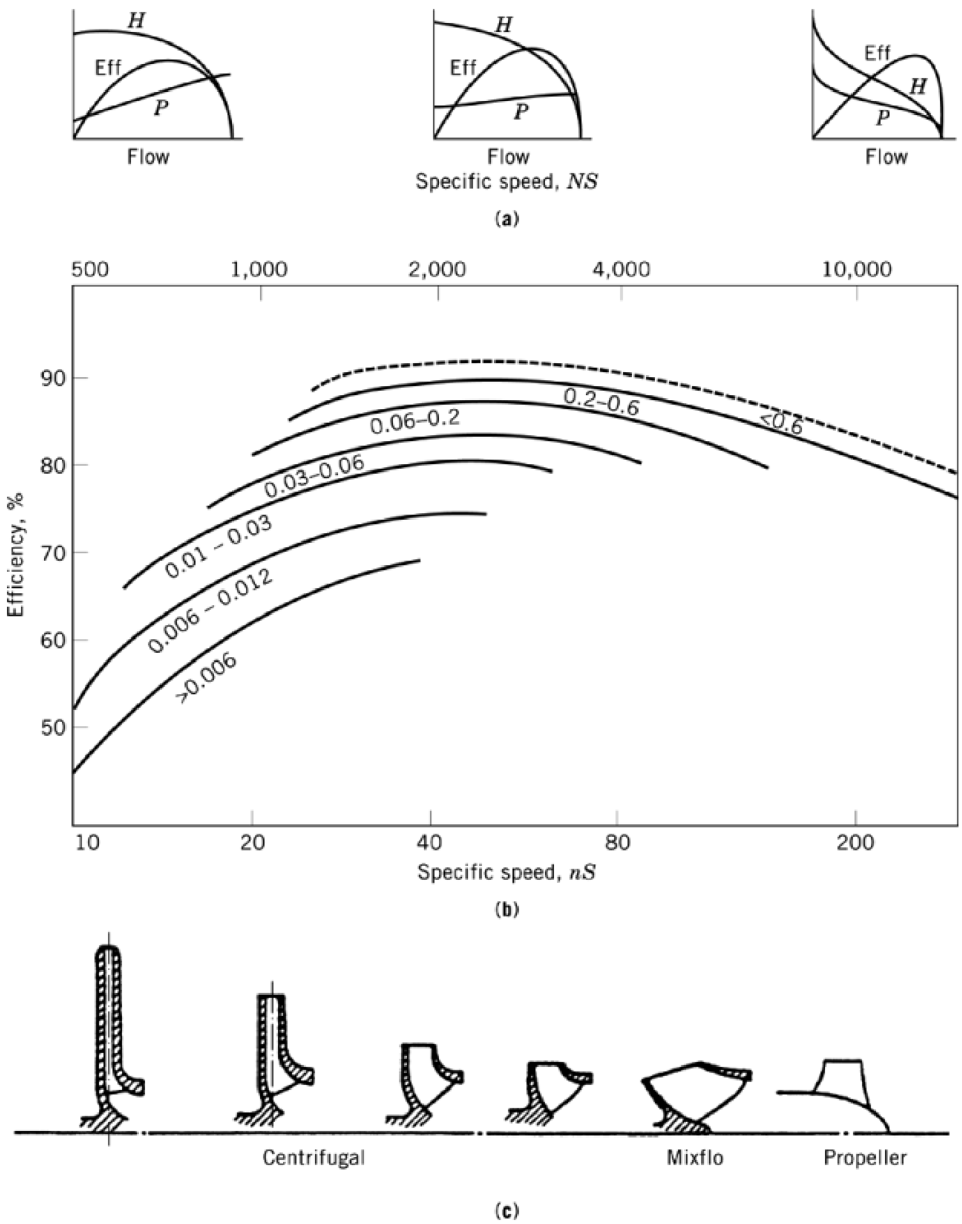

Fig. 4. Chart for efficiency estimates and curve shapes, where (**a**) represents curve shapes showing the relationship between efficiency (Eff), head (H), and power (P) as a function of flow; (**b**) specific speed, where the numbers represent flow in m^3/s; and (**c**) shape of impeller profiles, is determined by the specific speed.

2.6. Suction and Suction Specific Speed

Just as head, capacity, power, and efficiency describe a discharge performance of a centrifugal pump, a net positive suction head (NPSH) characterizes the suction performance (14). In order to be pumped, the fluid must be in liquid state. No or very little vaporization should occur. If the liquid pressure drops below a substance's vapor pressure in a pump, vaporization starts and eventually either the pump ceases to operate, or its operation is accompanied by loss of head, excessive noise, and vibrations (21). The vaporization of the liquid should not be confused with the presence of entrained or dissolved gas, often air, that may come out of the solution at lowered pressure. This gas can also cause the loss of performance, although no cavitation damage takes place (22, 23). Vaporization of liquid usually happens at the pump suction, and more often at the impeller entrance near the leading edges of the impeller blades, where high local velocities owing to curvature effects cause pressure drop so that vapor bubbles begin to form. As these bubbles move downstream into a higher-pressure region, they collapse, exerting high-pressure impacts within a microscopic area on the impeller material. Such hammering by collapsing bubbles against the pump internal surfaces occurs at frequencies ranging from 600 Hz to 25 kHz, and causes cavities, ie, damaging loss of material (24).

It is difficult to determine exactly the areas of localized pressure reductions inside the pump, although much research has been focused on this field. It is easy, however, to measure the total fluid pressure (static plus dynamic) at some convenient point, such as pump inlet flange, and adjust it in reference to the pump centerline location (as was shown on Fig. 3). By testing, it is possible to determine the point when the pump loses performance appreciably, such as 3% head drop, and to define the NPSH at that point, which is referred to as a required NPSH (NPSHR). The available NPSH (NPSHA) indicates how much suction head over vapor pressure head is available, referenced to an accepted datum, as defined by the Hydraulic Institute:

$$\text{NPSHA} = H_{\text{static}} + H_{\text{dynamic}} + Z_{\text{s}} - H_{\text{vapor}} = H_{\text{s}} + \frac{V_{\text{s}}^2}{2\,g} + Z_{\text{s}} - H_{\text{vapor}} \qquad (13)$$

where Z_{s} is the elevation of gauge centerline above or below the centerline of the impeller (see Fig. 3).

Such a definition of NPSHA (eq. 13) is convenient. In many applications, the pumpage is taken from the vessel, where static pressure is known, and the dynamic (velocity) head is zero. If the vessel is open to the atmosphere, then the static pressure is simply equal to atmospheric pressure plus elevation (Note: a common practical mistake is forgetting the absolute additive portion of the static pressure, ie, using liquid level elevation only). Hydraulic losses between the vessel surface and pump inlet must be calculated, in order to determine total head at the pump suction (near the flange), and adjusted to the datum (centerline) reference plane. NPSHA must always be greater than NPSHR, and a safety margin should be added. These margins are always important, but sometimes can be reduced for saturated hydrocarbons. Generally, signs of cavitation appear on the suction side (visible side, looking at impeller eye) of the impeller vanes. However, sometimes cavitation appears on the pressure side, in areas of casing inlet, especially for double-suction pumps, and even at the volute near cutwaters and diffusor tips (25). The vane–suction-side cavitation is usually attributed to positive incidence angles (blade inlet angle being higher than the incoming flow angle) during operation at low flow. The damage on the section side of vanes is the result of the impeller eye being too big or the pump operating at flow that is too low for the given impeller eye, which makes the impeller sensitive to backflow at the shroud (26). Backflow causes redistribution of the flow field at the inlet of the impeller in such a way that the effective flow angles near root streamlines (especially closer to the shaft) become too high, switching flow separation from the suction to pressure side and causing vortexing and damage.

Similar to the concept of the specific speed, a suction specific speed, S, is defined as

$$S = \frac{N(Q)^{0.5}}{\text{NPSHR}^{0.75}} \qquad (14)$$

A U.S. definition for S is accepted worldwide. For conversion to metric,

$$nSS = \frac{S}{51.6} \qquad (15)$$

Again, according to HI convention, for double-suction pumps, Q is half of the total pump flow, ie, taken per impeller eye. The value of S is calculated at the best efficiency point (BEP) at maximum impeller diameter.

Typically, suction specific speed of centrifugal pumps is within the 6,000–14,000 range (U.S. definition). Special inducer designs (27), installed at the inlet, can raise the suction specific speed to 20,000 or even higher to satisfy the requirements of low suction pressures. In theory, pumps designed for any suction specific speed run fully satisfactorily if operated close to the BEP (see Fig. 2). Unfortunately, the higher the suction specific speed, the less tolerant the pumps are to off-peak operation, ie, lower than BEP capacities (28). In practice, pumps seldom operate at the BEP, but rather at a wide range of capacities, depending on the demand. This practical consideration has prompted pump users to specify the upper limit of S not to be exceeded. This limit is typically between 9,500 and 11,000, and the Hydraulic Institute even recommended 8500. Special impeller designs can desensitize the pump to flow separation. This requires a comprehensive flow analysis, which involves computation flow dynamics (CFD) and research laboratory testing.

2.7. Affinity Laws

Centrifugal pump performance is affected by the rotating speed. When speed increases, the flow increases linearly, and the head increases as a square of the speed (14).

$$\frac{Q_2}{Q_1} = \frac{N_2}{N_1} \quad \frac{H_2}{H_1} = \left(\frac{N_2}{N_1}\right)^2 \quad \frac{\mathrm{BHP}_2}{\mathrm{BHP}_1} = \left(\frac{N_2}{N_1}\right)^3 \qquad (16)$$

A similar relationship guides the change in performance when the impeller outside diameter (OD) is cut:

$$\frac{Q_2}{Q_1} = \frac{\mathrm{OD}_2}{\mathrm{OD}_1} \quad \frac{H_2}{H_1} = \left(\frac{\mathrm{OD}_2}{\mathrm{OD}_1}\right)^2 \quad \frac{\mathrm{BHP}_2}{\mathrm{BHP}_1} = \left(\frac{\mathrm{OD}_2}{\mathrm{OD}_1}\right)^3 \qquad (17)$$

In the case of speed changes, the pump efficiency is not affected except for a minor change owing to Reynolds number change, but the diameter cut may reduce the efficiency appreciably on account of increased gap and losses between the impeller OD and a collector (casing or diffusor). Affinity laws do not apply to positive-displacement pumps, but only to centrifugal types.

3. Types of Pumps

3.1. Kinetic

Kinetic pumps, which act by impelling a fluid from one location to another, can be centrifugal, regenerative turbine, or special effect (see Fig. 1).

3.1.1. Overhung Impeller

3.1.1.1. Close-Coupled Single-Stage Horizontal End Suction. A closed-coupled pump has an impeller mounted directly on the shaft of the driver, thus eliminating the need for pump bearing housing. The driver bearings take all pump loads. These pumps are used for relatively light-duty services. They are often applied for sanitary and corrosive pumping requirements because of a clean-in-place (CIP) capability, ie, the pump can be flushed (cleaned) without much disassembly. This type of pump can handle liquids and semisolids having entrained vapors, pump viscous products, and sustain good vacuum characteristics under varying conditions. Designs are simple, and maintenance is easy. These centrifugal pumps use only four basic parts: pump housing or casing, impeller, shaft, and a seal. The shaft, connected to the power source, eg, a motor, rotates the impeller inside the casing.

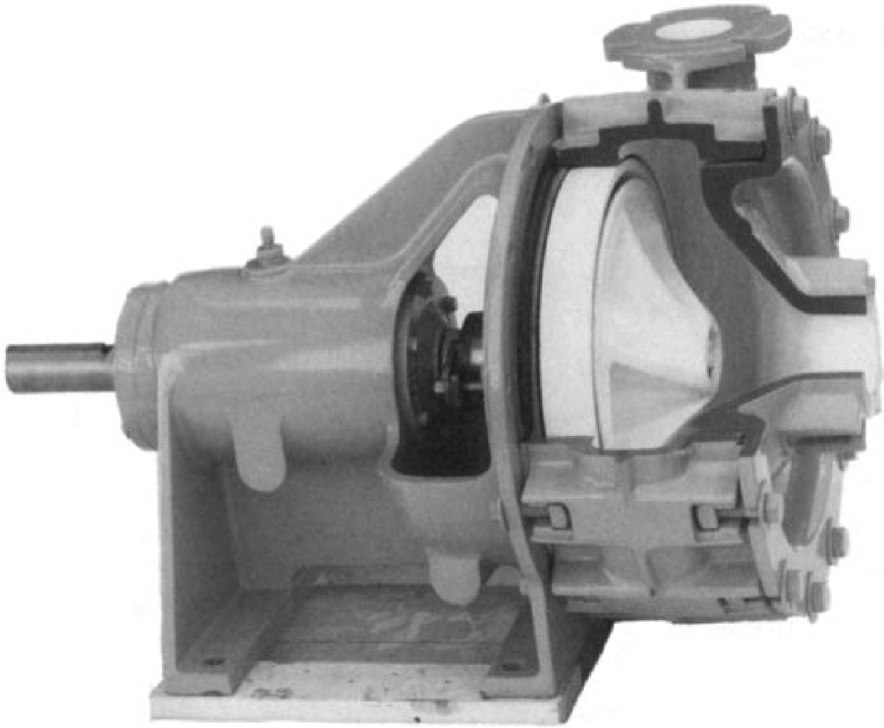

Fig. 5. Frame-mounted pump having recessed impeller.(Courtesy of Wemco Pump Co.)

3.1.1.2. Close- and Separately Coupled Single-Stage Vertical In-Line. Close-coupled vertical centrifugal pump designs have been largely replaced by separately coupled pumps. The latter eliminate hydraulic loads on the motor bearings and are also less prone to hazard if pumpage splashes on the motor, potentially causing fire in the event of a seal failure. The pump and the motor are separated by the bearing frame, providing better rotor rigidity and resulting in lower deflections and improved life of seals and bearings. These centrifugal pumps find applications in petrochemical and refinery plants.

3.1.1.3. Frame-Mounted. For medium and severe applications, when nozzle loads and thermal transients tend to impose high stresses and internal deflections inside the pump, frame-mounted designs are used. Typically, these designs use conventional impellers, but recessed impeller designs are also available (Fig. 5).

Recessed impeller design allows for solids handing. This design has somewhat lower efficiency in comparison to a conventional impeller design. In the latter, the impeller takes up all available space in the casing, giving much tighter clearances between the impeller and walls. Tighter clearance reduces leakage, vortexing and churning losses, thus resulting in better efficiency.

3.1.1.4. Standard Chemical Pump. In 1961, the American National Standards Institute (ANSI) introduced a chemical pump standard (29), known as ANSI B73.1, that defined common pump envelope dimensions, connections for the auxiliary piping and gauges, seal chamber dimensions, parts runout limits, and baseplate dimensions. This definition was to ensure the user of the availability of interchangeable pumps produced by different manufacturers, as well as to provide plant designers with standardized equipment. A typical ANSI chemical pump, known as of the mid-1990s as ASME B73.1M-1991, is shown in Figure 6.

3.1.1.5. Centerline-Mounted Refinery Pump. For refinery applications, temperatures, pipe loads, and product flammability are the prime factors in selecting a pump. The pump-mounting feet are located close

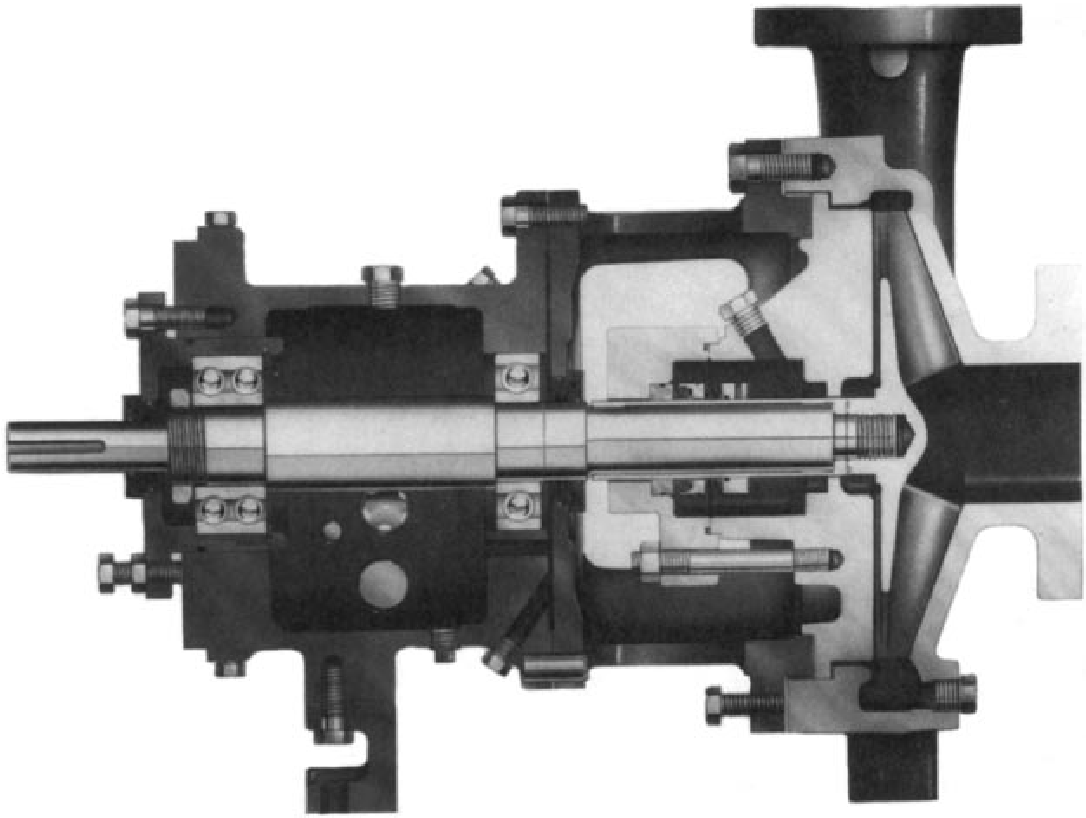

Fig. 6. ANSI chemical pump design.(Courtesy of Goulds Pumps, Inc.)

to the centerline. As a result, the pump thermal growth is uniform on both sides of the centerline, leading to minimal distortions. The design is governed by the American Petroleum Institute (API) standard 610, "Centrifugal Pumps for General Refinery Service" (30).

3.1.1.6. Wet-Pit Volute, Submersibles, and Dry-Pit Designs. For sump pumping applications, three basic approaches are taken. In the first, called a wet-pit design (Fig. 7**a**), the pump is lowered to the pumping depth, but the motor is mounted above the highest anticipated liquid level and coupled to the pump impeller via a long shaft or a series of connected shaft sections. The advantage of this arrangement is the ability to utilize a standard vertical motor, which remains on the ground surface. The disadvantages are the long shaft and greater susceptibility to high vibrations, unless special care is taken to determine lateral critical speeds (31).

For greater depths or as an alternative to the wet-pit design, a submersible pump (Fig. 7**b**) having a special water-tight motor can be used. Significant improvements have occurred in the designs of these motors (the most critical issue is sealing) so that the submersible pump is a reliable option to a wet-pit pump.

Yet another option is the dry-pit design (Fig. 7**c**). This pump is installed in a dry pit and connected to a well via a pipe. Because the dry pit is usually dug out wider than the wet pit, enough room is available for pump maintenance, troubleshooting, and repair without pulling the pump to the surface for servicing.

3.1.1.7. Axial-Flow Propeller-Type. The axial-flow propeller-type pumps are used to handle large volumes of pumped liquid at low head. Flow rates of $\geq 12\ m^3/s$ ($\geq 200,000$ gal/min) can be accommodated.

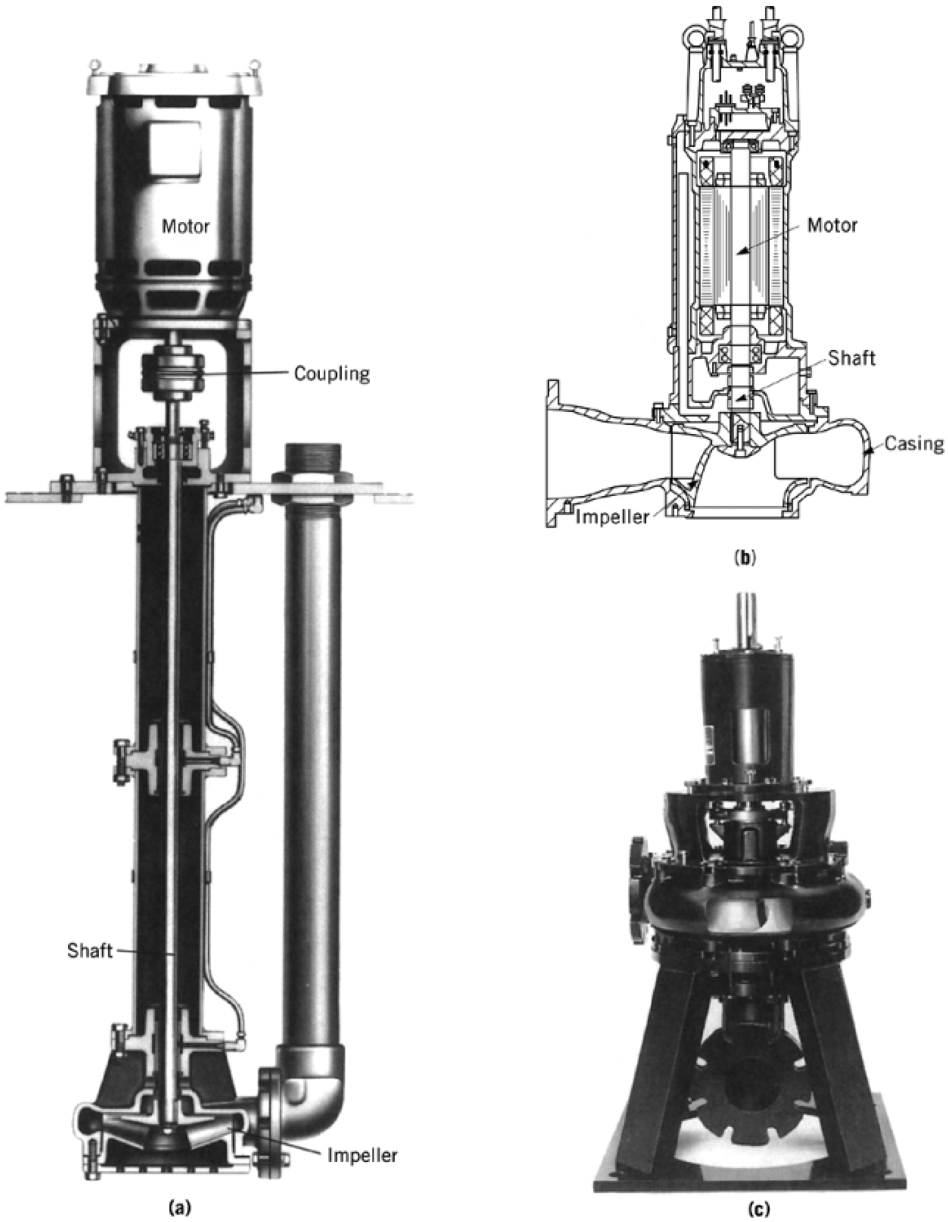

Fig. 7. (**a**) Wet-pit pump design. Courtesy of Goulds Pumps, Inc. (**b**) Submersible pump. Courtesy of Homa Pump Technology. (**c**) Dry-pit pump.(Courtesy of Patterson Pump Co.)

3.1.2. Impeller Between Bearings

These pumps are grouped into single- or multistage designs, and each type is available in either axially or radially split configurations. Sometimes axially split designs are referred to as horizontally split. The term axial, however, is broader. The axial split is not always in the horizontal plane, but can be at an angle to it.

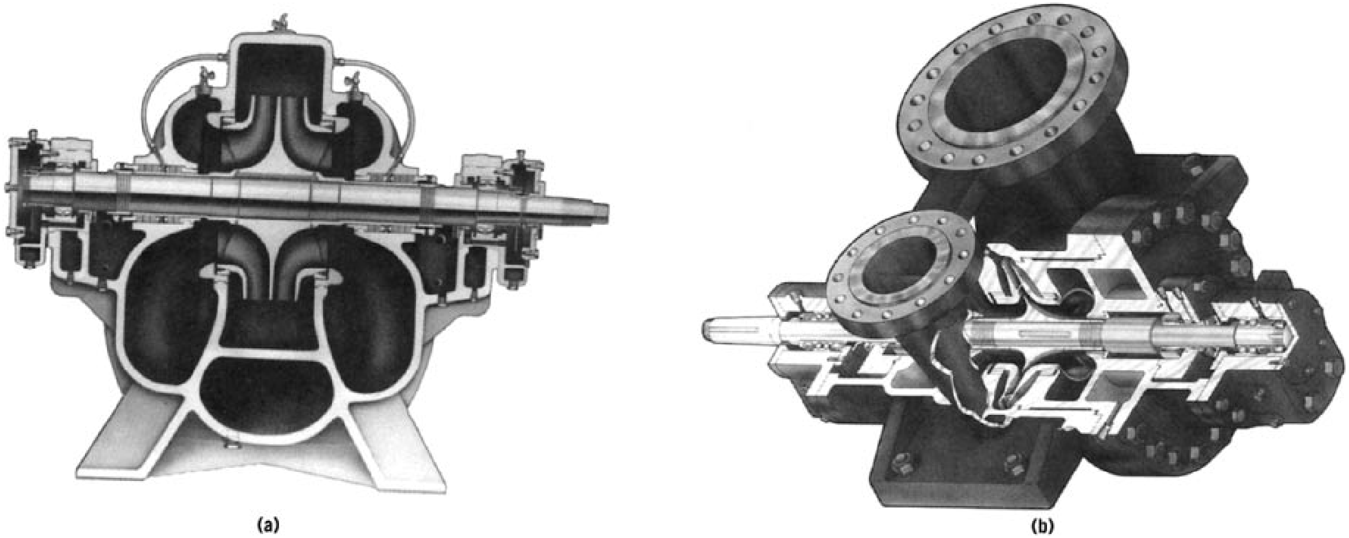

Fig. 8. Single-stage between-bearings double-suction pumps: (**a**) axially split design; and (**b**) radially split design.(Courtesy of Goulds Pumps, Inc.)

3.1.2.1. Single-Stage. Axially split pumps are designed for convenient maintenance, inspection, replacement of the impeller or wear parts, and easy access to pump internals. The suction and discharge piping is connected to the lower half of the casing, and does not need to be disturbed for rotor removal. Owing to the nature of the axially split configuration and the large area of the gasket required to seal the two halves, these pumps are limited to relatively moderate developed heads (150 m (500 ft)) and temperatures under 175°C (350°F). They are used as cooling-water pumps at process plants and paper mills, for paper stock transfer, as pipeline pumps of desalinization plants, and for a variety of other applications. Sizes vary from small (1.2 m^3/min (300 gal/min)) to large (>300 m^3/min (80,000 gal/min)).

For the high pressure applications as well as more critical hot-temperature applications such as oil refineries service, radially split casing designs are required. These designs, governed by API 610 specification, allow better sealing at the vertically split joint by using a fully confined gasket. Because these pumps see higher pressures than the axially split pumps, they are generally of a more robust design, such as thicker walls, centerline mounting, ribbing,and stiffer baseplates. A comparison between these two designs is shown in Figure 8.

3.1.2.2. Multistage Designs. There are traditional differences in the design philosophy between the axially and the radially split multistage pumps. Historically, the axially split design was preferred in the United States, whereas a radially split (segmental) design was preferred in Europe, especially for moderate pressures and temperatures. There are exceptions, of course, and both designs have been applied successfully worldwide.

An axially split pump typically has a volute, and offers the advantage of easy maintenance because of quick upper-half casing and rotor removal that is similar to a single-stage version. The piping, attached to the nozzles located at the lower half of the casing, remains undisturbed.

Radially split construction is typically a diffusor design, offering the advantage of reduced radial thrust, which is better from the standpoint of lowering shaft deflections, resulting in longer life of seals and bearings. The reason for radial thrust reduction is an inherent geometrical symmetry of the diffusor (having multiple vanes), which results in uniform pressure distribution at the impeller periphery at wide flow range. The volute design has an uneven pressure distribution, off-BEP which results in radial load. The double volute has two tongues, called cutwaters, which, as compared to a single volute, results in more uniform pressure distribution around the impeller periphery. This distribution, however, is not as uniform as in a diffusor design. The disassembly of the diffusor pump is more involved because a complete pump must be disconnected from the piping for service. Both designs are limited to temperatures around 175°C (350°F), as determined by gasket sealing and safety considerations. For more critical and higher temperature applications such as boiler feed services and refinery applications of volatile or hot ($>175°C$) temperatures, a segmental ring design is enclosed into a cast or forged barrel, which becomes a pressure-containing vessel having a fully confined gasket. This gasket seals the ends via suction and discharge heads.

3.1.3. Verticals- or Turbine-Type

Historically, all diffusor-type pumps were called turbine pumps (32) because hydraulic turbines have nozzle vanes and resemble the diffusors. Although the word turbine, as applied to vertical pumps, is obsolete, this definition is still widely used in the industry as of the 1990s. Applications for the turbine pumps include agricultural irrigation, storm water drainage, hot-well and booster pumps in power plants, refinery applications for low NPSH requirements, and more. Designs vary in complexity, cost, and robustness, depending on applications.

3.1.3.1. Deep-Well Turbine and Canned Version. By stacking up the stages, heads up to 1100 m (3500 ft) can be achieved. The deep-well pump is submerged into the well, and liquid comes into contact with the pump body and the well walls. Alternatively, a pump pit may be of dry design, and a can, ie, an enclosure around the pump, having connections to the source of liquid is provided (Fig. 9).

3.1.3.2. Axial-Flow Propeller-Type. For high capacity, single-stage propeller-type vertical pumps are used. One important application of this design is as an intake of cooling water from a river or a holding basin via a pipeline.

3.1.4. Specialty Pumps

There are a multitude of other pump designs in the family of kinetic pumps. Many of these designs have been around for years; others continue to emerge. Some of these special-purpose pumps are described herein.

3.1.4.1. Vacuum Pumps. Vacuum pumps are more related to compressors than to pumps. What keeps these pumps in the hydraulic family is that they utilize a "liquid ring" concept to evacuate gas, usually air, from the vessel or pipe. These can be of a single-, two-, or multistage design. A typical two-stage design can function to 735 mm (29 in.) Hg of vacuum. The pump size determines how fast the desired vacuum can be reached in a given size vessel. Explanation of the operating principle can be found in Reference 33.

3.1.4.2. Self-Priming Pumps. Self-priming pumps are designed to operate in two regimes. The first, utilizing a liquid ring principle, is as a vacuum pump for a short period of time, during which the gas, eg, air, is being evacuated. This is called the priming cycle. When the pump is fully primed, a second regime begins automatically. The pump operates just as a conventional centrifugal pump. Typically, such pumps can self-prime up to a 6–8-m (20–25-ft) lift, and take 30–90 s of priming time. The pump must be filled with liquid manually once, for initial prime, to ensure that there is liquid in the casing, and to create a liquid ring effect, which is required for air removal during the priming cycle.

3.1.4.3. Disk Pumps. When pumping shear-sensitive or more viscous fluids, it is desirable to reduce internal turbulence caused by the vanes. The disk pump design relies on the centrifugal frictional effect of a vaneless disk. Whereas the efficiency of this pump is lower than that of similar centrifugal pumps with vanes, it is often the only solution to certain pumping applications.

3.1.4.4. Jet Pumps. A jet pump is a clever way of using a high pressure motive fluid and a venturi to aspire a low pressure fluid to an intermediate pressure level. The high pressure fluid can be liquid, vapor, or gas, which enters the pump suction and exchanges its potential energy to a kinetic energy of the fluid.

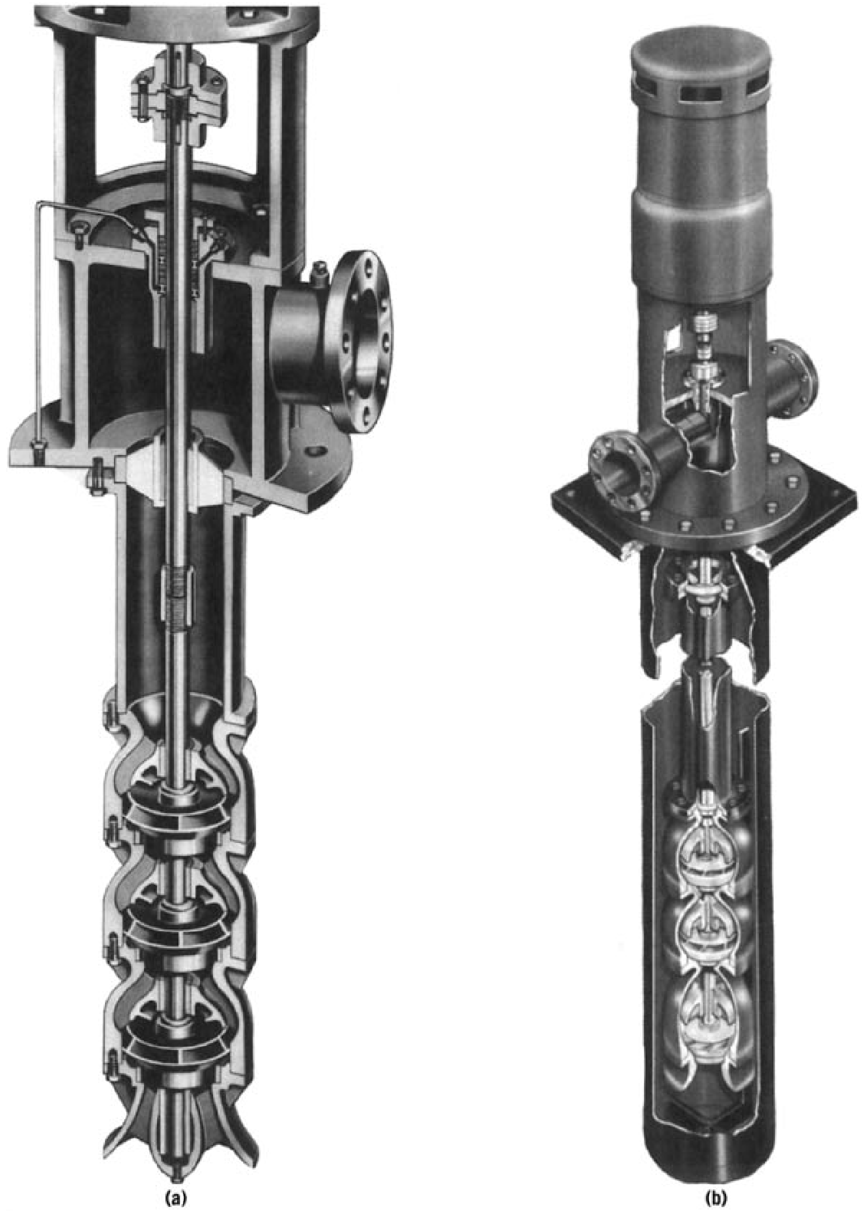

Fig. 9. Turbine-type pumps. (**a**) Deep-well wet. Courtesy of Goulds Pumps, Inc. (**b**) Canned version.(Courtesy of Johnston Pump.)

There are other unique types of kinetic pumps, serving different applicational niches of industrial and residential needs. Among these are reversible centrifugal, rotating casing (pitot), regenerative turbines, and inertia pumps (32).

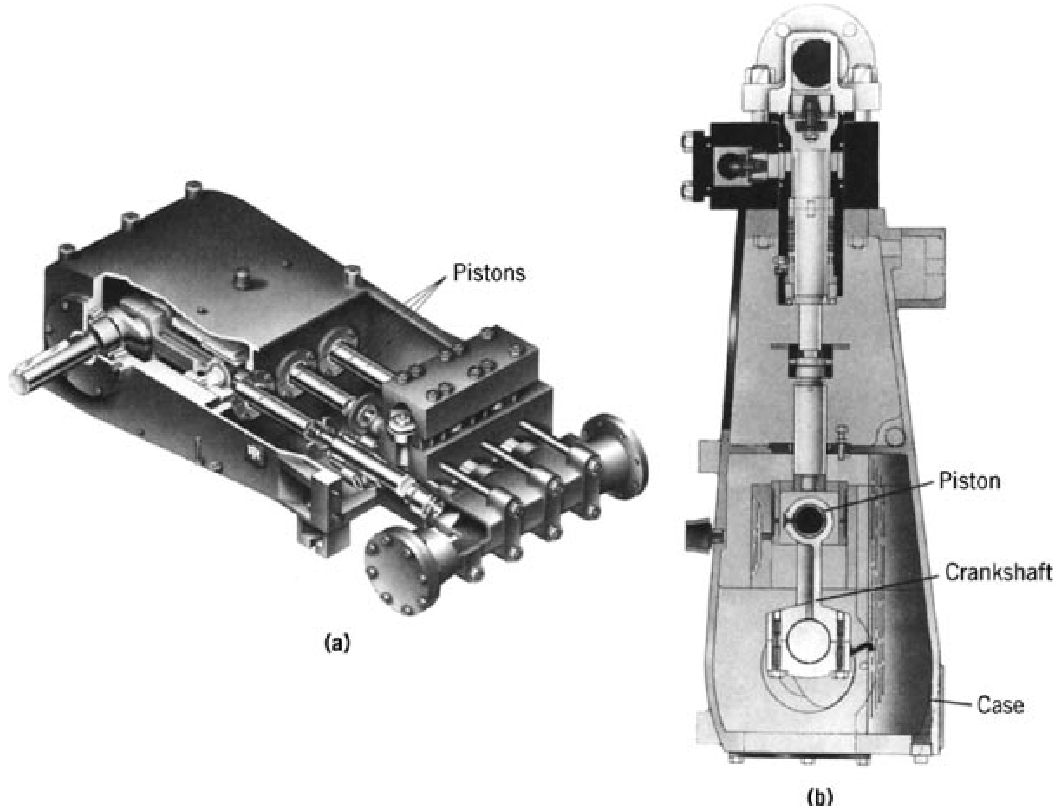

Fig. 10. (**a**) Reciprocating pump and (**b**) cross-sectional view.(Courtesy of Ingersoll-Dresser Pumps.)

3.2. Positive Displacement Pumps

Positive displacement pumps follow HI convention (see Fig. 1). As a rule, these pumps work against significantly higher pressures and lower flows than do kinetic, particularly centrifugal, pumps. Positive displacement pumps also tend to operate at lower rotational speeds. There are many types of positive displacement pumps, for which designs are constantly being developed. Some of these are discussed herein.

3.2.1. Reciprocating Pumps

A classic example of a positive displacement pump is that of the reciprocating pump (Fig. 10), in which the principle of operation is probably the easiest to understand. These pumps are the most familiar to a nonspecialist. The energy of the reciprocating piston of this pump generates pressure, causing liquid to move from the chamber in which the fluid is acted upon by the piston, to the discharge piping. On the return stroke, the space vacated by the retrieving piston is filled by the liquid, which is moved by the suction pressure. Required valving and timing mechanisms supply necessary opening and closing of ports to provide overall transfer of liquid from suction to discharge.

By the nature of the design, the biggest problem with the application of reciprocating pumps is high pressure pulsations. To reduce pulsations, multiplex (multipiston) pumps are used. The frequency of pressure pulses is decreased in multiplex pumps owing to an increase in the number of pistons properly spaced in phase. Designs of up to 16 cylinders are available. In addition, special pressure-dampening devices are commonly

Fig. 11. Radial and lateral clearances of gear pumps. A stainless steel pump employs stainless drive gear diving the Teflon idler. Shafts are supported by the carbon bushings and carbon wear plates. Courtesy of Liquiflo Equipment, division of Picut Industries.

used near the suction and discharge flanges, as well as at certain critical locations along the piping, where the pressure pulsations would otherwise be unacceptably high (34). Pistons are generally used to 7 MPa (1000 psi), and plungers from 7 to 207 MPa (1,000–30,000 psi).

3.2.2. Rotary Vane Pumps

These pumps are often used for liquefied gases, solvents, light oils, and similar light products. As pump designers tried to combine the high throughput capabilities of kinetic, mostly centrifugal, pumps and the high pressure capabilities of positive displacement pumps, various designs spanning these two extremes emerged. The rotary vane pump features an eccentric rotor having multiple plates (vanes). The rotation of the rotor translates into a reciprocating movement of pistons. This results in pumping action. These pumps are known to pump very thin (low-viscosity) fluids, such as in gasoline truck unloading.

3.2.3. Gear Pumps

These are the most general-purpose and widely applied of all rotary pumps. Applied in a wide range of applications, from heavy viscous polymers, to thin aggressive chemicals, gear pumps are available in a wide range of metallurgy, sizes, and speeds. A typical external gear pump is shown on Fig. 11. Liquid enters the inlet port, fills the space between gear teeth, and is trapped in these spaces by the gear teeth and pump housing. It is carried around to where the gear teeth mesh, which forces the liquid out through the discharge opening. The rotor speeds are greater than those for reciprocating pumps, leading to a smaller pump size for the same throughput. Many designs allow operation at full motor speed (1750 rpm), which eliminates the speed reducer, optimizes the pump size, and results in a cost-effective design. Pressure pulsations are also significantly reduced.

Specific speed (Ns) is one of the most popular dimensionless parameters, used in centrifugal pumps. Positive-displacement pumps, however, do not use it, since the affinity laws do not apply here. Gear pumps are rotary machines, a subset of the positive displacement family, use a *unit flow*, which characterizes a throughput capacity per revolution, and thus relates to a pump size, With each revolution of a shaft, a known constant amount of fluid is “generated” by the gear pump. Thus a flow is directly proportional to speed:

$$Q_0 = q_0 \times \text{RPM}$$

The coefficient of proportionality (q_0) is unit flow. If pump flow is in gallons per minute (gpm) and rotational speed is in revolutions per minute (rpm), the q_0 constant is then in gallons per revolution (gpr). Other units can be used also; for example, if flow is in cm^3/min, then (q_0) is in cc/rev.

The index "0" means "theoretical" flow, ie, assuming zero differential pressure. However, in the real world, there is always differential pressure across the pump, which drives some amount of fluid back, from discharge side to suction, through the clearances (radial, between the gear tips and the casing, and lateral between the gear ends and the endplates, or wearplates), as shown in Fig. 11.

The actual pump flow is thus less then theoretical, by the amount of *slip*:

$$Q_{\mathrm{net}} = Q_0 - Q_{\mathrm{slip}}$$

The amount of slip depends primary on three things: differential pressure, fluid viscosity and pump clearances. It also depends on rotational speed, but not very significantly.

Gear pumps are commonly used for chemicals, such as transfer, dozing, and process control. Materials of construction for chemical applications typically include stainless steel (such as 316ss) casing, and stainless drive gear against Teflon idler gear. *Stainless drive gear against stainless ilder gear should not be used* for low-viscosity liquids (such as chemicals), as such combinations of running pairs of similar stainless steel causes galling (transfer of metal from one mating part onto another), seizing and failure. Design should include the renewable wearplates, typically of carbon, Teflon, or other materials.

For high-pressure metering applications, such as viscous polymer injection, where pressures reach 1000–4000 psi, gear pump designs use martensitic stainless steels, with very tight clearances, to ensure accuracy as precise as 0.1%. Because due to extreme precision, these pumps are more expensive. However, such pumps should not be applied for chemicals, since martensitic stainless steels are significantly less resistant to chemicals as compared to stainless steels. Also, with tight clearances, in the absence of good lubricating fluids (chemicals are usually thin fluids witty poor lubricating qualities), the rotating parts can easily seize up in the clearances.

Gear pumps for chemicals are also made in higher alloy materials, such as Hastelloy, titanium, and other combinations. An example of a gear pump for tough chemicals is shown on Fig. 12.

A good practical reference on gear pump applications, and various methods of flow control is in (68).

Just as for centrifugal pumps, improper inlet conditions can cause suction problems for the gear pumps. Instead of specifying the NPSHr (centrifugal pumps), rotary pumps use the minimum required suction pressure (p_{smin}), needed to ensure fluid properly entering the inlets and filling the spaces between the gear teeth completely. The Hydraulic Institute defines p_{smin} as such value of suction pressure when a rotary pump looses 5% of its flow (compare with 3% head loss for centrifugal pumps).

3.2.4. Rotary Lobe Pumps

Rotary lobe pumps are similar to gear pumps in principle. These pumps have an added advantage of non-contacting metal parts by use of external gears, which reduces the wear, but adds complexity. Low wear and improved shear characteristics make these pumps applicable to the food industry, where cleanliness and absence of contamination are required and where lobe pumps have been traditionally used, with typical flow rates 200–300 gpm. These pumps produce a strong pulsing flow that must be addressed.

3.2.5. Multiple Screw Pumps

Constructed by using external timing gears and bearings, two-screw pumps are suited for capacities between 0.1–35 m^3/min (20–9000 gal/min), high suction lift capabilities, and differential pressures of up to 10 MPa (1500 psig). Available in a wide range of materials, two-screw pumps are utilized in a variety of applications in the petrochemical, refining, offshore, and production industries. Three-Screw pumps is another variation of multiple-screw design type, - it is generally less expensive and is used for higher viscosities, as compared to two-screw pumps, which are known to pump very thin liquids, such as naphtha, a fuel, used in gas turbine plants in some countries.

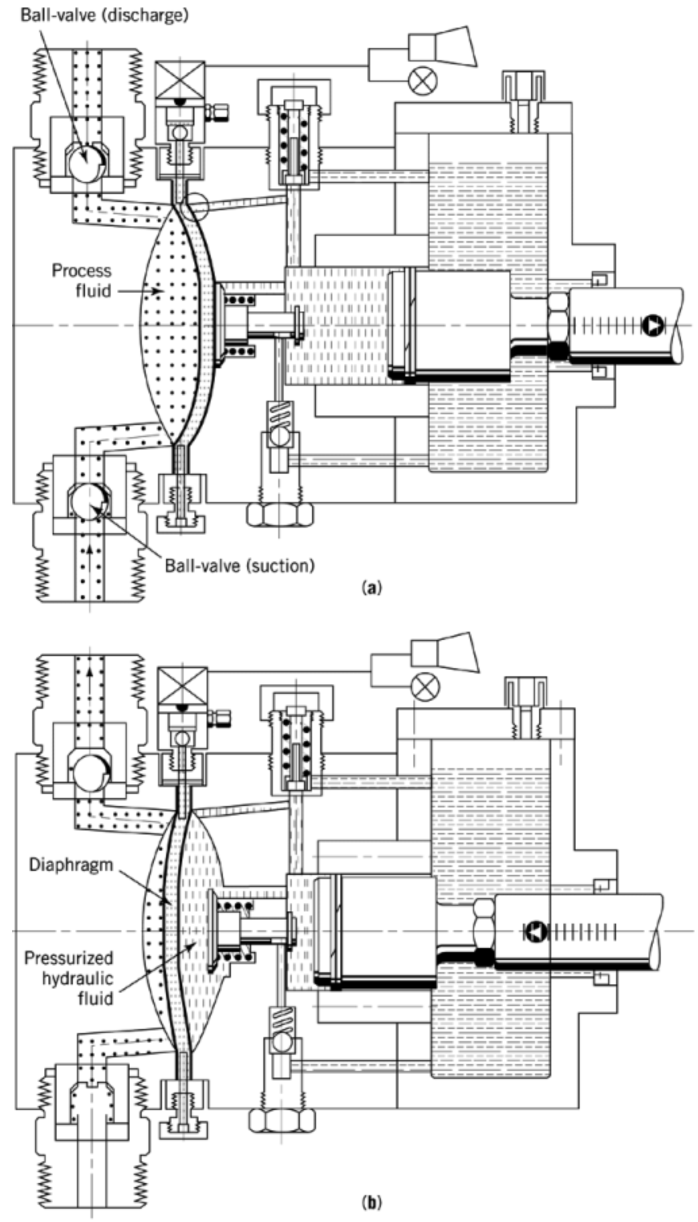

Fig. 12. Gear pumps (sealed and seal-less mag-drives) for chemicals applications. Courtesy of Liquiflo Equipment, division of Picut Industries.

3.2.6. Progressive Cavity Pumps

Capable of handling viscosities up to $1,000,000$ mPa·s (=cP), progressive cavity pumps (Fig. 13) are suitable for transferring fluids containing high percentages of solids. Additionally, progressive cavity pumps impart extremely low shear into the fluid being handed as a result of their gentle, pulsation-free transfer of the media

Fig. 13. Progressive cavity pump.(Courtesy of Monoflo Pump Co., division of Haliburton.)

through the pumping elements. Their typical applications are in wastewater sludge processing, and for other extremely viscous fluids.

3.2.7. Flexible Tube

The simplicity of design and the absence of seals and valves make the flexible tube or peristaltic pump a good choice for low capacity and low pressure applications in the pharmaceutical industry or wherever shear-sensitive or moderately abrasive fluids are pumped. Because of the continuous flexing of the tube, the tube material of construction presents a challenge regarding life cycle. For the same reason, pressures are kept relatively low. Due to high pulsations, peristaltic pumps have low speeds.

3.2.8. Flexible Impeller Pumps

Flexible impeller pumps are designed for general-use service. They can be utilized for pumping in either direction, have low pulsations, and are easy to service and maintain. These pumps are typically limited to low capacities and pressures.

3.2.9. Diaphragm Pumps

Diaphragm pumps translate the reciprocating action of a piston into displacement of hydraulic oil (see Hydraulic fluids). This causes controlled flexure of the diaphragm within the pumping chamber and the equivalent displacement of process fluid. The diaphragm is hydraulically balanced and statically sealed at its outer diameter. As is the case with other reciprocating pumps, one-way check valves are used. These pumps can operate at high discharge and suction pressures up to 35 MPa (5000 psi) using polytetrafluoroethylene (PTFE) diaphragms, or even higher pressures using metal diaphragms. The pumps are available in a wide range of sizes. Flow can range from a few mL/h up to 170 m^3/min (45,000 gal/min). Such low capacities result in a formal way of designating capacity specification in the pumping industry, ie, using capacity per hour, rather than minutes. Common uses of low-flow diaphragm pumps are for metering and for controlled-volume process services in the chemical process industries.

4. Application Examples

Selection of a pump for a given application is not a trivial task. Often more than one pump type can accomplish the required job. Thus a final choice on a pump type is often a result of personal experience and usage history. As a rule of thumb, the choice of a kinetic, such as centrifugal, or a positive displacement pump is made on the basis of the specific speed. Whereas specific speed is applicable primarily for centrifugal but not positive displacement pumps, the *NS* value can be used as a guide. Generally, for calculated values of specific speed, eg,

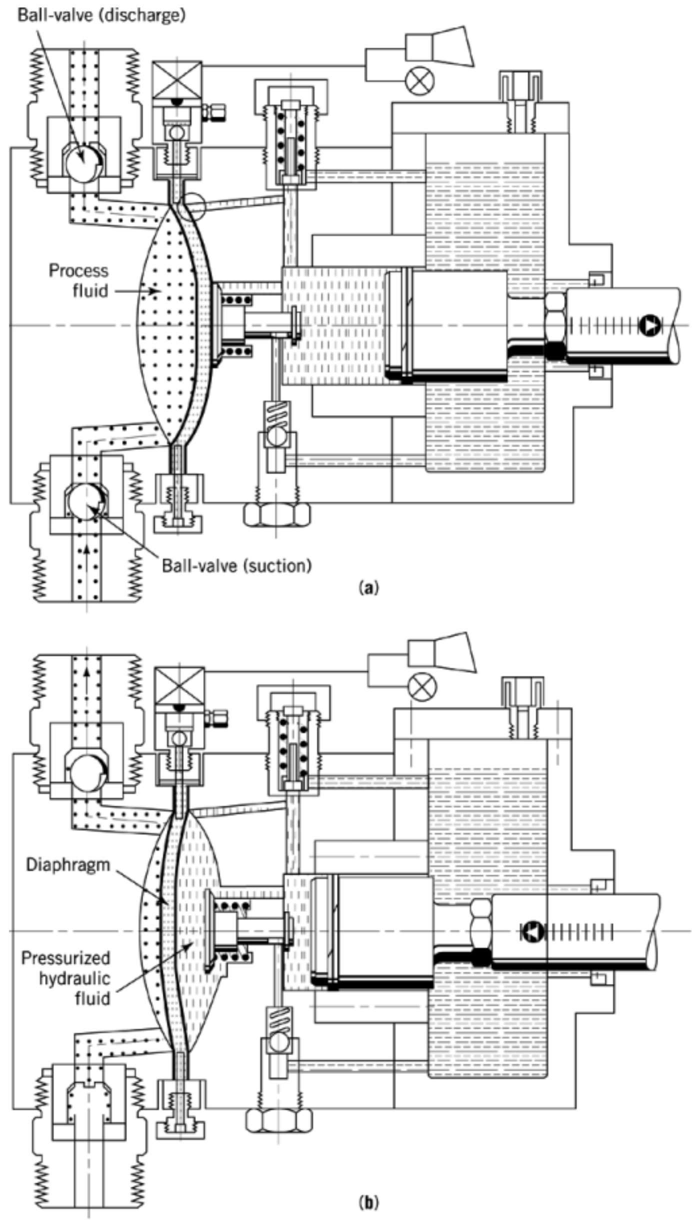

Fig. 14. Diaphragm (Small) pump: (**a**) suction cycle, and (**b**) discharge cycle.(Courtesy of American Lewa Inc.)

$nS > 10$ ($NS > 500$), kinetic-type pumps are usually selected. For $nS < 10$ ($NS < 500$), positive displacement pumps are typically applied.

From the definition of specific speed (eqs. 9 and 10), it follows that reciprocating pumps operate at high pressures and low flow rates. Conversely, centrifugal pumps are applied at lower pressures and higher flow rates. Many rotary pumps are selected for viscous liquids having pressures equal to or less than, and capacities lower than, centrifugal pumps. However, these limits are relative and a gray area exists as some pump types

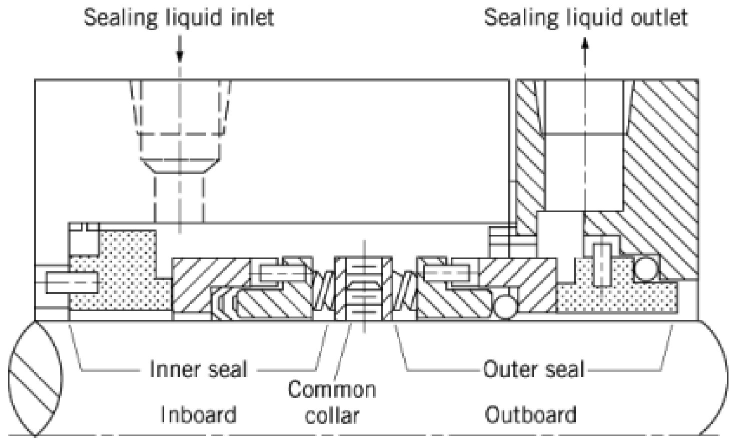

Fig. 15. Magnetically coupled pump, where the outer and inner drives are magnets. A pumped liquid provides lubrication to bearings of the inner (driven) rotor.(Courtesy of Liquiflo Equipment, division of Picut Industries.)

cross boundaries into the domain of other types. For example, gear pumps are applied successfully for chemicals, even at extremely low viscosities, low flows, and pressures often higher than feasibly achievable by centrifugal pumps.

4.1. Plastic Pumps for Corrosive Liquids

The main limitation is lower pressure and temperature capabilities. If plain carbon steel or iron pump metallurgy is applied for corrosive pumpage, the damage to shafts, impellers, casings, and other wetted parts can be quick and extensive, seriously affecting performance, efficiency, and reliability (see Corrosion and corrosive control). For some applications, even a trace of metallic contamination cannot be tolerated. For these reasons, as well as for extremely broad chemical resistance, a variety of polymer pump designs have been developed.

Nonmetallic centrifugal pumps utilize polymers for all components coming into contact with the pumping media. Even the steel shaft is sleeved with plastic to isolate the shaft from the fluid. Mechanical seals are reversed so that the metal face does not come into contact with the corrosive liquid. A wide variety of nonmetallics have become available for such applications (35). These pumps are furnished with casing, flanges, and an impeller molded from solid homogeneous polymers, eg, polypropylene. Most polymers are limited to chemical applications of $\leq 120°C$ ($\leq 250°F$).

4.2. Pumps for Leakage Prevention

When leakage to atmosphere must be contained to essentially zero, magnetically coupled or canned motor sealless pumps can be used. These pumps do not have rotating seals between the pumped liquid and the atmosphere.

In magnetic drive designs, the torque is transmitted from the motor and drive magnet assembly to a driven magnetic assembly having a driven shaft and impeller. The drive and driven assemblies are separated by the containment shell. The joint between the shell and the adapter or casing is sealed by an O-ring or gasket, which is stationary and provides reliable sealing of leakage, as compared to rotating seals between the shaft and the stationary parts. An example of such a pump is shown in Figure 15

4.3. Pumps for Extendly Corrosive and Toxic Applications

For some applications, pumpage is not only severely corrosive but also toxic, so that the pumpage must be contained with zero leakage (see Waste treatment, hazardous wastes). One possibility is to employ a magnetically driven pump having a nonmetallic, usually Teflon, liner at the liquid end.

Another possibility is a canned motor pump, which is essentially a pump and an induction electric motor built together into a single self-contained unit. The pump impeller is mounted on one end of the rotor shaft that extends from the motor section into the pump casing. The rotor is submerged in the fluid being pumped. The stator is isolated from the pumped fluid by a corrosion-resistant nonmagnetic alloy liner. Bearings are submerged and continually lubricated by the process fluid.

Canned motor pumps have only one moving part, a combined rotor–impeller assembly driven by the magnetic field of the induction motor. A small portion of the pumped fluid is allowed to recirculate through the rotor cavity to cool the motor and lubricate the bearings. On some designs this recirculation fluid can be filtered to extend the life of the bearings. Canned motor pumps have fewer bearings as compared to other pump types and do not require any special alignment. A close-coupled centrifugal pump can be compact, and operates with less noise because there are no external moving components. This type of pump finds many applications in hazardous and toxic pumpage because the design can offer secondary containment of the process fluid. High pressure designs are also available, which utilize backup sleeves to support the stator liner. This pumps, however, can not be used for applications containing solid particulates, or if a possibility of dry running (such as complete emptying of a tank, or a suction valve inadvertent closure, suction side filter clogging, etc) is real. Since the operation of regular mag-drive and canned motor pumps depends on lubrication of their internal journal bearings by the pumped fluid, the loss of pumpage (dry running) is a real problem. A modification of a mag-drive design, called a gas-barrier technology can be applied for such problem applications, as explained in section on dynamic sealing.

4.4. Pumps for Slurries

In mining and minerals applications (see Mineral recovery and processing), abrasive wear by hard particles such as rocks, coal, and other slurries can cause quick damage to a pump's liquid end unless special abrasion-resistant materials are utilized. The wet-end parts, such as the casing, can either be made of hard metals or use solid liners. For certain applications, rubber-lined casings have been successful. Rubber has a unique quality to resist wear because of its ability to comply and resist the shearing wear of particles. Rubber linings often provide good choice for applications where both abrasion and corrosion can occur, such as desulfurization pumps for liquid ash transport at power plants. Because the flow of slurries is multiphased, ie, composed of a liquid having solids and entrained gases, the nature of such a pumping regime is unstable. This results in fluctuating radial and axial loads, which have detrimental effect on pump bearings and seals. To withstand these loads, a heavy-duty support frame having heavy-duty bearings, often roller rather than ball bearings, are used. These designs provide rugged, dependable operation for most demanding applications.

4.5. Trash or Contractor Pumps

For general applications where trash and debris are present in the pumped liquid, medium-size submersible centrifugals or diaphragm self-priming pumps can be utilized. These pumps may be equipped with inlet screens to prevent grit and sand from entering the internal parts of the pump.

4.6. Metering Pumps

For small flow rates, such as dosing chemical additives where precise control is required, such as rotary gear pumps, diaphragm pumps. They can be packaged as self-contained pumping units or mounted on a baseplate with a flow meter with 4–20 mA signal control to automatically adjust the pump speed, and to keep constant flow. Ref. (68) is a good overview of metering variations by gear pumps.

4.7. Pumps for Nonclogging Applications

When pumping stringy fibrous materials, it is desirable to keep the internal passages as wide as possible. All sharp edges must be blended, and impeller vanes made thicker at the inlets for contouring purposes. Additionally, special designs are available, where the entire rotating assembly can be removed and the clog cleared.

In certain applications, such as canning in food processing, the pumped solids, ie, fruits, vegetables, etc, must survive the pumping action. At the other extreme, when preservation of pumped solid particles is undesirable, special chopper pumps are used. These pumps break up the incoming solids into smaller pieces, thus enabling the main impeller to pump without plugging. Chopper pumps are also useful in the chemical industry for handling waste sump applications, eg, where solids tend to plug standard nonclogging pumps. These chemical industry applications include detergent cakes, hand cleaners with pumice, latex skins, lead oxide slurries, paint sludges, plastics, and shredded filter pads. Progressing cavity, diaphragm, and hose (peristaltic) pumps are also popular in these applications.

5. Couplings and Seals

Various coupling designs are available to transmit torque from the driver, eg, electric motor, to a pump. In order to contain the pumped fluid inside the pump and prevent the pumpage from leaking, several types of sealing methods are used. A few options are described herein.

5.1. Couplings

5.1.1. Gear-Type Couplings

Gear-type couplings (Fig. 16**a**) are usually used for heavy-duty applications having high torques. Gear mesh allows a small parallel misalignment between shafts, usually less than 0.2 mm (0.008 in.). Gear requires lubrication (see Lubrication and lubricants), usually of a grease type.

5.1.2. Disk Coupling

Diaphragms are used to transmit torque between the inside and outside diameters of the flexible element in disk coupling (Fig. 16**b**). Up to 0.1-mm (0.004-in.) parallel misalignment can usually be tolerated. Whereas no oil lubrication is required as compared to gear couplings, disks are limited to moderate torques.

5.1.3. Grid-Type Couplings

Another type of flexible couplings, similar to disk couplings, is the grid type (Fig. 16**c**). These are applied for moderate loads and allow parallel misalignment up to 0.1 mm (0.005 in.). Lubrication is also required.

5.1.4. Special Types

Many other coupling types are available. These vary in degree of complexity, maintenance time, torque capabilities, and price. A good comparison study on coupling is available (36). An elastomer-type coupling is one of the most popular types used, particularly in the chemical industry.

Among relatively new designs are magnetic couplings, sometimes referred to as magnetic drives. When these drives are applied to magnetically driven pumps, a containment shell isolates the inner assembly (driven) from the outer (drive). The revolving magnetic field, which actually performs a coupling action, induces eddy

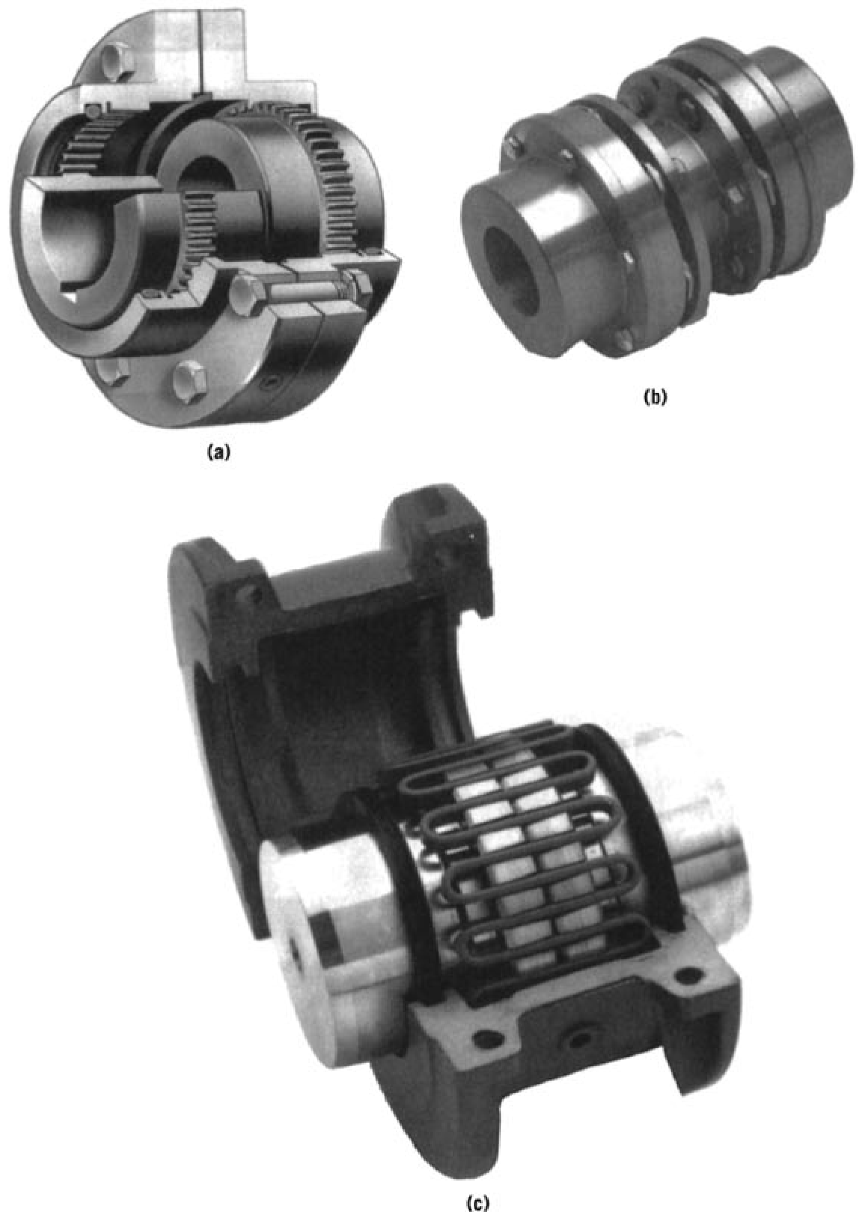

Fig. 16. Couplings: (**a**) gear; (**b**) disk; and (**c**) grid.

currents in the shell if the shell is metallic. The eddy currents result in shell heating, ie, power loss, and lower efficiency. The heat is rejected into pumped fluid, raising the fluid temperature and the vapor pressure. If vapor pressure of the fluid becomes higher than the liquid pressure, vaporization (flashing) occurs, and no more heat is removed from the shell. Shell overheating and distortion occur and may lead to failure. To prevent eddy currents, polymer, ceramic, or similar shells are made. This limits application to the low temperature pumpage, and excludes these shells from more demanding applications such as in refineries.

A unique solution combining the strength of metals with electrical nonconductivity of plastics became available in the early 1990s. In this design, a shell is made of two layers. The inner layer is made of a set of rings, compressed together, and insulated by gaskets. These gaskets contain eddy currents within each small ring, resulting in significant overall reduction in losses. The outer ring has axial slots, which similarly break up large eddy currents into several small ones. The total power loss is small because it is proportional to the square of the eddy current (Ohm's law). The inner shell takes up the radial stresses, and the outer shell takes up the longitudinal stresses. The liquid is contained inside the shell and is sealed statically by gaskets between the rings.

A common requirement for all coupling types, although to a different degree, is the need for good alignment. Alignment minimizes coupling strain and prevents shaft bending. The latter can cause seal leakage and reduced bearing life owing to the high loads caused by misalignment. A straight-edge coupling alignment is sometimes used, especially for low horsepower pumps. However, studies in pumping equipment reliability have demonstrated (37) that significant improvements in bearings and seal life can be obtained by proper maintenance and installation procedures. Better coupling alignment of pumps can be achieved by using precision alignment tools and even laser alignment (38) especially for critical high energy pumps. Good baseplate installation methods make alignment much easier. Improper installation, on the other hand, leads to uneven mounting of the equipment, ie, pump and driver. This can result in high casing stresses, excessive bearing loads, seal distortions, and a reduction in reliability and equipment life (67).

5.2. Sealing

5.2.1. Packings

The oldest method of sealing is through the use of packings. It is also the simplest and least expensive. A packing material is inserted into a stuffing box, which may include a lantern ring to provide injected liquid to flush the box from particulates or to cool it. To prevent the shaft from rubbing wear by packing material, hardened sleeves are often used over the shaft (39). Some packing designs include graphite or Teflon particles for better self-lubricating properties to minimize wear damage. Packings need a certain amount of leakage for lubrication, although the latest developments in packings design have reduced this leakage significantly. Packings are typically limited to noncorrosive, nontoxic applications such as paper mills and waterworks. However, some packing materials, eg, Teflon, have been used in moderately corrosive environments.

5.2.2. Mechanical Seals

Mechanical seals provide good sealing for a wide variety of applications. Leakage rates are considerably less than for packing materials, but the mechanical seals are more expensive. Applied in refineries as standard, mechanical seal emissions can be as low as 50 ppm, much lower than the allowable minimum established by the Clean Air Act (40). The sealed faces are kept together by the force of springs or bellows, or, for unbalanced seals, hydraulically. Mechanical seals can be installed in single, double (Fig. 17), or tandem configuration. Good descriptions of the principles of operation and application guidelines are available (41, 42). If the single seal fails, the pumpage leaks out to the atmosphere. Double and tandem designs have a barrier fluid between the inner and outer seal. In the event of inner seal failure, the outer seal takes over the sealing, the increased seal cavity pressure can be detected, and the pump can be shut down.

5.2.3. Dynamic Sealing

In pulp and paper industry applications, it is often desirable to prevent paper fibers from reaching the sealing chamber. Fibers can cause plugging, overheating, and failure. A device known as an expeller is often successfully utilized in such applications. Based on the liquid ring principle (33), the dynamic seal or expeller keeps a liquid ring within its passage. The radial height of this ring is automatically determined in such a way that the centrifugal force caused by the rotating liquid mass is balanced by the differential pressure across it, ie, the

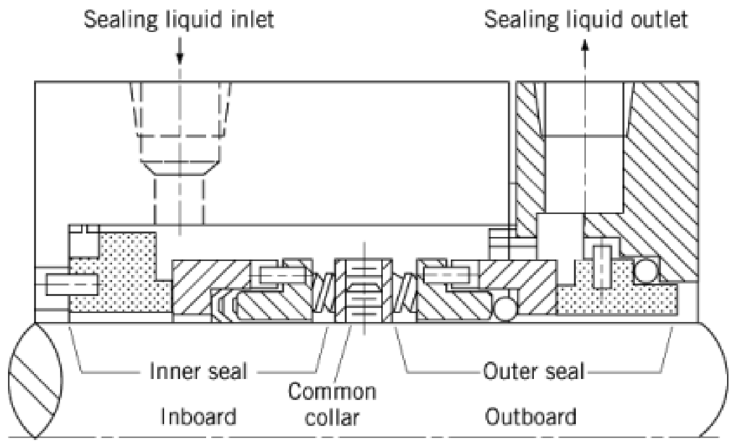

Fig. 17. Double-seal arrangement for mechanical seals.(Courtesy of Durametallic Corp.)

pressure difference between the expeller outer diameter, which is equal to the pump impeller back hub pressure, and the sealing chamber pressure. If the impeller has balancing holes and the sealing chamber is connected to the atmosphere via packing, a differential pressure between pump suction pressure and the atmosphere exists across the expeller. This determines a certain liquid height within the expeller. Thus, if the suction pressure increases, the amount of liquid in the expeller correspondingly increases. At some level of suction pressure the liquid fills the expeller completely, and the expeller reaches the limit of its sealing capability. Upon further increase in suction pressure, paper fibers, if contained in pumpage, can reach inside the seal chamber. Therefore, the application of the dynamic seal is limited by suction pressure.

5.2.4. A GasBarrier Design

In the mid-1990s, a unique design of a seal-less mag-drive centrifugal pump has been developed. As was noted in earlier section on pumps for corrosive applications, regular mag-drive and canned motor pumps cannot run dry, which destroys their bearings and cause pump failure very quickly. These conventional mag-drive pumps cannot pump solids, which clog the clearances of their journal bearings. A GasBarrier design places a gas seal (69) directly behind the impeller, thus isolating the fluid end from the drive end. A regular antifriction ball bearings can then be used, greased for life, and not dependent on lubrication by the pumped fluid. as a secondary containment, a magnetic coupling, with isolation shell between the drive and driven magnets is still employed, in the event the gas seal fails. As a seal-less mag-drive design, the benefits of zero leakage, and thus suited for chemicals and substances where leakage to atmosphere cannot be tolerated at all, these pumps find applications in chemical plants, wastewater facilities, liquid terminals, paper mills, and others. The usage of gas is very small (0.003 sofm), and if even this amount of air can not be allowed into the process stream, nitrogen can be used, at injection pressure of 30–80 psig, depending on the pump size. Another benefit is low shaft overhang, due to bearing in close proximity behind the impeller. This reduces shaft deflections significantly, and allows the pump to operate significantly below the best efficiency point (70), well into the region where the hydraulic radial thrust imposed on the impeller would be prohibitive for a conventional design.

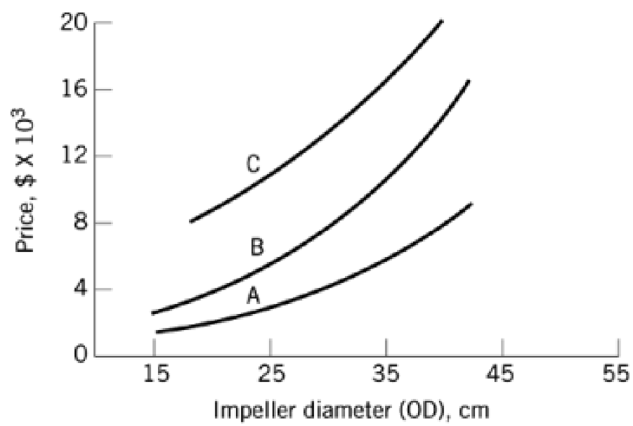

Fig. 18. GasBarrier design, a combination of seal-less mag-drive with gas seal technology, allowing infinite dry running and solids contaminants to 40%. Courtesy of Liquiflo Equipment, division of Picut Industries.

6. Application Guidelines

Pumps are designed to give trouble-free operation for a long period of time. The ANSI B73.1M pumps are designed for a bearing life of no less than two years (29), and API 610 pumps for a minimum of five years (30). However, in real applications, a typical mean time between failures (MTBF) is often found to be significantly less, and sometimes it is as short as a few weeks. Whereas in some installations the seals last from three to four years, in others these are replaced monthly. The reason for such wide variations in pump component life is often not poor pump design but equipment misapplication.

6.1. Inlet Piping Configurations

An incorrect piping configuration can create undesirable flow distortions that, especially for double-suction pumps, means turbulence. Air pockets trapped in the pipe can cause uneven flow, vibrations, and pump damage. For best results, a certain portion of the inlet pipe should be straight, without bends and restrictions, to avoid vortexing and flow turbulence, which could all contribute to cavitation, vibrations, and equipment damage.

6.2. Low Flow Operation

The optimum operation of a pump is near the best efficiency point. Some manufacturers' curves indicate the minimum allowable continuous stable flow (MCSF) limits for every pump (43). In the 1980s, the processing industry experienced a reduction in flow requirement as a result of business downturn and installation capacity downsizing. The pumping equipment, however, was generally not replaced by smaller pumps, but was forced to operate at reduced flow rates, often below allowable MCSF. This has resulted in increased failure rates and reduced pump component life.

There are two main reasons why a pump should not operate below its MCSF: (*1*) the radial force (radial thrust) is increased as a pump operates at reduced flow (44, 45). Depending on the specific speed of a pump, this radial force can be as much as 10 times greater near the shut off, as compared to that near the BEP; and (*2*) the low flow operation results in increased turbulence and internal flow separation from impeller blades. As a result, highly unstable axial and radical fluctuating forces take place.

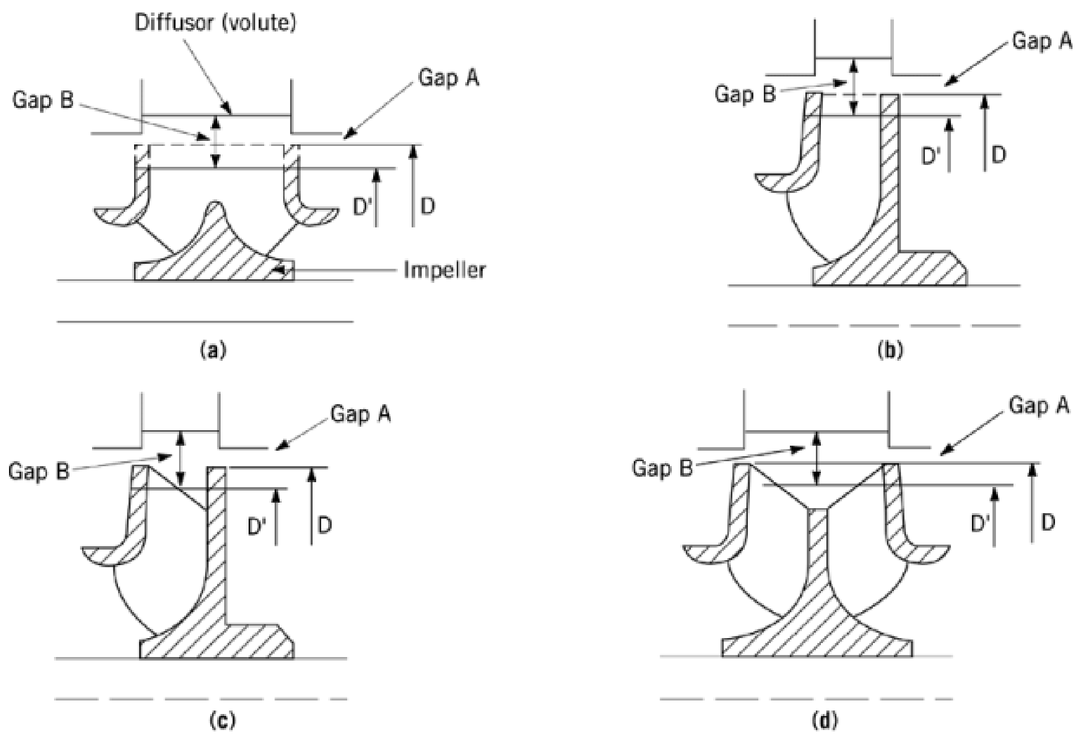

Fig. 19. Gaps A (between impeller shrouds and diffusor wall) and B (between impeller and diffusor vane tips) corrective option where D=diameter and D′ is diameter reduction: (**a**) overall diameter reduction; (**b**) vane diameter reduction; (**c**) angle-cut, single-suction impeller; and (**d**) angle-cut, double-suction impeller.(Courtesy of E. Nelson and *Pumps and Systems Magazine*.)

The relationship between force and the life of ball bearings is cubic (46), ie, doubling the force decreases bearing life eight times. In addition, the increased shaft deflections cause distortion at the faces of mechanical seals, causing leakage and failure. To reduce the radial thrust, some manufacturers have developed special casing designs refered to as circular volutes. These designs result in significantly reduced loads owing to a more even pressure distribution around the impeller periphery, as compared to conventional expanding volutes. The radial thrust reduction can be as high as 85%, depending on the design.

For chemical and petrochemical applications, a GasBarrier design, described in the section on dynamic sealing, is another way to improve reliability of pumps where dry running, or dirty liquid, pumpage is a problem.

6.3. Impeller–Diffusor Gaps

A tight gap between the impeller outer diameter and the diffusor inner diameter can result in high pressure pulsations, vibrations, noise, and vane breakage (47–51). By modifying gaps A and B, as shown in Figure 19 (52), it is possible to reduce significantly these adverse effects for problematic pumps in the field. This is especially true for high energy equipment, such as boiler feed pumps.

6.4. Cavitation

The subject of cavitation in pumps is of great importance. When the liquid static pressure is reduced below its vapor pressure, vaporization takes place. This may happen because (*1*) the main stream fluid velocity is too high, so that static pressure becomes lower than vapor pressure; (*2*) localized velocity increases and static pressure drops on account of vane curvature effect, especially near the inlets; (*3*) pressure drops across the valve or is reduced by friction in front of the pump; or (*4*) temperature increases, giving a corresponding vapor pressure increase.

Pump manufacturers publish data indicating a minimum required net positive suction head (NPSHR) to ensure that suction pressure is adequate and that pumps do not cavitate. A user must therefore calculate the available net positive suction head (NPSHA) by going through the system losses analysis. The NPSHA should always be greater than the NPSHR plus a safety margin. A recommended margin added to NPSHA is 1.5 m (5 ft) or 35% over NPSHR, whichever is greater (53). There are several good reasons for this. The NPSHR published in manufacturers' curves is based on pump tests. These tests, in accordance with HI Standards (9), defines NPSHR as equal to the value at which the pump actually loses 3% of its developed head. However, the incipient cavitation, ie, when first cavitation bubbles just begin to form, starts significantly before a 3% drop in head occurs. The value of the incipient NPSH is difficult to determine in commercial testing. Sophisticated research methods have been applied (54), but despite extensive research there is no single accepted method to predict incipient cavitation for different pump types or different specific speeds. Figure 20 illustrates the phenomenon of incipient cavitation as it progresses toward fully cavitating pump and eventual loss of performance at 3%; head drop (5%; flow drop for PD pumps) (55). Positive-displacement pump are less prone to cavitation damage, and can satisfactory pump even at significantly cavitating regime, when flow is reduced way below 5%;. The reason for that is that PD pumps require no special hydrodynamic design considerations of their internal passages, while centrifugal pump operation depends heavily on proper flow regimes. PD pump internal voids capture the entering fluid, regardless of the vortexes, irregularities, and other disturbances.

A further complication arises from the fact that the point of maximum damage to the pump by cavitational pitting happens somewhere between the incipient cavitation and the point of complete performance loss (56). Whereas it is difficult to predict incipient cavitation, it is even more difficult to determine the point of maximum damage. Many attempts have been made, but no method has been accepted. It has been reported that the damage is related to fluid velocity to the fifth or sixth power (57).

The vapor pressure may be dependent on the amount of the dissolved, not the entrained, air in the liquid. This issue is important to applications of cooling-water double-suction pumps (58, 59). Because of the unknowns, a safety margin is always recommended for use to minimize the effects of cavitation.

For existing installations where cavitation is known to be a problem but the cost of modification to the system is prohibitive, an effective way to resist cavitation is by upgrading pump metallurgy. For example, upgrading from cast-iron construction to Type-316 stainless steel can improve the impeller life by as much as 10 times. CA15 or CD4MCu alloys can improve this even further (32). Special designs of impellers and volutes can be made to reduce internal velocities and prevent or minimize cavitation damage. These designs usually require a computer analysis of three-dimensional effects of flow inside the impeller and casing (56). Sophisticated computational fluid dynamics methods (CFD) having complete viscous flow solutions are employed (60).

6.5. Reliability

There has been a significant rise in interest among pump users in the 1990s to improve equipment reliability and increase mean time between failures. Quantifiable solutions to such problems are being sought (61). Statistical databases have grown, improved by continuous contributions of both pump manufacturers and users. Users have also learned to compile and interpret these data. Moreover, sophisticated instrumentation has become available. Examples are vibration analysis and pump diagnostics.

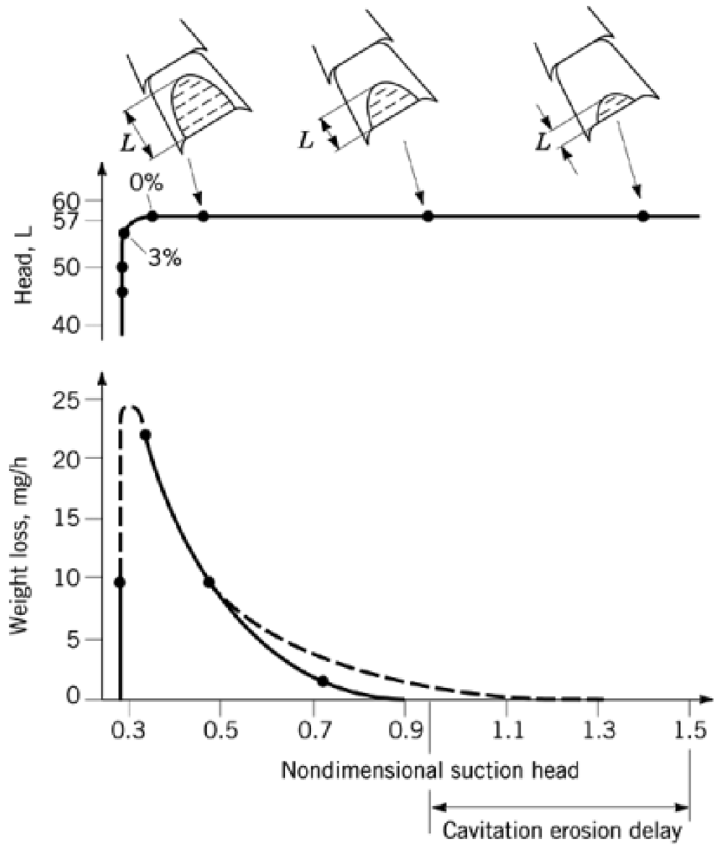

Fig. 20. Cavitation phenomenon in pumps showing cavitation bubble distribution and rate of weight loss as a function of cavitation coefficient at constant speed where (▬▬) represents actual and (–) theoretical measurements and L is bubble length growth. Courtesy of S. Florancic and *Pumps and Systems Magazine*.

Pump vibration measurements and analysis of the amplitude–frequency spectrum have led to the determination of possible equipment ailments. Examples of frequency relationships to particular pump vibration problems are available (62). For example, a one-time $(1\times)$ frequency corresponding to a running speed usually indicates the presence of rotor unbalance, a $2\times$ peak may mean a bent shaft, and a $\sim 0.4\times$ may be a symptom of an oil whip at the sleeve bearing.

Training and education of maintenance people (71), operators, and engineering personnel, with plant-wide involvement, can lead to better understanding of pumps and systems interactions, and equipment limitations. Early identification of problems, and corrective actions in response to an alarming trends, is much more economical then waiting for a problem to occur, with failures costing millions of dollars in replaced equipment, and plant downtime (72).

7. Economic Aspects

When selecting and purchasing pumping equipment, there are one-time capital investment as well as operating and maintenance costs to consider. The cost of pumps and auxiliaries, such as driver, coupling, baseplate, and sealing equipment, is capital investment, a one-time cost. Replacement of worn or broken parts, energy to operate, and service costs are maintenance and operating costs. A lower first time cost, ie, a less expensive pump, may not necessarily indicate a lower total cost. Usually, the opposite is true. Pump prices vary from one design to another and from one manufacturer to the next, but as a rule of thumb, an estimate can be made. A total cost of equipment ownership is becoming a popular concept, versus a more traditional approach of purchasing of a low-cost machine. Reliability, mean time between failures and lost production because of it become important factors in making purchasing decisions.

U.S. pump manufacturers include the following contributors to this work:

Company	Location
Liquiflo Equipment Company, division of Picut Industries	Garwood, NJ
Tri-Clover Inc.	Kenosha, Wis.
Goulds Pumps, Inc.	Seneca Falls, N.Y.
WEMCO Pump Co.	Sacramento, Calif.
Patterson Pump Co.	Toccoa, Ga.
Homa Pump Technology	Stamford, Conn.
KSB, Inc.	Richmond, Va.
Ingersoll-Dresser Pumps	Phillipsburg, N.J.
Johnston Pump	Brookshire, Tex.
Gast Manufacturing Corp.	Benton Harbor, Mich.
Fristam Pumps Inc.	Middleton, Wis.
Bornemann Pumps Inc.	Monroe, N.C.
Barnant Co.	Barrington, Ill.
ITT Jabsco	Costa Mesa, Calif.
American Lewa, Inc.	Holliston, Mass.
Spirax Sarco Inc.	Allentown, Pa.
Vanton Pump	Hillside, N.J.
Chempump	Warrington, Pa.
Vaughan Co., Inc.	Montesano, Wash.
A-Line Mfg. Co. Inc.	La Porte, Tex.
Durametallic Corp.	Kalamazoo, Mich.

Excellent sources of information are also available (9, 63).

7.1. Cost Example (applicable for certain centrifugal pumps)

If a single-stage overhung pump having a 200-mm (8-in.) impeller outside diameter (OD) and 316-ss metallurgy is of interest, then, an iron construction pump cost is approximately $2000 in 1999 dollars. Doubling that cost for the stainless steel gives $4000. Adding 50%; for motor, coupling, baseplate, and the seal makes the price $6000. Finally, applying a 70%; multiplier (30%; discount), a final selling price in the range of $4200 can be expected. (*Note*: numbers are for rough comparision only).

8. Nomenclature

Symbol	Definition	Units
Eff	pump efficiency	
$\mathrm{Eff}_{\mathrm{motor}}$	motor efficiency	
g	gravitation constant	9.81 m/s^2 (32.174 ft/s^2)
H	total developed pump head	m
H_{d}	total discharge head	m
H_{s}	total suction head	m
H_{dynamic}	velocity head ($\frac{V^2}{2\,g}$)	m
$H_{\mathrm{loss,A--B}}$	hydraulic losses from A to B	m
H_{static}	static head	m
H_{vapor}	vapor head	m
$\mathrm{HP}_{\mathrm{fluid}}$	net power delivered to fluid being discharged	kW (hp)
$\mathrm{HP}_{\mathrm{loss}}$	power losses inside the pump	kW (hp)
N	rotational speed	rpm
NPSH	net positive suction head	m
NPSHA	net positive suction head available	m
NPSHR	net positive suction head required	m
NS	specific speed in U.S. notation	
nS	specific speed in metric notation	
nSS	suction specific speed in metric notation	
OD	outside diameter of impeller	mm
γ	density conversion (SG/2.31)	psi
Ω	rotational speed	rad/s
Ω_{s}	universal dimensionless specific speed	
$p_{\mathrm{d,g}}$	discharge static gauge pressure	N/m^2
$p_{\mathrm{s,g}}$	suction static gauge pressure	N/m^2
PF	power factor of electric motor	
Q	capacity	m^3/s
S	suction specific speed in U.S. notation	
SG	specific gravity	
T	torque	N·m
V_{d}	discharge velocity	m/s
V_{s}	suction velocity	m/s
Z_{d}	discharge gauge elevation (+) or submergence (−) from pump centerline	m
Z_{s}	suction gauge elevation (+) or submergence (−) from pump centerline	m

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