CHAPTER 5.3

Pumps

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A pump is a machine that lifts, pressurizes, and moves liquids or solid–liquid slurries from place to place. The mineral processing industry uses pumps in all phases of wet processing, from grinding to tailings disposal.

TERMINOLOGY AND NOMENCLATURE

Common terms in the pumps industry are described in this section.

- **Measurement of performance.** The amount of useful work that any fluid-transport device performs is the product of the mass rate of fluid flow through the device and the total pressure differential measured immediately before and after the device, usually expressed in the height of a fluid column equivalent under adiabatic conditions. The first of these quantities is referred to as capacity, the second as head.
- **Capacity.** Capacity, Q, is expressed in cubic meters per hour (m^3/h); in U.S. customary units, it is expressed in gallons per minute (gpm). Since all these are volume units, the specific gravity or specific weight of the liquid must be used for conversion to mass rate of flow.
- **Head.** Head, H, is a term for pressure, expressed as the height of a column of liquid. Other expressions for head include the following:
  - **Friction head.** Friction head, H_f, is the pressure required to overcome the resistance to flow in the pipe and fittings.
  - **Net positive suction head.** Net positive suction head, NPSH, is the difference between the liquid pressure at the pump's suction and the vapor pressure of the fluid being pumped. The available net positive suction head, NPSHA, is a measure of how close the fluid at a given point is to flashing, and the inception of cavitation, as described below. The required net positive suction head, NPSHR, is the head value at the inlet of the pump required to keep the fluid from cavitating. In a positive displacement (PD) pump, the NPSHA should be large enough to open the suction valve, overcome the friction losses within the pump liquid end, and overcome the liquid acceleration head.
  - **Total dynamic head.** The total dynamic head, H, of a pump is the total discharge head, h_d, minus the total suction head, h_s.
  - **Total suction head.** Total suction head is the reading h_{gs} of a gauge at the suction flange of a pump, corrected to the pump centerline, plus the barometer reading of atmospheric pressure, atm, and the velocity head, h_{vs}, at the point of gauge attachment:

    $$h_s = h_{gs} + \text{atm} + h_{vs}$$

    If the gauge pressure at the suction flange is less than atmospheric, requiring use of a vacuum gauge, this reading is used for h_{gs} in the preceding equation with a negative sign.
  - **Static suction head.** The static suction head, h_s, is the vertical distance from the free surface of the liquid source to the pump centerline plus the absolute pressure at the liquid surface.
  - **Total discharge head.** The total discharge head, h_d, is the reading, h_{gd}, of a gauge at the discharge flange of a pump (corrected to the pump centerline, or on vertical pumps, to the eye of the suction impeller), plus the barometer reading and the velocity head, h_{vd}, at the point of gauge attachment.
  - **Velocity head.** Velocity head, h_v, is the vertical distance through which a body must fall to acquire the velocity, V.

    $$h_v = V^2/2g$$

    where V is the flow velocity in m/s, and g is the acceleration due to gravity in a pumping system; the velocity head derives from the flow of the fluid at the velocity, V.
- **Speed.** Pump speed, N, is denoted in revolutions per minute, or rpm.
- **Specific speed.** Specific speed, N_S, of a pump is the speed at which a geometrically similar impeller would need to

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operate to deliver a fixed volume flow rate at a fixed head. The specific speed is calculated using metric units as

$$N_S = (N\sqrt{Q})/(H^{0.75})$$

where N is rotational speed in rpm, Q is capacity in m^3/min, and H is head in m.

Specific speed is usually thought of as a dimensionless number. In fact, specific speeds calculated with this formula are not dimensionless, but the dimensions are not meaningful, so they are not used. It is thus critically important that the units used are reported along with the pump specific speed value. A dimensionless expression for specific speed can be created by multiplying the pump head by the gravitational constant and measuring the shaft rotation in radians, thus:

$$\Omega_S = (N_{\text{rad}}\sqrt{Q})/(g\,H^{0.75})$$

where N_{rad} is rotational speed in rad/s, Q is capacity in m^3/s, g is the acceleration due to gravity in m/s^2, and H is head in m.

The dimensionless specific speed, Ω_S, is related to the specific speeds calculated with metric or U.S. units by the following expressions:

$$\Omega_S = N_S\,(\text{U.S.})/2733.016$$

where N_S is calculated with N as rotational speed in rpm, Q as capacity in gpm, and H as head in ft.

Similarly,

$$\Omega_S = N_S\,(\text{metric})/51.64$$

In general, pumps with low specific speed have a low capacity and high head, while pumps with high specific speed have higher capacity and lower head.

- **Velocity.** The linear velocity, V, is found from the ratio of the quantity flowing through a given area in a given time to that area, thus

  $$V = Q/A$$

  where V is the average flow velocity, Q is the flow quantity, and A is the cross-sectional area of the pipe or channel. For circular pipe, this relationship in metric units is

  $$V = 3.54\,Q/d^2$$

  where V is the average flow velocity in m/s, Q is the flow quantity in m^3/h, and d is the inside pipe diameter in cm.
- **Viscosity.** The existence of internal friction or the internal resistance to relative motion of the fluid particles is called viscosity. The viscosity of liquids usually decreases with rising temperature. Viscous liquids tend to increase the power required by a pump; to reduce pump efficiency, head, and capacity; and to increase friction in pipelines.

  Care must be taken to distinguish between dynamic viscosity and kinematic viscosity. *Dynamic viscosity* is the resistance of a fluid to shearing flows, where adjacent layers move parallel to each other with different speeds. The Greek letter μ is commonly used by engineers and some physicists to denote dynamic viscosity, while η is used by chemists and other physicists. Dynamic viscosity has units of pascal-second, or Pa·s. *Kinematic viscosity*, also called momentum diffusivity, is the ratio of the dynamic viscosity, μ, to the density of the fluid, ρ. It is usually denoted by the Greek letter ν and is measured in units of square meters per second, or m^2/s.
- **Work performed in pumping.** Liquid to flow occurs when work is expended. A pump may raise the liquid to a higher elevation, force it into a vessel at higher pressure, provide the head to overcome pipe friction, or perform any combination of these. Regardless of the service required of a pump, all energy imparted to the liquid in performing this service must be accounted for; consistent units for all quantities must be employed in arriving at the work or power performed.

  When analyzing the performance of a pump, it is customary to calculate its power output, which is the product of the total dynamic head and the mass of liquid pumped in a given time. In metric units, power is expressed in kilowatts; horsepower is the conventional unit used in the United States.

  In metric units,

  $$kW = HQ\gamma/3.670 \times 10^5$$

  where kW is the pump power output in kW, H is the total dynamic head in N·m/kg (column of liquid), Q is the capacity or flow in m^3/h, and γ is the liquid density in kg/m^3.

  When the total dynamic head H is expressed in pascals, then

  $$kW = HQ/3.599 \times 10^6$$
- **Efficiency.** The power input to a pump is greater than the power output because of internal losses resulting from friction, leakage, and so on. The efficiency of a pump is therefore defined as

  pump efficiency = (power output)/(power input)

  In the United States, the power input is often described as the *brake horsepower* (BHP), an anachronistic term that refers to the power output at the shaft of the driving unit, with no accessories attached. A similar term, *brake kW*, is sometimes employed when using the metric system.

The best efficiency point (BEP) is the maximum possible efficiency and is usually shown as an efficiency curve on the head versus capacity chart for a given pump.

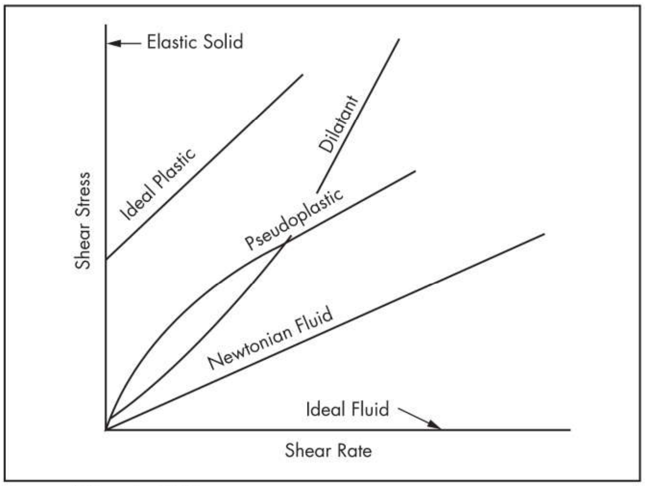
HYDRAULICS FOR PUMPING

Properties of Fluids

Design and selection of pumps requires a thorough knowledge of the fluid to be pumped, including specific weight and volume, specific gravity, vapor pressure, and kinematic viscosity. Fluids may be categorized on their shear behavior, as shown in Figure 1. In Newtonian fluids, shear stress versus shear rate is characteristically linear. Thixotropic or pseudoplastic fluids exhibit a shear-thinning characteristic where the shear stress decreases with increasing shear rates. Dilatant fluids exhibit a shear-thickening characteristic where the shear stress increases with increasing shear rates.

Bernoulli's Equation for Energy in Incompressible Fluids

The law of conservation of energy states that energy can neither be created nor destroyed, and that all forms of energy



Source: Link et al. 1985
Figure 1 Shear stress–shear rate relationship for various types of viscous behavior

are equivalent. Energy considerations in closed pipe systems are analyzed readily by the following general energy equation, known as Bernoulli's equation:

$$P_1 + V_1^2/2g + Z_1 + H_a - H_f - H_e = P_2 + V_2^2/2g + Z_2$$

The left-hand side of the equation describes the energy at the pump inlet. The first term, P_1, is the pressure head. The second term, $V_1^2/2g$, is the velocity head and accounts for the kinetic energy of the moving fluid. The third term, Z_1, is the elevation head with respect to a defined reference point. The pump head, H_a, is the energy added by the pump. The fifth term is the energy lost to fluid friction, H_f; and the sixth term is the energy extracted, H_e. The terms on the right-hand side of the equation represent similar quantities at the pump discharge. All quantities are expressed in units of length.

Cavitation

Cavitation occurs in a flowing liquid when there is a decrease and subsequent increase in local pressure. The pressure decrease results in localized vaporization and the formation of bubbles. When the pressure recovers, the bubbles implode. Cavitation occurs only if the local pressure declines below the saturated vapor pressure of the liquid and subsequently recovers to above the vapor pressure. If the recovery pressure is not above the vapor pressure, then "flashing" occurs. In pipe systems, cavitation is usually the result of an increase in the kinetic energy through a constricted area, or an increase in the elevation of the pipeline.

The generation, growth, and collapse of cavitation bubbles result in very high-energy densities, and in high local temperatures and pressures at the surfaces of the bubbles. These conditions exist only for a very short time, and the overall liquid medium remains at ambient conditions. Uncontrolled cavitation is damaging and can cause noise, vibration, and a loss of efficiency. Highly localized cavitation can cause erosion and pitting, and can dramatically shorten a pump's lifetime. After a surface is initially affected by cavitation, it tends to erode faster. The cavitation pits increase the turbulence of the fluid flow and create crevices that act as nucleation sites for additional cavitation bubbles. The pits also increase the components' surface area and leave behind residual stresses. This makes the surface more prone to stress corrosion.

Cavitation in pumps may occur in two different forms. *Suction cavitation* occurs when the pump suction is under a low-pressure/high-vacuum condition, and the liquid turns into a vapor at the eye of the pump impeller. When this vapor is carried to the discharge side of the pump, it is no longer under vacuum and is compressed to a liquid state by the discharge pressure. This imploding action can be violent and damaging. It may remove large chunks of very small bits of material, depending on local conditions.

Discharge cavitation occurs when the pump discharge pressure is extremely high, as may be the case in a pump that is running at less than 10% of its BEP. The high discharge pressure causes most of the fluid to circulate inside the pump instead of flowing out the discharge. As the liquid flows around the impeller, it passes through the small clearance between the impeller and the pump housing at very high velocity. This flow velocity generates a vacuum to the housing wall, vaporizing the liquid. A pump that has been operating under these conditions shows premature wear of the impeller vane tips and the pump housing, and the high pressures generated can cause premature failure of the pump's mechanical seal and bearings.

All pumps require well-developed inlet flow to operate efficiently. When poorly developed flow enters a pump impeller, it strikes the vanes and is unable to follow the impeller passage. The liquid separates from the vanes, causing cavitation and vibration resulting from turbulence and poor filling of the impeller. For a well-developed flow pattern, a straight pipe run of about 10 pipe diameters is recommended upstream of the pump inlet flange. Unfortunately, space and equipment layout constraints may not allow this distance. Instead, it is common to use an elbow close-coupled to the pump suction, which creates a poorly developed flow pattern at the pump suction. Cavitation may be avoided by increasing suction pressure, decreasing liquid temperature, throttling back on the discharge valve to decrease flow rate, or venting gases off the pump casing.

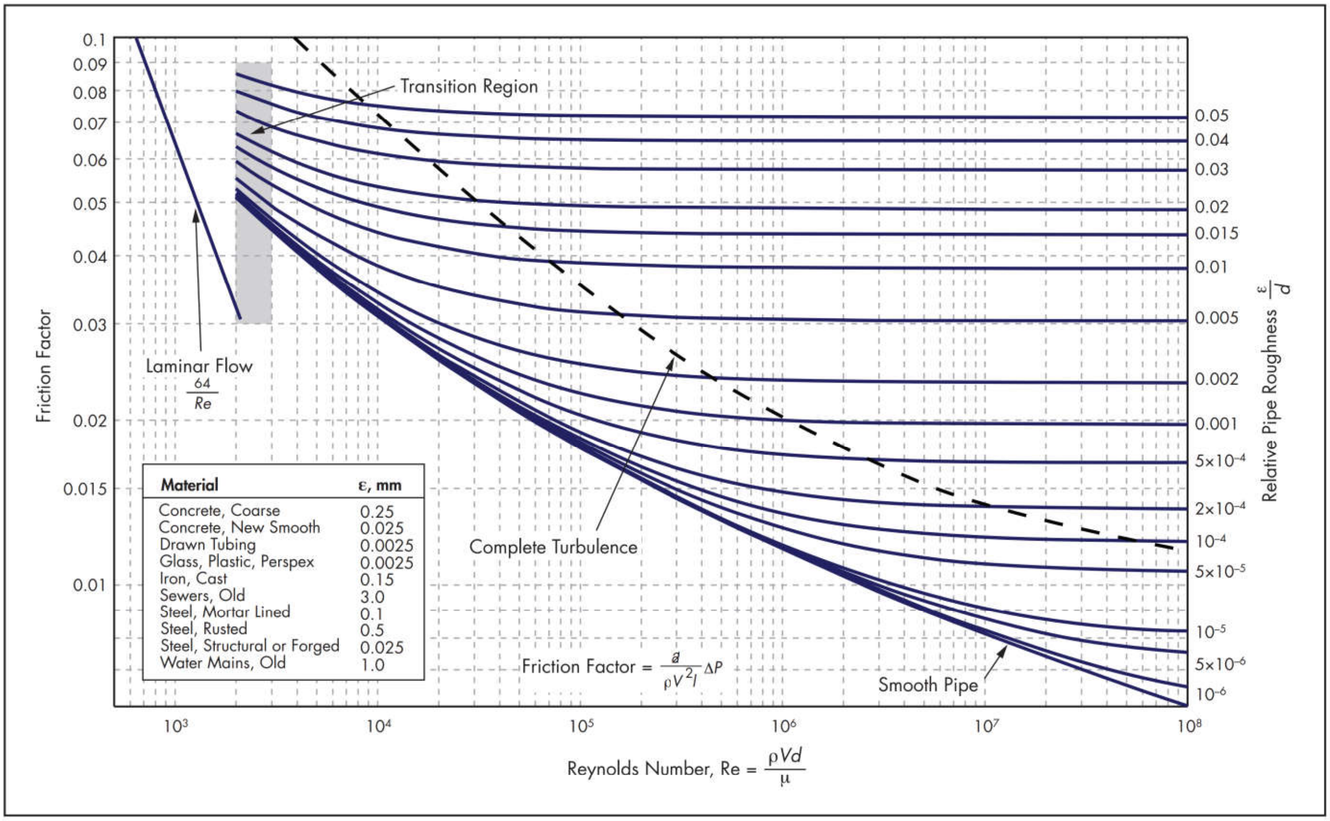
Flow in Pipes

When a fluid is flowing in a pipe, the shear stress on the fluid is highest at the walls of the pipe, which is stationary, and is lowest at the center of the pipe. The velocity profile across the pipe cross section is a function of the fluid type.

Liquids flowing in filled pipes at low velocities tend to follow straight lines. This is called laminar flow. As liquid velocities increase, the flow regime eventually becomes turbulent. At intermediate velocities, fluids flow in a transition zone. In the transition zone, the fluid flow is unstable and may oscillate unpredictably between laminar and turbulent flow. In most pumping systems, the flow is either transitional or turbulent.

The nature of flow in a pipe depends on the pipe diameter, the density and viscosity of the fluid, and the flow velocity. These quantities can be combined into a single, dimensionless number called the Reynolds number, N_{Re}, which is defined as

$$N_{Re} = DV/\nu$$



Source: Beck and Collins 2008

Figure 2 Moody diagram

where D is the inside pipe diameter, V is the fluid velocity, and ν is the kinematic viscosity. The Reynolds number may also be calculated as

$$N_{\mathrm{Re}} = \rho DV/\mu$$

where ρ is the specific gravity and μ is the kinematic viscosity of the fluid. While the Reynolds number is dimensionless, care must be taken to use consistent units in its computation and use.

The transition from laminar to turbulent flow occurs at a Reynolds number of about 2,000. This is called the critical Reynolds number and is important in computing fluid friction losses in pipes. In fact, critical Reynolds numbers may vary substantially with different fluids, but engineering practice assumes that laminar flow occurs below a Reynolds number of 2,000, turbulent flow occurs above 4,000, and a transition zone exists between these values.

Friction Losses

Liquid flow in pipes is always accompanied by friction losses, which result from internal friction or viscous forces of the liquid, and from the surface roughness of the pipe. These losses are indicated by a reduction in head in the direction of liquid flow. For a given length of pipe, L, this pressure drop can be measured, or predicted with reasonable accuracy using the Darcy–Weisbach equation:

$$H_f = f\,LV^{2}/2gD$$

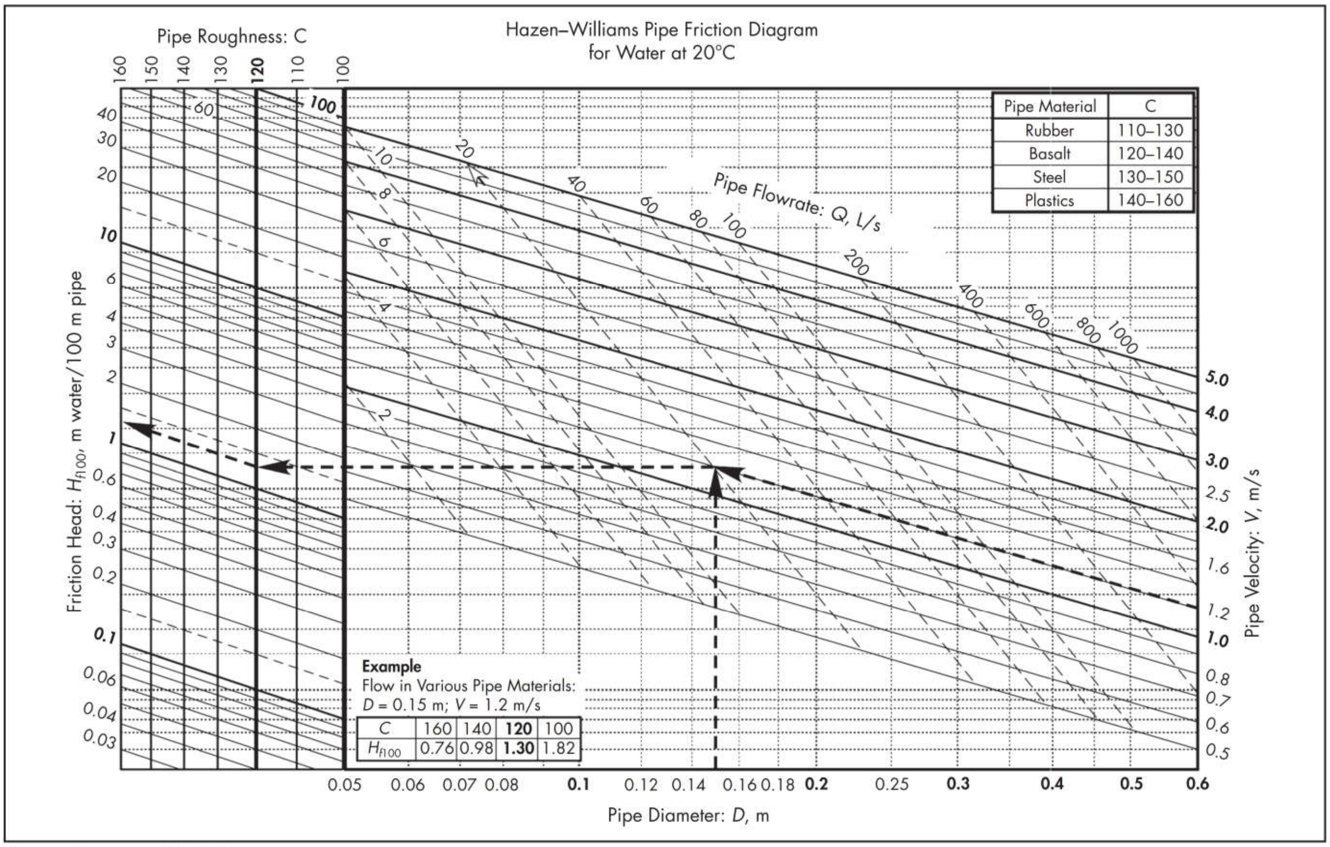
where H_f is the pressure drop in m, f is the friction factor, V is the flow velocity in m/s, g is the gravitational constant in m/s, and D is the inside diameter of the pipe in m.

The friction head loss, H_f, is often expressed in meters or feet of loss for a fixed length of pipe. The Darcy friction factor, f, is determined experimentally and varies with the Reynolds number. In laminar flow, the friction head loss primarily results from viscous forces within the liquid. When the Reynolds number is below 2,000, the friction factor may be estimated by

$$f = 64/N_{\mathrm{Re}}$$

In turbulent flow, the roughness of the interior pipe wall must be considered in addition to the viscosity of the liquid. The relative roughness, ε/D, is found by dividing the absolute surface roughness of the pipe, ε, by the diameter of the pipe, D. The surface roughness may be thought of as the average height of projections on the pipe wall. With consistent units, the relative roughness is a dimensionless number. The relationship among f, N_{Re}, and ε/D is often plotted on a Moody diagram, as shown in Figure 2, so that values of f may be determined graphically.

Churchill (1977) developed an equation that is too cumbersome for occasional calculation of single values of f, but it is useful when plotting the whole Moody diagram, including laminar, transition, and turbulent flow regions, say, from $N_{\mathrm{Re}} = 10^3$ to 10^8 and from $\varepsilon/D = 0$ to 0.01, yielding f-values from $f = 0.01$ to 0.04. The Churchill equation is especially useful in spreadsheets and computer programs:



Source: Grzina et al. 2002

Figure 3 Hazen–Williams diagram

$$f = 8\left\{\left(\frac{8}{N_{Re}}\right)^{12} + \left[\left(\frac{37{,}530}{N_{Re}}\right)^{16} + \left\{-2.457 \cdot \ln\left[\left(\frac{7}{N_{Re}}\right)^{0.9} + 0.27\frac{\varepsilon}{D}\right]\right\}^{16}\right]^{-1.5}\right\}^{1/12}$$

To use the Moody diagram, first compute the Reynolds number and the relative roughness for the fluid system. Next, locate the appropriate relative roughness value along the right side of the diagram. Then move to the left, parallel to the group of curved lines. Continue to move left, parallel to the nearest curved line, until the vertical line passing through the appropriate Reynolds number is reached. Then move horizontally to the left side of the diagram and read the friction factor.

Values for the absolute surface roughness for various pipe materials are also included in the Moody diagram, for convenience. These values are for new pipe. Pipe roughness generally increases with age. Changes in pipe wall roughness adversely affect both pipe diameter and friction factor.

The widely used Williams and Hazen (1920) formula provides an alternative to the use of the Moody diagram:

$$V = 0.849CR^{0.63}S^{0.54}$$

where V is the flow velocity in the pipe in m/s, C is the friction factor resulting from roughness only, R is the hydraulic radius (area/perimeter) of the pipe, and S is the friction head loss in m/m. The advantage of the Hazen–Williams equation is that the coefficient C is not a function of the Reynolds number, but the disadvantage is that it is only valid for water. Also, it does not account for the temperature or viscosity of the water. A graphic form of the Hazen–Williams formula is also available and is shown in Figure 3.

Minor head losses are those caused by local disturbances in the liquid system. Disturbances can be caused by entrance and discharge conditions; changes in pipe diameter; pipe fittings, such as valves, elbows, and tees; and minor irregularities in the interior pipe wall. Although these losses are termed *minor losses*, they may become a significant portion of the total head loss in short lines or in in-plant lines where numerous fittings are used. Minor losses are frequently reported as equivalent to a length of straight pipe, or as a constant to be applied to the velocity head. If the latter method is used, the minor head losses, H_{ml}, are given by the following equation:

$$H_{ml} = KV^2/2g$$

where K is an empirically determined constant, and V is the linear flow velocity. Table 1 shows loss coefficients for various pipe fittings.

As liquid flows from a large reservoir into a pipe, turbulence develops at the entrance to the pipe, causing a decrease in pressure head known as an entrance loss. The effect of this turbulence is to restrict the opening size. The degree of restriction depends on the shape of the pipe opening. Figure 4 shows typical entrance configurations and their loss coefficients, K_e.

Table 1 Head loss coefficients for various pipe fittings

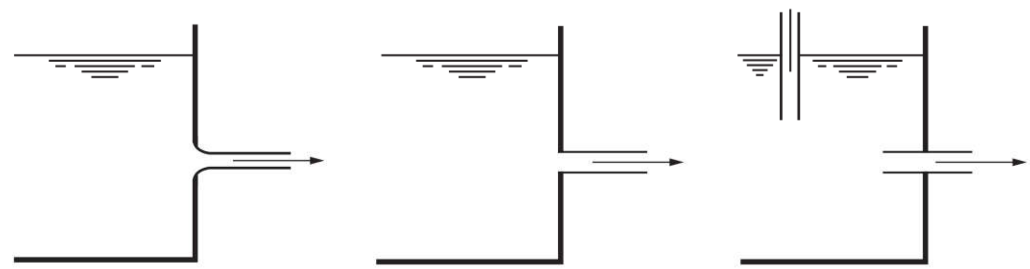
Fitting	K	L/D
Globe valve, wide open	10	350
Angle valve, wide open	5	175
Close return bend	2.2	42
T, through side outlet	1.8	67
Short-radius elbow	0.9	32
Medium-radius elbow	0.75	27
Long-radius elbow	0.60	20
45° elbow	0.42	15
Gate valve, open	0.19	7

Source: Scott and Hays 1985

Table 2 Contraction head loss coefficients for various pipe diameter ratios

D_2/D_1	K_c
0.1	0.45
0.2	0.42
0.3	0.39
0.4	0.36
0.5	0.33
0.6	0.28
0.7	0.22
0.8	0.15
0.9	0.06

Source: Scott and Hays 1985



Source: Foust et al. 1964
Figure 4 Entrance losses

As liquids flow from an open pipe into a reservoir, the velocity is dissipated, and the discharge head loss coefficient is usually considered to be 1.0. However, the discharge loss occurs outside the pipe system and thus does not affect the pressure at the pipe end. Sudden contractions in pipe diameters result in contraction head losses similar to those associated with entrance conditions. Table 2 shows loss coefficients for various pipe diameter ratios. To reduce these losses, a conical or smooth-tapered section should be inserted between the two pipe diameters. When the total angle lies between 20° and 40°, the loss is minimized and the coefficient for contraction, K_c, equals about 0.1.

Head losses from a sudden enlargement in pipe diameter, H_{mlc}, are found as follows:

$$H_{mlc} = (V_1 - V_2)^2/2g$$

where V_1 and V_2 are the respective linear flow velocities before and after the enlargement.

The analysis of friction losses in slurry pumping is discussed later in the "Slurry Pumping" section.

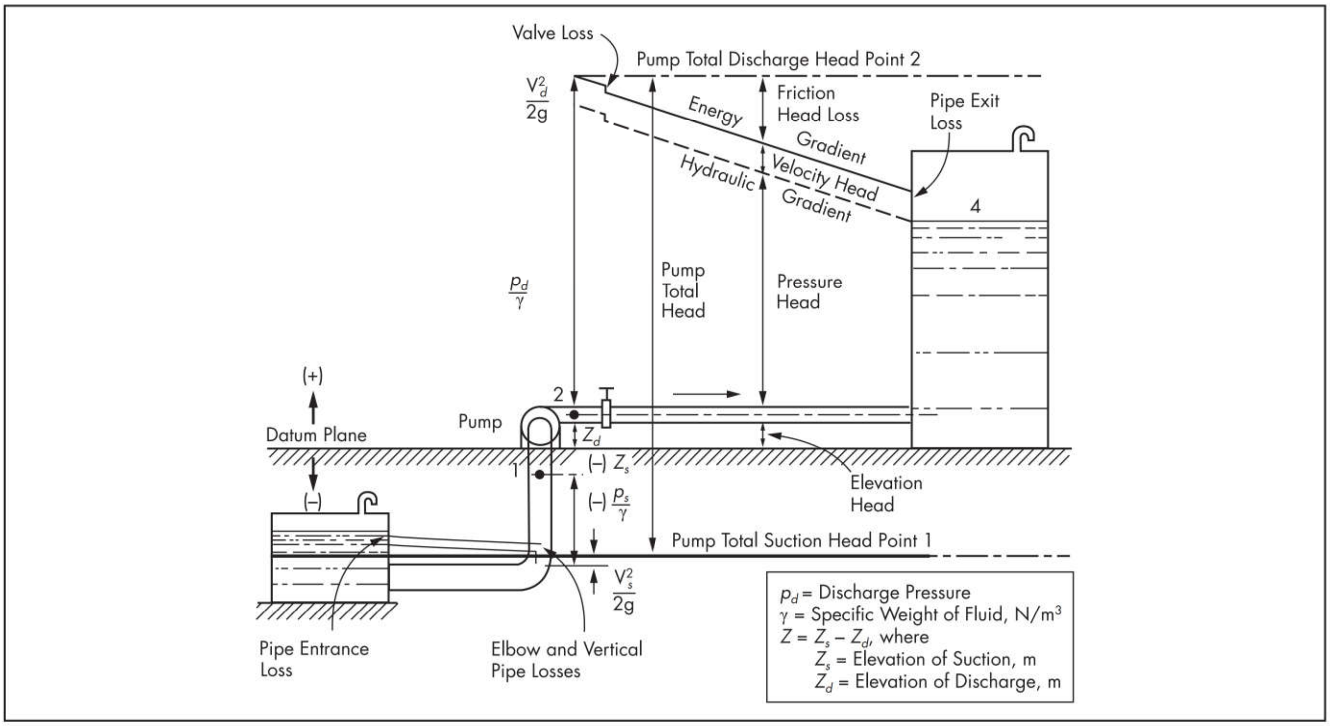
Hydraulic Gradient Profile

The total energy or head at any point in a pumping system can be calculated from Bernoulli's equation. If a convenient reference elevation or datum plane is chosen and the total energy is calculated at locations along the system, the calculated values may be plotted to scale to show the energy gradient of the system. Figure 5 shows the variation in total charge energy, H, measured in meters from the suction liquid surface to the discharge liquid surface. The dotted line through the sum of the pressure and elevation heads in Figure 5 is called the hydraulic gradient. The velocity head is the difference between the energy gradient and the hydraulic gradient lines. The total head for the pump is the difference between total energy at the pump discharge, point 2, and the pump suction, point 1.

System-Head Curve

A pumping system may include pipes, valves, fittings, vessels, meters, process equipment, and other components through which flow must pass. When a system is analyzed for pump selection, the flow resistance of the liquid through all these components must be calculated. That resistance increases with flow at a rate approximately equal to the square of the flow through the system. Besides overcoming flow resistance, additional head may be required to raise the liquid from the pump inlet point to a higher level at the discharge. In addition, the atmospheric pressure at the discharge point will usually be different than the pressure at the inlet, and this must be accounted for. Required heads that result from a change in elevation or a change in atmospheric pressure are called *fixed system heads* or *static heads* because they do not vary with rate of flow. Fixed system heads can be positive or negative.

A system-head curve is a plot of total system resistance, variable plus fixed, for various flow rates. It has many uses in centrifugal pump applications. Because centrifugal pumps are rated in units of length, the curve should show head in the same way, and not in pressure units. System-head curves usually show flow in volume per unit time.



Adapted from Karassik et al. 2001

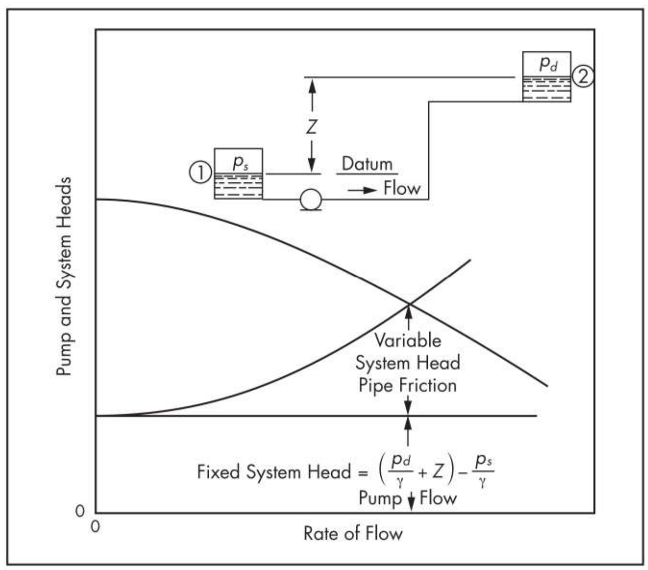
Figure 5 Energy and hydraulic gradients

When the system head is required for several flows or when the pump flow is to be determined, a system-head curve is constructed using the following procedure:

1. Define the pumping system and its length.
2. Calculate (or measure) the fixed system head, which is the net change in total energy from the beginning to the end of the system due to elevation or pressure head differences. An increase in head in the direction of flow is a positive quantity.
3. Calculate, for several flow rates, the variable system total head loss through all piping, valves, fittings, and equipment in the system.

Figure 6 illustrates construction of a system total-head curve, with a pumping system that starts at point 1 and ends at point 2. In this figure, the system-head curve shows head increasing with increasing flow rate. The fixed system head is the net change in total energy and does not change with flow, because the pressures and liquid levels do not vary with flow. The total head is p_s/γ at point 1 and $p_d/\gamma + Z$ at point 2, where p_s is the suction pressure, γ is the specific weight of fluid in N/m^3, p_d is the discharge pressure, and $Z = Z_s - Z_d$, where Z_s is the elevation of suction in m, and Z_d is the elevation of discharge in m. The pipe, valves, and fittings provide the variable system head. The curve of total system head versus flow is plotted by adding the fixed head and variable heads for several flow rates.

The flow produced by a centrifugal pump varies with the system head, while the flow of a PD pump is independent of the system head. In Figure 6, the head-capacity curve of a centrifugal pump is also shown on the system total-head curve. It shows a decreasing head with increasing flow. The flow of the pump in this system is indicated by the point at which the pump total head and system total head intersect. This intersection should be at or near the pump's best efficiency.

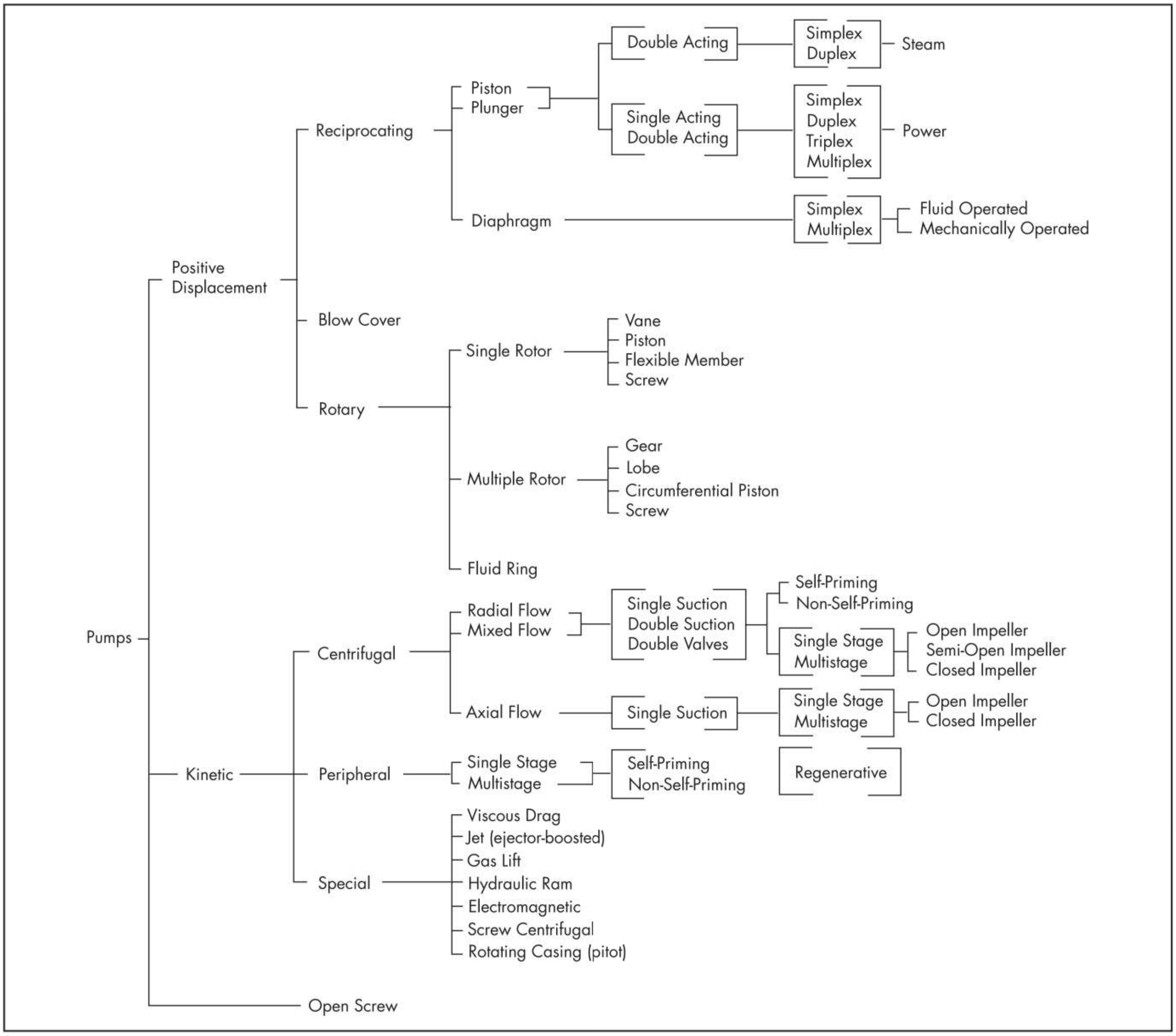


Source: Karassik et al. 2001

Figure 6 Construction of the system total-head curve

TYPES OF PUMPS

The two major pump classes are kinetic and positive displacement. Each of these classes is further subdivided based on operating principle and construction features. The classification system for pumps is shown in Figure 7.



Source: Hydraulic Institute 1975
Figure 7 Classification of pumps

Kinetic (Centrifugal) Pumps

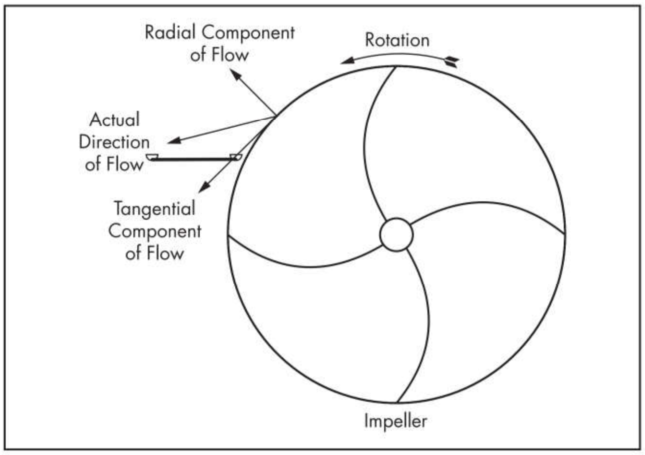
Kinetic pumps impart kinetic energy to the fluid to be pumped and then convert that kinetic energy into an increased pressure, as described by Bernoulli's principle. The most common type of kinetic pump, by far, is the centrifugal pump. As its name implies, this pump converts energy supplied by a prime mover into pressure energy and velocity energy by imparting centrifugal force to the liquid entering the eye of a rotating impeller, as shown in Figure 8. Liquid entering the eye is guided to the outer periphery of the impeller through several vane passages. As the liquid leaves the impeller, it has two components of flow. One component is in the radial direction, traveling through the vane; the other is tangential, resulting from the direction of rotation of the impeller.

The pump casing or volute that surrounds the impeller guides the liquid exiting the impeller to the discharge nozzle. The path between the impeller and casing gradually increases from the closest clearance (called the volute tongue or cutwater) to the discharge nozzle. This is to maintain a relatively constant velocity as more liquid is accumulated from the impeller along the path. Prior to reaching the pump discharge flange, the casing flares out in the diffuser area, as shown in Figure 9. This results in a reduction in velocity with an accompanying increase in pressure energy.

Casing Design

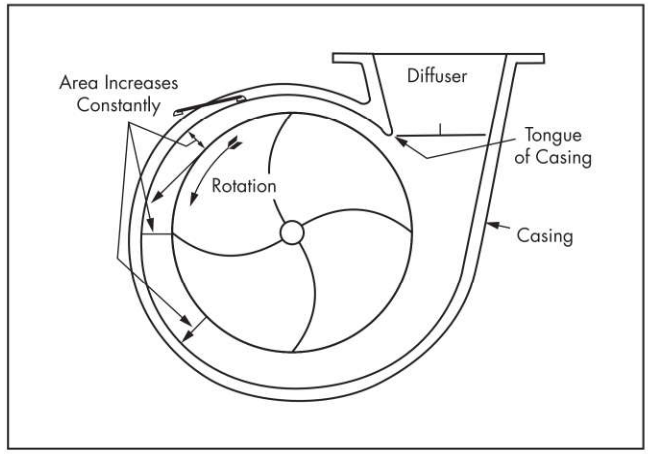
The three basic casing design types used in a centrifugal pump are the true volute, near or semi-volute, and circular volute, as illustrated in Figure 10.

The true volute, Figure 10A, has the highest efficiency because of its tight impeller-to-cutwater clearance. The cutwater is the V-shaped diverter/flow splitter between volute and discharge. This tight clearance increases efficiency by minimizing recirculating flow at the BEP. However, at flows below



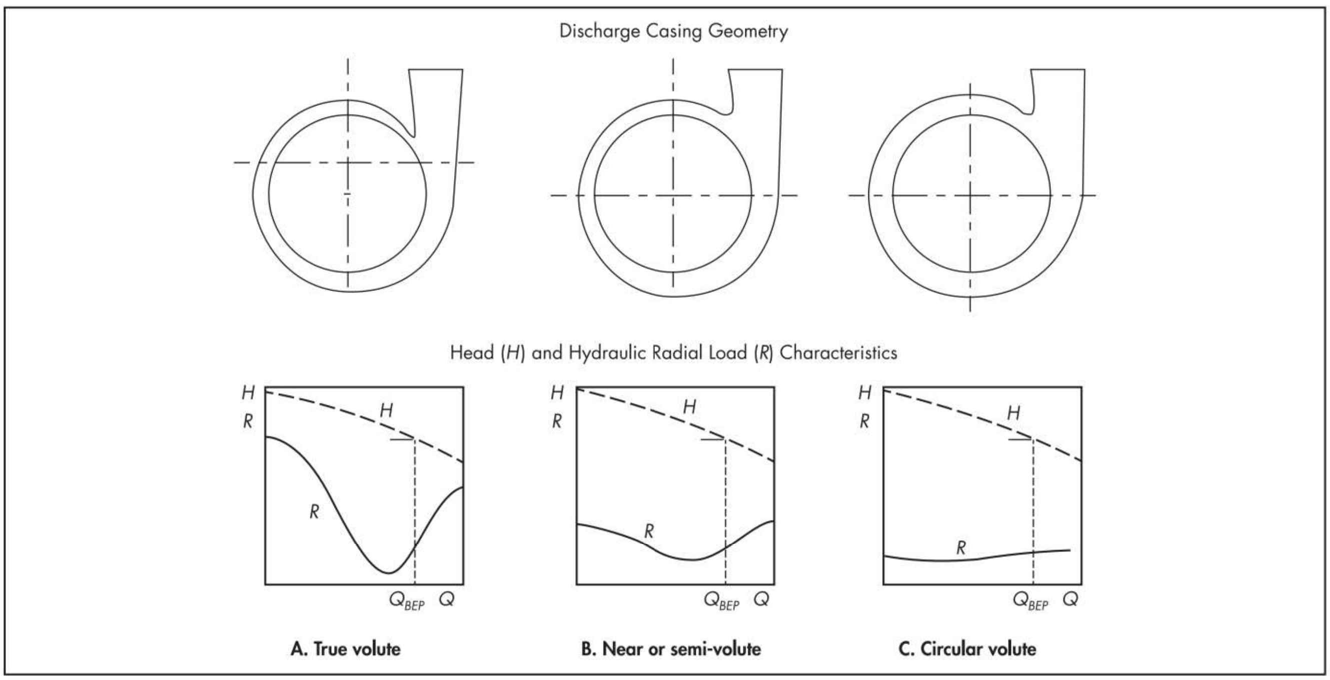
Source: Link et al. 1985

Figure 8 Centrifugal pump force diagram



Source: Link et al. 1985

Figure 9 Centrifugal pump terminology



Source: Roudnev and Angle 1999

Figure 10 Slurry pump casing types

the BEP, the increased recirculating flow has a high velocity through the small impeller to the cutwater area. This results in high wear at low flows for this design in the area at and just beyond the cutwater. This design also has the highest hydraulic radial loading at flows other than the BEP.

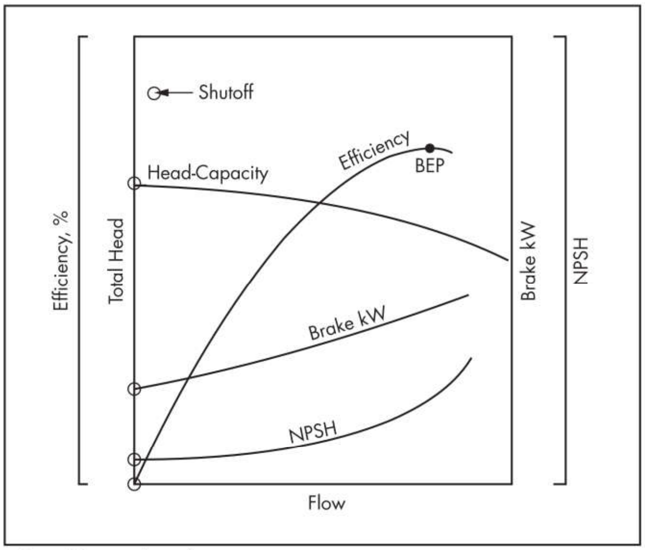
The near volute, Figure 10B, is similar to the true volute, but the impeller-to-cutwater clearance is increased to reduce velocity at the cutwater area for lower flows. This design results in significantly reduced wear in this area at low flows. This design also has significantly lower hydraulic radial force at conditions away from the BEP. For most slurry applications, this design offers a good balance of wear life and efficiency.

The circular volute, Figure 10C, has uniform area between the impeller and casing all around the volute. This volute design has the lowest hydraulic radial force at conditions off the BEP. This makes it ideal for high-head, low-flow applications.

Normal recommended operating conditions for the three volute types are 80%–120% of BEP flow for the true volute, 60%–110% of BEP flow for the near volute, and 40%–100% of BEP flow for the circular volute. For severe duty, including heavy slurries, the ideal selection would be a pump with an operating point in the middle of one of the previous ranges, to minimize wear on the volute. For light slurries, the preceding ranges can be widened.

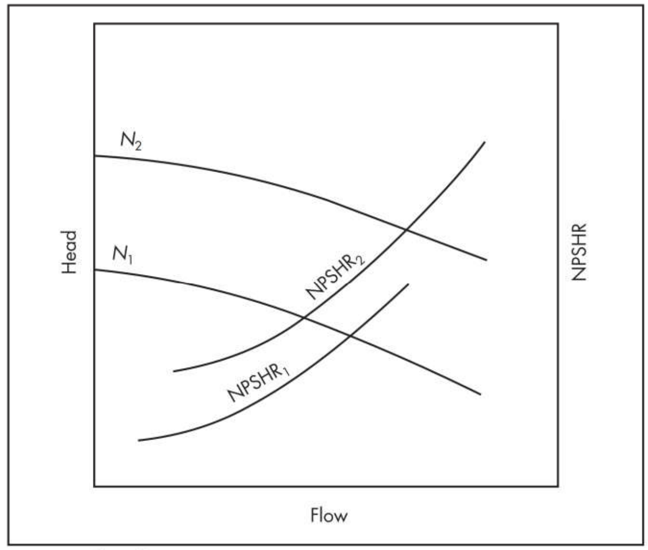
Performance

Performance of a centrifugal pump is conveniently characterized in a set of performance curves. These curves consist of plots of total dynamic head *TDH*, total system head *TSH*,



Adapted from Link et al. 1985

Figure 11 Centrifugal pump characteristic curves



Source: Link et al. 1985

Figure 12 Relationship of *Q*, *N*, and NPSHR

or just head H, along with power and efficiency, all plotted against capacity Q at constant speed N. For a centrifugal pump, the total dynamic head is defined as

$$H = H_d - H_s + (V_d^2 - V_s^2)/2g$$

where H_d is the discharge head as measured at the discharge nozzle and referred to the pump shaft centerline, expressed in meters; H_s is the suction head expressed in meters as measured at the suction nozzle and referred to shaft centerline; and $(V_d^2 - V_s^2)/2g$ is the difference in the suction and discharge velocity heads.

Figure 11 shows a typical set of characteristic curves for one impeller speed, N. Salient points to note are the shutoff head H_{so} or the head at zero flow, and the best efficiency point, *BEP*. Pump efficiency, E, at any given coordinate of H and Q is calculated as

$$E = 9{,}797QH\gamma\ /P$$

where H and Q are in units of m and m^3/s, respectively; g is the fluid specific weight; and P is the shaft power in kW.

Efficiency is a measure of the degree of hydraulic and mechanical perfection of the pump. The foregoing ratio is, therefore, that of ideal hydraulic power to brake kW.

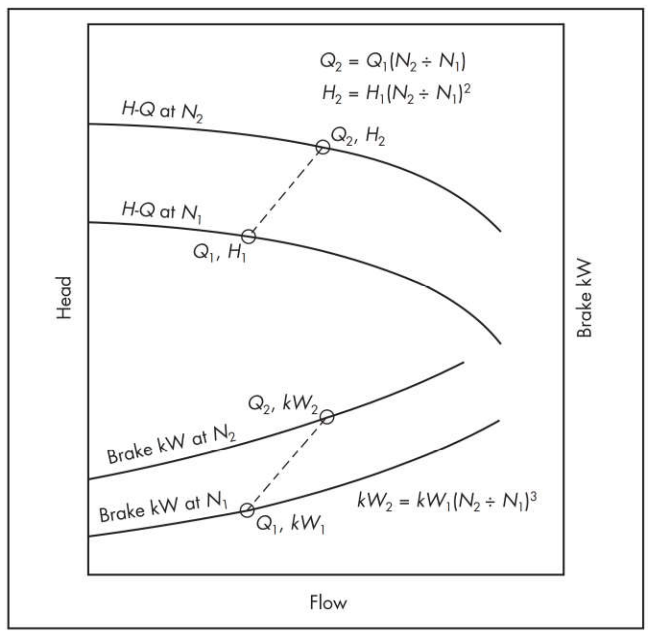
In every pump design, the NPSHR must be considered. The NPSHR increases with pumping capacity Q for a given pump speed N and also increases with pump speed at a constant Q, as shown in Figure 12.

The Affinity Laws

Figure 13 shows *H-Q* and *P-Q* curves for two pump speeds, N_1, and N_2. Coordinates of points between the corresponding characteristic curves of two different speeds are related in a definite manner. For example, a point Q_1, H_1 on the *H-Q* curve at speed N_1 has a corresponding point Q_2, H_2 on the *H-Q* curve at speed N_2. The relationships are

$$Q_2 = Q_1\ (N_2/N_1)$$

$$H_2 = H_1\ (N_2/N_1)^2$$



Adapted from Link et al. 1985

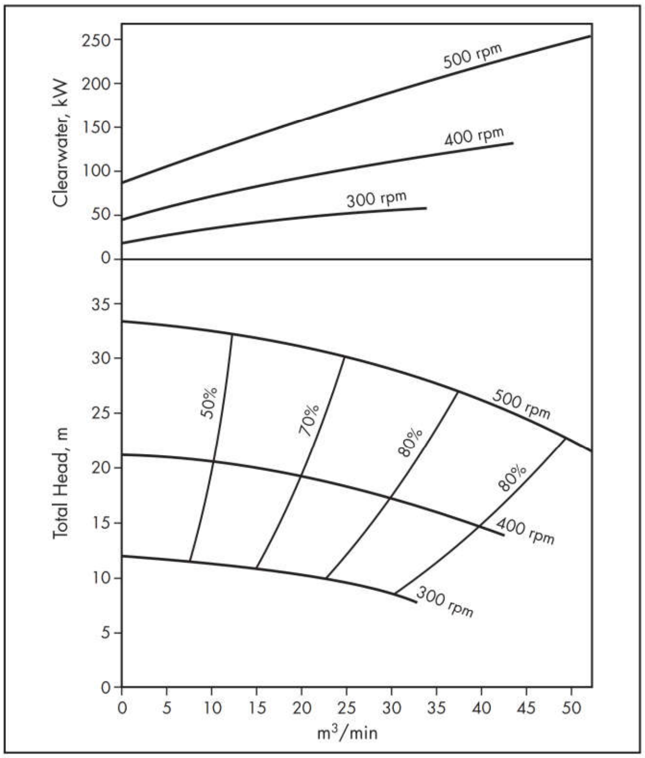
Figure 13 Affinity law relationships

$$P_2 = P_1\ (N_2/N_1)^3$$

These relationships form the affinity laws. Further, any set of corresponding points occurs at an almost identical pump efficiency.

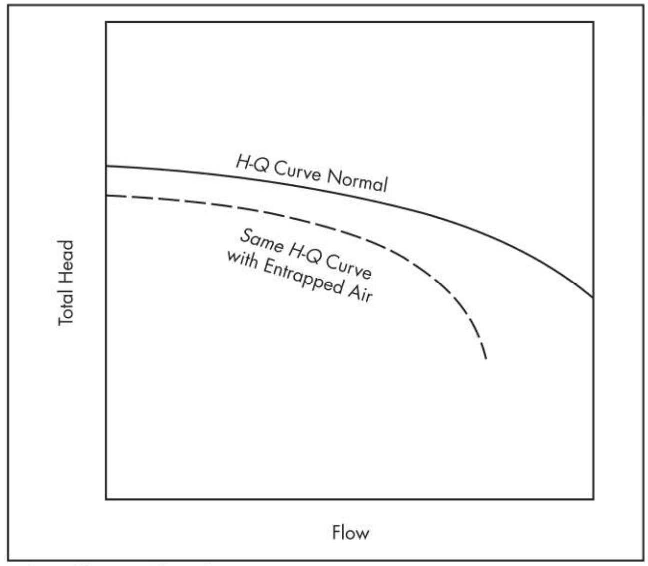
The affinity laws can be used to generate a family of pump curves such as shown in Figure 14. Accuracy in generating these curves is good if N_1 and N_2 are sufficiently close. However, as the difference between N_1 and N_2, increases, the accuracy of the curve will decrease because of variances in mechanical losses at different speeds.

Pump manufacturers publish a family of curves for each available pump model and standard impeller. These curves



Adapted from Link et al. 1985

Figure 14 A family of pump curves



Adapted from Tomb et al. 1985

Figure 15 Effect of entrained air on *H* versus *Q*

are almost always developed by using clear, cold water as the liquid being pumped and are called *clearwater* performance curves. Power is referred to as clearwater power. Because some liquids do not have physical properties identical to those of water, specific gravity and viscosity of the liquid in question must be known before clearwater curves can be used. The corrections to be applied to clearwater performance are considerably more complex if the liquid to be pumped has a viscosity much different from that of water. The pump manufacturer should be consulted in these cases. Pumping of slurries is discussed later.

Pump performance can also be affected by entrained air in the liquid. Because the pump is a centrifugal device, the heavier liquid is thrown away from the impeller eye, while the lighter air will remain behind at the eye until enough air is accumulated to choke off the eye area. This can result in an *H-Q* curve that is actually lower than, and drops off faster than, the desired *H-Q* curve, as shown in Figure 15. Lack of sufficient NPSH will affect pump performance in the same way as entrained air.

Specific Speed

Specific speed, as described previously, is to some extent a characterization of pump type. All pumps of a similar geometric configuration have a fairly constant specific speed. This relationship is shown in Figure 16.

Centrifugal pumps are the most widely used in industry, and impellers for centrifugal pumps have been further classified by various configurations in relation to optimum specific speed for each. Figure 17 shows the classification of centrifugal pump impellers and the associated specific speeds. The four types of impellers are shown in cross section.

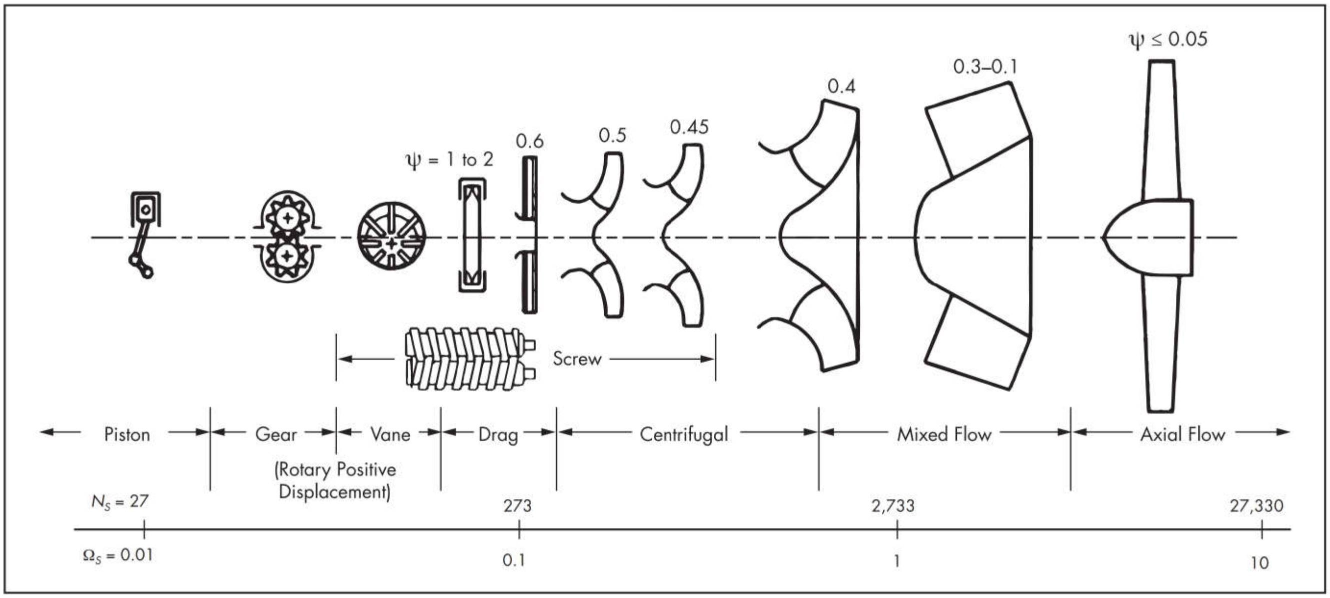
Radial-vane pumps develop pressure principally by the action of centrifugal force. In straight-vane radial pumps, the vane surfaces are generated by straight lines parallel to the axis of rotation. Pumps in this class with single-inlet impellers have the lowest specific speeds, with rotational speeds below 4,200 rpm, while pumps with double-suction radial impellers have rotational speeds below 6,000 rpm. In pumps of this class, the liquid normally enters the impeller at the hub and flows radially to the periphery.

In mixed-flow pumps, the head is developed partly by centrifugal force and partly by the lift of the vanes on the liquid. The impeller vanes in these pumps have a double curvature, which imparts both a radial and an axial component to the flow out of the impeller. Mixed-flow pumps that operate at lower specific speeds use Francis vane impellers. This type of pump has a single-inlet impeller with the flow entering axially and discharging in an axial and radial direction. Pumps of this type usually have rotational speeds between 4,200 and 9,000 rpm.

Axial flow or propeller pumps develop most of their head by the propelling or lifting action of the vanes on the liquid. Each has a single-inlet impeller with the flow entering axially and discharging nearly axially. Pumps of this type usually have rotational speeds above 9,000 rpm.

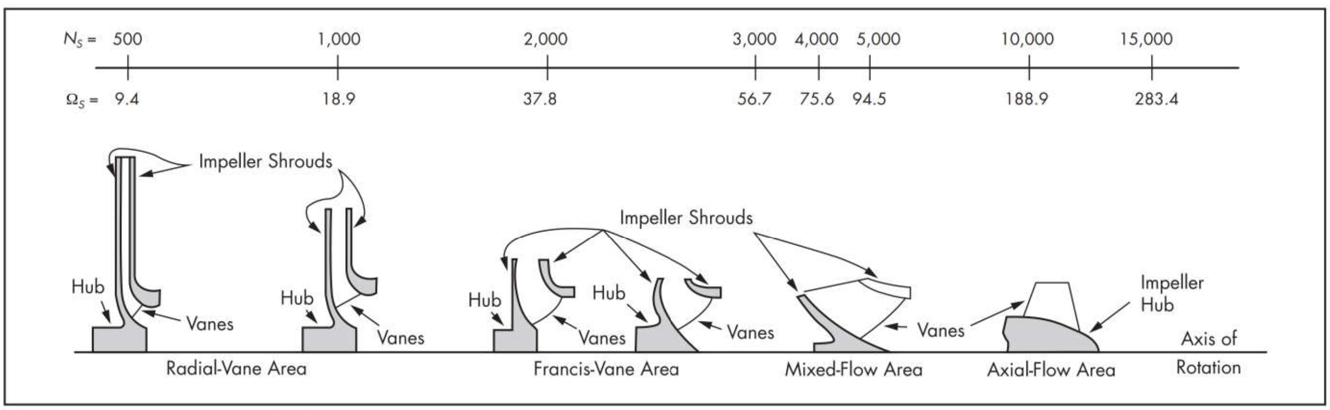
The specific speed equation shows that, for a given head-capacity duty, pumps with higher specific speed will run at higher rotational speeds than those with lower specific speed. Higher specific-speed pumps have smaller-diameter impellers than their lower specific-speed counterparts. Efficiency is generally lower for low-capacity, high-speed pumps and greater for high-capacity, high-speed pumps. Peak efficiencies for ordinary centrifugal pumps range from 90% to 92%.

Specific speed is often the first concept used by a pump designer to estimate the general hydraulic characteristics required for specific markets and applications. Once a pump type is selected, characteristics are refined by the details of volute and impeller geometry—diameter, number of vanes, shape of vanes, and so forth. For example, impellers in radial-flow, end-suction pumps may or may not have a suction shroud



Source: Karassik et al. 2001

Figure 16 Optimum pump geometry as a function of BEP specific speed (unitless) for single-stage rotors



Adapted from Hydraulic Institute 1975

Figure 17 Types of centrifugal pump impellers related to pump specific speed

that covers the outside vanes. An open impeller has such a shroud, while a closed impeller does not. A closed impeller may result in a higher pump efficiency, but an open impeller may be desirable for pumping difficult materials such as froths, to prevent air entrainment at the eye. Both types of impellers are shown in Figure 18.

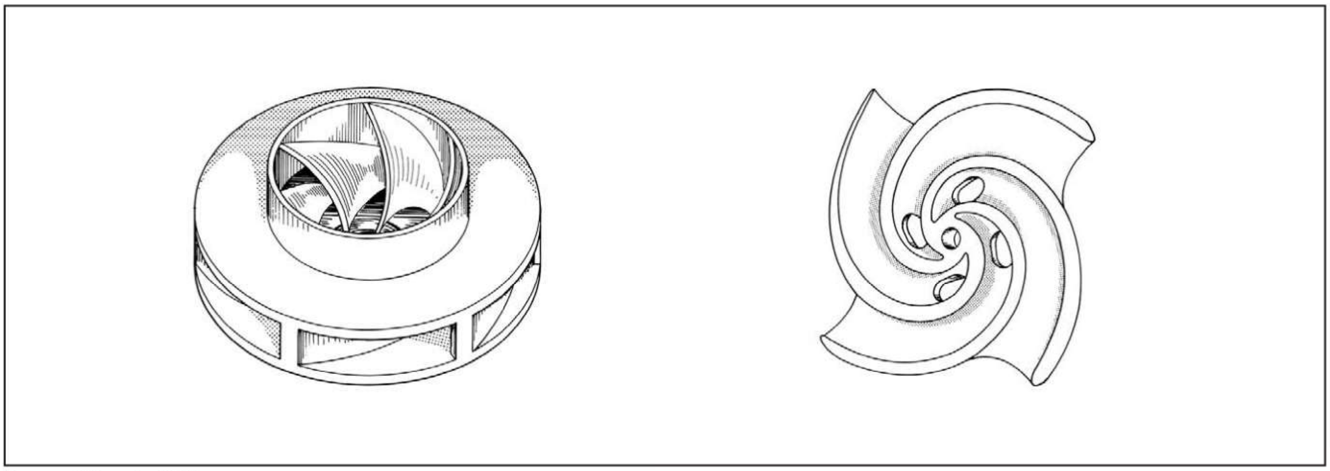
Figure 19 shows various types of pump characteristics. Head-capacity curves *a*, *b*, *c*, and *d* are called stable curves because only one flow is associated with a given head. In contrast, curve *b* has an unstable characteristic. The power-capacity curves labeled *a*, *b*, and *c* correspond to the head-capacity curves *a*, *b*, and *c*. Curve *a* is called a non-overloading characteristic, since power decreases when the head is either increased or decreased. Curve *b* is called an overloading curve with a reduction in head, and curve *c* is called an overloading curve with an increase in head. Curve *c* is typical of an axial flow pump.

Pumps in Series and Parallel Operation

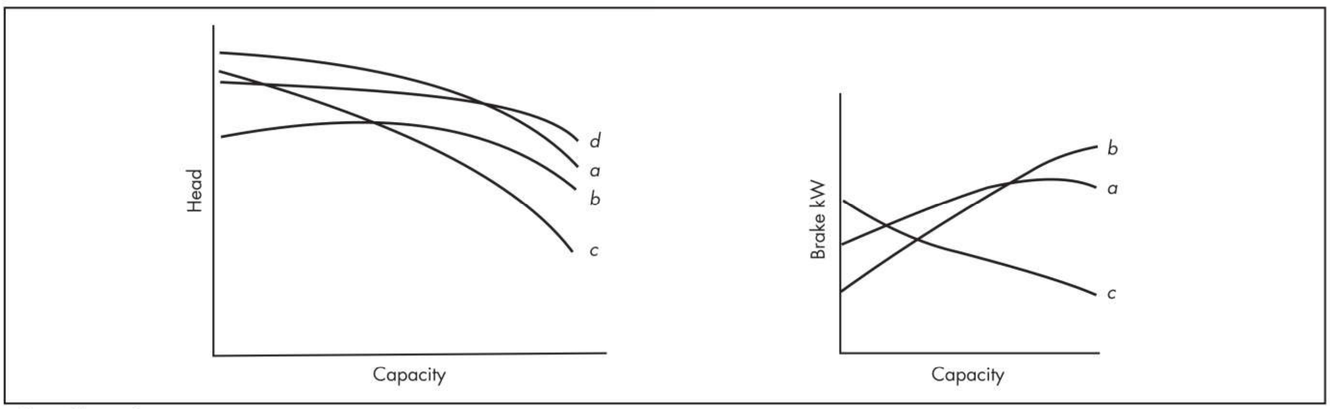
Centrifugal pumps can be used in parallel or series operation to multiply, respectively, the volume or head capability of a single unit, as shown in Figure 20. In parallel pumping, the discharge lines of two or more pumps are combined into a common discharge line.

Figure 21 shows a combined head-capacity curve for two identical pumps operating at the same speed and installed using identical symmetrical piping to a common discharge point. Curve A is an *H-Q* curve for single pump running at speed N_1. Curve B is the *H-Q* curve for two identical pumps at speed N_1 operating in parallel. The intersection of the system-head curve with curves A and B indicates the increase in flow, $Q_B - Q_A$, possible by going to parallel operation.

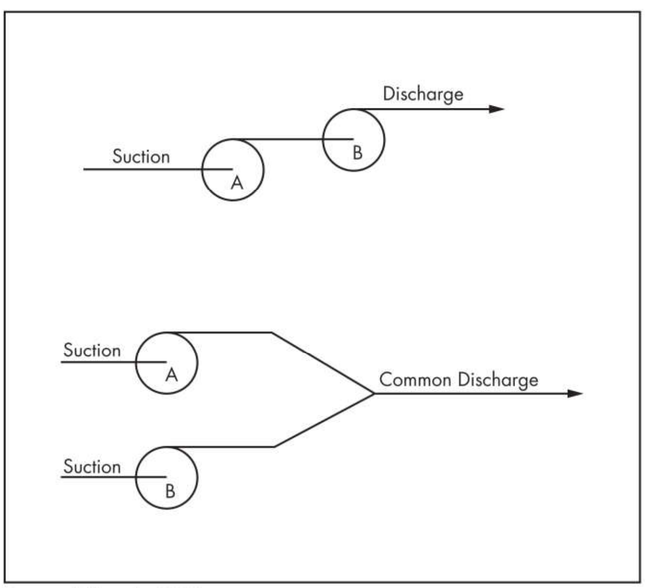
Theoretically, each pump in this situation should handle one-half of the pumped capacity. However, imbalances between the two pumps can occur because of minor variations



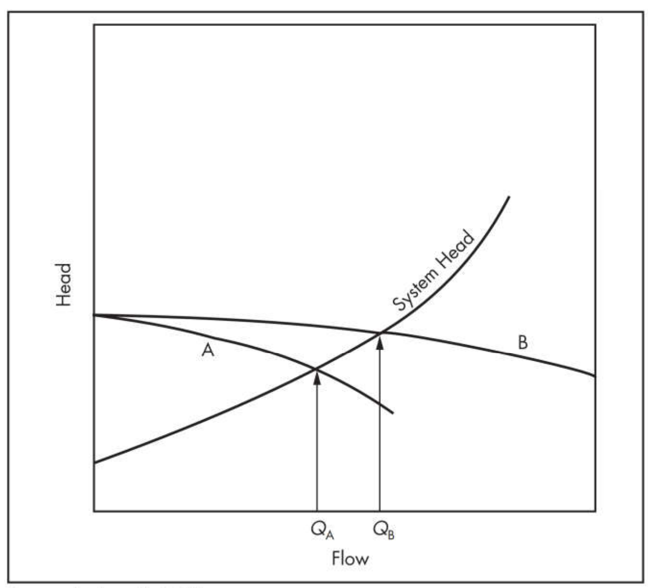
Source: Link et al. 1985
Figure 18 Closed (left) and open (right) impellers



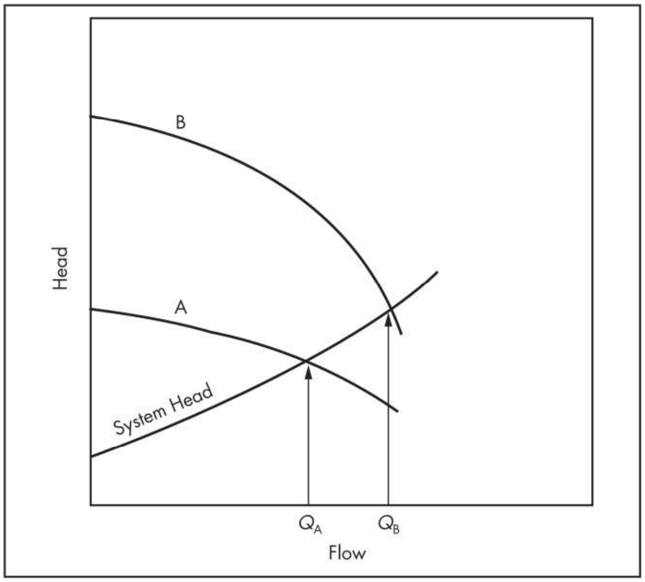
Adapted from Olson 1998
Figure 19 Centrifugal pump characteristic related to head and capacity



Source: Link et al. 1985
Figure 20 Pumps in series and parallel connections

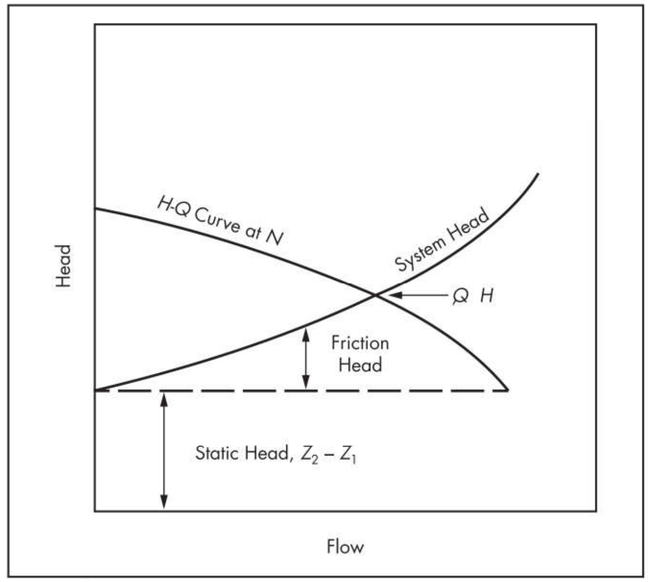


Source: Link et al. 1985
Figure 21 Head versus capacity for centrifugal pumps connected in parallel



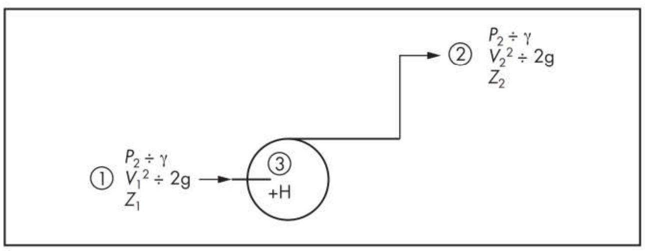
Source: Link et al. 1985

Figure 22 Head versus capacity for centrifugal pumps connected in series



Adapted from Link et al. 1985

Figure 24 Typical system-head curve for a centrifugal pumping system



Adapted from Link et al. 1985

Figure 23 Energy in a simple pumping system

in motor speeds, V-belt drive slippage, pump wear, clearance differences, and so on. These imbalances can sometimes be modulated by the use of control valves or variable-speed drives for trimming respective pump speeds.

Parallel operation is best achieved by using pumps with fairly steep head-capacity characteristics. Notably, the flow of one pump will not necessarily be doubled or tripled by the corresponding addition of two or three pumps. Rather, this is a function of the system-head curve with special attention given to the friction head contribution.

In series pumping, the head capability of one pump is increased as long as it is possible to feed the discharge of one pump into the suction of the next without exceeding the pressure rating of the last-stage pump. In multistage pumps, this can be accomplished in a single unit by using one casing with diffuser plates. For example, in boiler feed pumps, thousands of pounds of inlet pressure provide a self-priming ability better than that of other kinetic pumps.

Separate, horizontal centrifugal pumps may be staged such as is done in the transport of mine tailings. Staging is done by connecting pumps, either in series or in parallel.

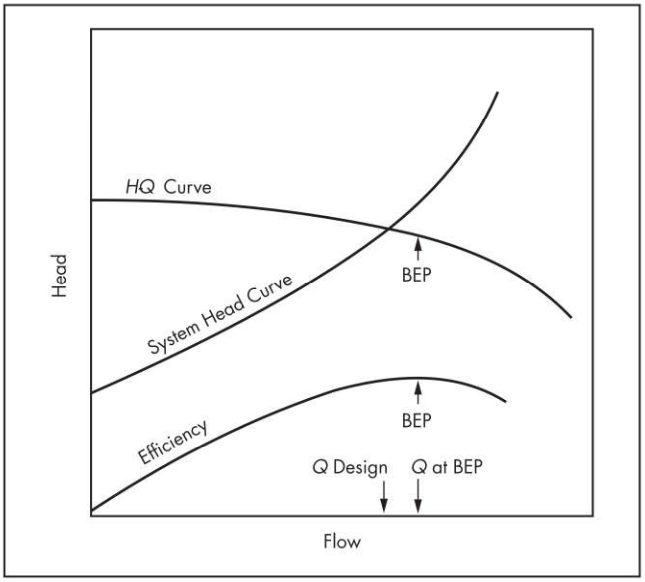
Figure 22 illustrates the combined head-capacity curve for two identical centrifugal pumps operating at the same speed in series. As for the pumps connected in parallel described above, the superposition of the system-head curves shows the flow capability of the pumping system. Curve A is an *H-Q* curve for a single pump running at speed N_1, and Curve B is an *H-Q* curve for two pumps connected in series and operating at speed N_1. The intersections of the system-head curve with the two operating curves indicates the respective flows possible for each configuration. The potential increase in flow quantity by connecting a second identical pump in series is given simply by the difference, $Q_B - Q_A$.

Selection and Operation

Proper pump selection must begin with the nature of the liquid to be pumped. Specialty centrifugal pumps exist for practically every service, such as pumping of corrosive materials, clear liquids or solid suspensions, viscous oils, sludges, high- and low-temperature liquids, and so forth. Many applications can be served by more than one pump type and by a variety of manufacturers.

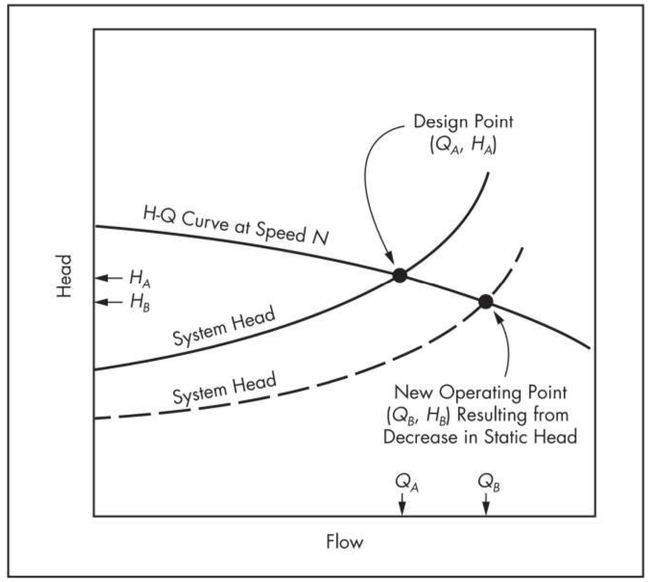
A system-head curve should be developed as previously described. Consider the simple system in Figure 23. The system-head curve showing the energy requirement in meters of liquid H at each flow Q may look like that shown in Figure 24, depending on the line, valve, and fitting losses that exist in the system.

The system-head curve reflects the static head, $Z_2 - Z_1$, that must be overcome regardless of the flow, in addition to the friction losses in the suction line from point 1 to point 3, and the discharge line from point 3 to point 2 in Figure 23. In Figure 25 the system-head curve is compared to a selected pump curve. The intersection of the appropriate pump curve at point (Q_A, H_A) defines the quantity and head that the pump will provide when operating at rotational speed N_A. From the complete characteristics of the pump at speed N_A, the pump power, efficiency, and NPSHR can also be determined. It is best to have the pump operating point as close as possible to the BEP without going too far to the right of this BEP, as shown in Figure 25. At the BEP, the flow within the pump



Source: Link et al. 1985

Figure 25 Pump curve showing selection for best efficiency point



Source: Link et al. 1985

Figure 26 Change in operating curve resulting from change in static head

produces the least turbulence and recirculation is minimal so that efficiency is maximized.

Operation too far to the right of the BEP invites cavitation, and NPSH demands rise rapidly; operation too far to the left of the BEP invites high radial loads and operating instabilities due to the flatter nature of the *H-Q* curve near shutoff. As a rule of thumb, the operating point should be between 40% and 115% of BEP flow.

Variations in the system-head curve should also be considered. System friction may be increased or decreased, for example, throttling a discharge valve. Static head can vary with changes in the suction receiver level or discharge point. Variation in the system-head curve that is imposed will automatically define a new pump operating point, as shown in Figure 26. This is because the pump will operate at the intersection of the system-head curve and pump curve. If such potential variations are disregarded, the pump driver could be underpowered.

Control of Cavitation

Cavitation always increases pump wear. This is especially true for slurry pumps. Severe cavitation can also cause severe damage, leading to failure of the pump. Depending on the severity of the cavitation, high noise and vibration may result, along with a reduced head output, lower flow, or complete loss of prime.

Whether or not cavitation occurs depends on the suction characteristics of the pump and the system in which it is placed. To prevent cavitation, the NPSHA should exceed the NPSHR by 15% to 30%. The NPSHR for a particular pump is determined by the pump design and may be found on the manufacturer's performance curve at the pump duty point in the system. The NPSHR usually increases with flow and pump speed, but an increase can also occur at extremely low flows near pump shutoff.

Values of NPSHA for the system are not simply the height of liquid above the centerline of the pump. Rather, NPSHA is a function of the local atmospheric pressure, the height of the liquid relative to the pump centerline, the friction losses in the suction pipe, and the vapor pressure of the liquid at the pumping temperature. Figure 27 shows how to calculate NPSHA for various suction conditions (Roudnev and Angle 1999).

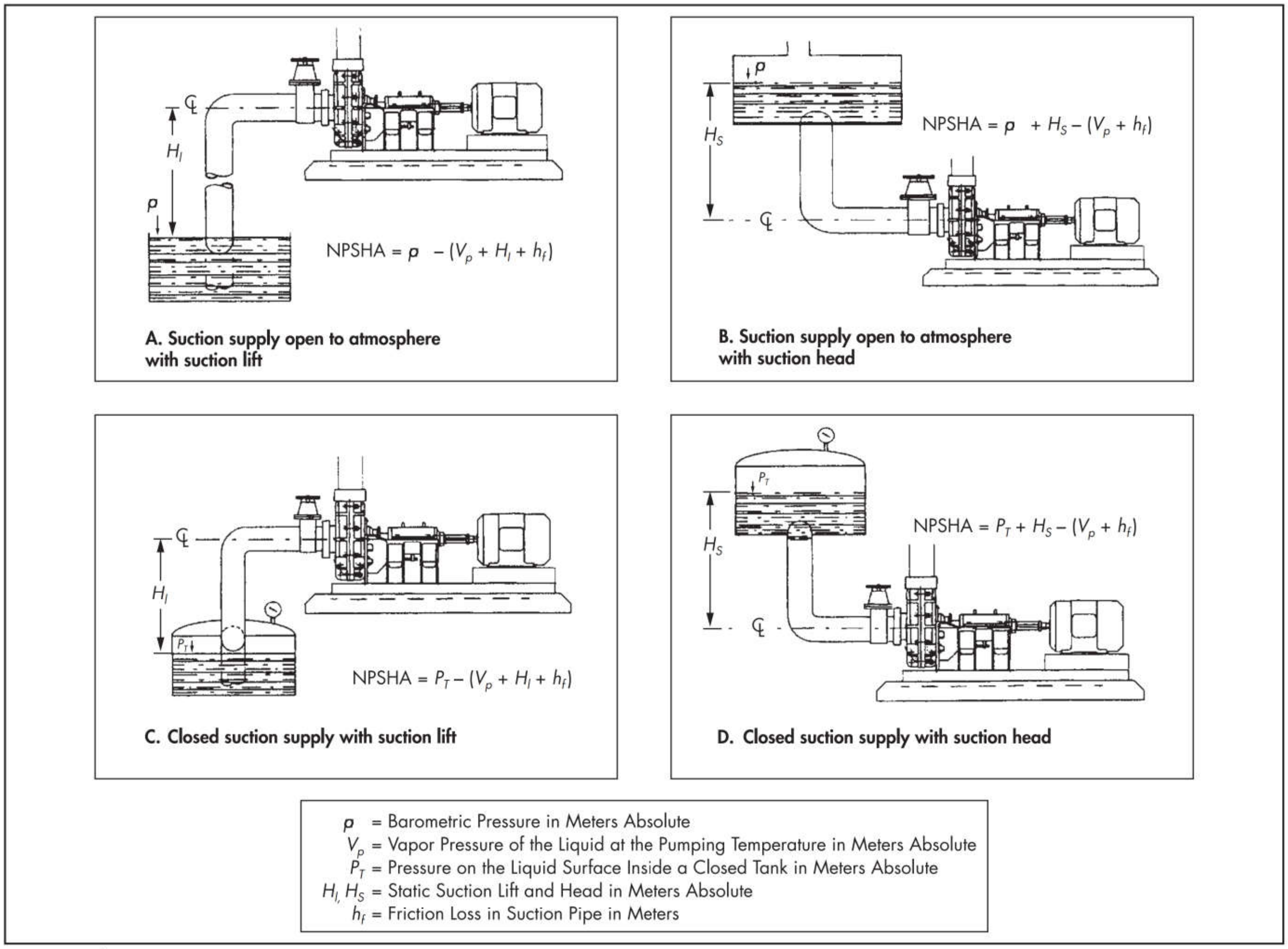
For calculations under suction lift conditions, as shown in Figure 27A, the atmospheric pressure *pa* should be reduced by dividing it by the specific gravity of the liquid. This is to account for the weight of the liquid in the column and the negative effect this has on the suction pressure at the pump. The application of any positive correction under flooded conditions is not recommended.

Cavitation can be easily prevented during the system design process by ensuring there is adequate suction height, minimal suction friction, and proper pump selection. After system construction, these factors are much more difficult to change.

Installation

The proper selection of a suitably designed centrifugal pump is not sufficient to ensure its success. Equally important is the installation and maintenance of the unit. The pump should be placed as close as possible to the reservoir or suction source as practical. Discharge lines and suction lines should be kept as short and direct as possible to minimize friction losses and hence TDH. Valves and fittings should be kept to the necessary minimum for operation and control. The pump should have ready access for inspection and maintenance. Sufficient headroom should be provided for the installation of a hoist if needed.

Experienced operators report that most pump problems occur on the suction side of the pump. Careful attention should be paid to the suction piping and construction of the sump or reservoir from which the pump is fed. Sump design should provide sufficient volume or retention for the required pump flow with a suitable geometry to prevent the formation of vortices and potential air entrapment in the liquid. Flow



Source: Bootle 2002

Figure 27 Calculation of NPSHA for various pump system configurations

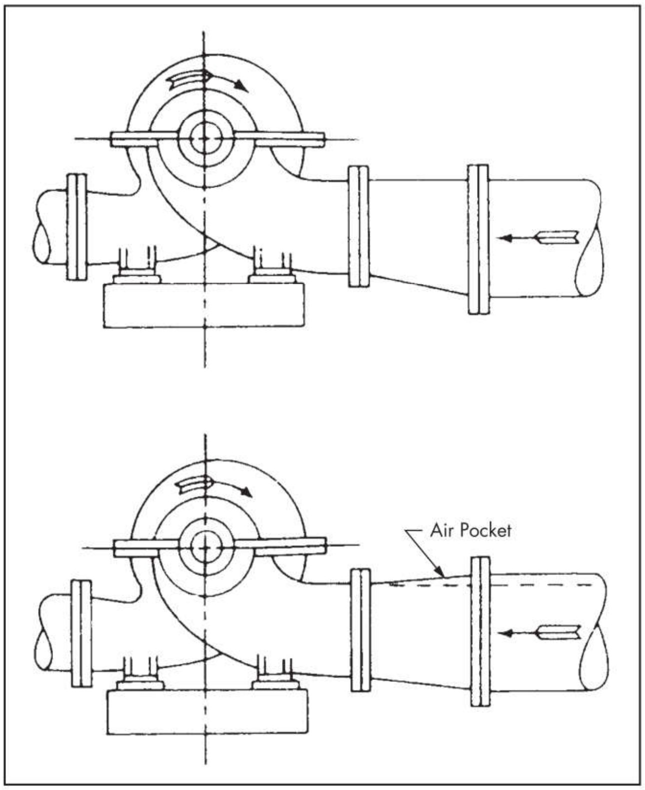
should be directed to the suction piping in evenly distributed streamlines. This applies to both vertical pumps submerged in the sump and horizontal pumps located outside the sump confines. Very often a flared bell-mouthed fitting is employed at the end of a suction pipe protruding into the sump to capture the entering flow at a low average velocity and minimize friction losses.

It is, of course, imperative that sufficient NPSH be available for the pump to operate free from vibration and cavitation. Beyond this, however, there must be sufficient height of liquid in the sump above the suction intake of the pump to preclude vortexing and gulping of air. While each pump manufacturer will have its own recommendation of this minimum height, a safe rule of thumb is to have one meter of height for each meter per second of suction intake velocity.

For flows of 4 m^3/min or less, the sump volume should usually be on the order of one to two times the maximum quantity to be pumped. If the pump is operating in an on–off mode, as in response to a float switch, for example, the sump volume should be sized to allow no more than three to four starts per hour of the pump motor unit. Sump volumes may be minimized using a variable-speed pump drive, or by the action of a discharge control valve that adjusts the system-head curve. Again, sump design and operation should provide sufficient NPSH and liquid-level submergence for all pump conditions.

Generally, inlet flow to the sump should be from the sump wall opposite the pump suction intake. The influent should not jet into the pump inlet or enter the sump in such a manner as to cause a swirling flow. For this reason, rectangular-shaped sumps are usually preferred over cylindrical sumps. Baffles may be used effectively in sumps to break swirls, direct flows to pump intakes as required, and separate intakes when more than one pump is drawing from the sump.

Suction piping, as previously mentioned, should be as short and direct as possible with limited fittings to preserve NPSH characteristics. A surge isolation valve is generally required and should be either totally open or totally closed. It is extremely bad practice to attempt throttling flow in the suction line. A removable suction spool piece, installed with grooved-type couplings, is often a clever idea to permit access to the pump. A drain valve may be installed to remove residual liquid from the piping and associated piping or pump shutdowns. Suction piping may have a rise of at least 3° to 4° from sump to the pump to rid any entrained air from the system. If the piping must be reduced to the suction flange, an eccentric, rather than a concentric, reducer is preferred to



Source: Hydraulic Institute 1975

Figure 28 Correct installation of a reducer in the suction line

prevent trapping air in the inlet line, as shown in Figure 28. Packed-type valves used on the pump suction line should be sufficiently tight to prevent air leakage into the liquid stream.

Discharge piping should usually contain an on–off valve, such as a gate valve, and a check valve placed between the pump and gate valve. The gate valve is used in pump priming, starting (except with radial- and mixed-flow pumps), and shutting down. The check valve is installed to prevent backflow of the liquid through the pump, particularly when a drive power failure occurs. Such backflow can damage the pump and its accessories, especially when a high static head exists.

A pump should never be allowed to operate under the condition of a simultaneously closed suction and discharge. Energy fed by the drive into recirculating liquid within the pump can cause highly dangerous explosive forces.

The pump should be bolted to a foundation that is of adequate strength and design to absorb any normal forces transmitted to the pump by pipe loads, applied motor torque, and counter-produced impeller torque, and so on, to maintain alignment of the pump drive train. Usually the pump and its drive components are mounted on a solid base or pedestal that is anchored to foundation bolts of proper size imbedded in concrete. Piping connected to the pump should be properly supported so that minimum forces and moments are transmitted to the pump flanges.

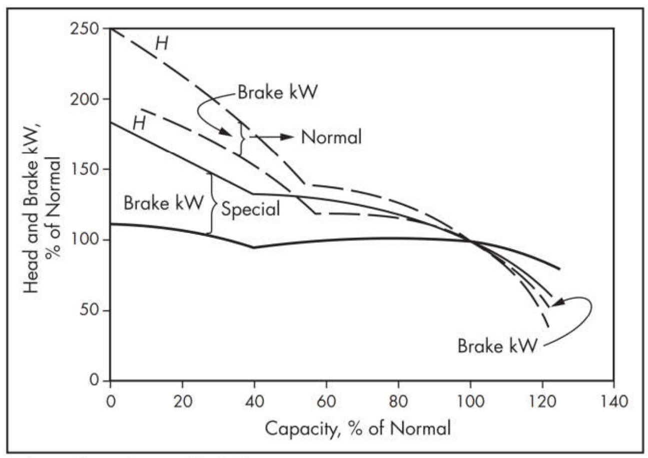
Axial-Flow Pumps

Axial-flow or propeller pumps are used for conditions of high capacity and low head—15 m or less per stage—and are particularly applicable in closed-loop circulation where the pump becomes a fitting in the line. A typical characteristic curve for an axial-flow pump is shown in Figure 29. Note the discontinuity in the head-capacity curve. Because the power requirement at shutoff may be double that of the design point, the pump discharge valve is often kept open on starting to keep the system head as low as possible. This is opposite to the practice used for radial-flow pumps where the discharge valve is usually shut on starting. The blade angle in an axial-flow pump can often be mechanically changed to alter performance and efficiency to a desired operating condition. An axial-flow pump is very sensitive to suction configurations, and even minor changes in inlet suction conditions can be very detrimental.

Regenerative or Peripheral Pumps

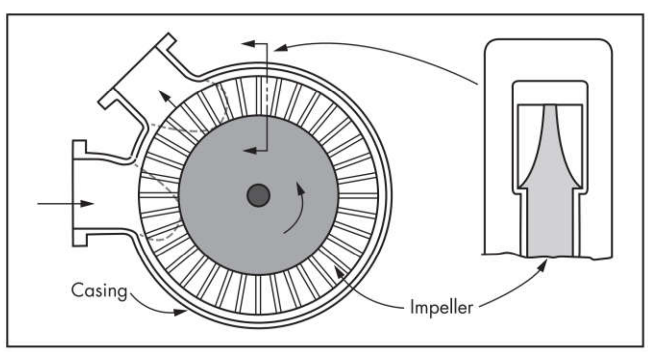
Regenerative pumps, also called peripheral or turbine pumps, depend on a combination of mechanical impulse and centrifugal force to generate heads up to 750 m at flows generally less than 0.5 m^3/min. In these pumps, as shown in Figure 30, the liquid spins within narrow recesses at the periphery of the impeller. This type of pump is excellent for handling vapor-laden liquids and is often used for boiler feed and process applications.

Regenerative turbine pumps typically have good NPSH performance and a steep *H-Q* curve. They are used as small boiler feed pumps, seal water pumps, and chemical process pumps. In a regenerative turbine pump, liquid recirculates



Adapted from Stepanoff 1948

Figure 29 Performance curve for a typical axial-flow pump



Source: O'Keefe 1972

Figure 30 Regenerative or peripheral pump

between the impeller vanes, so the fluid flows in a helical path as it is carried forward. Consequently, energy is added to the fluid in a regenerative motion by the impeller's vanes as it travels from suction to discharge. This regenerative action has an effect similar to that in a multistage centrifugal pump, wherein the fluid's pressure is the result of energy added in the various stages. In the same way, in a regenerative turbine pump, pressure at its discharge is the result of energy added to the fluid by many impeller vanes.

Special Centrifugal Pumps

Centrifugal pumps can be mounted vertically, with the packing box located well above the highest liquid level. These pumps have the suction immersed in a tank or sump and can occupy less overall floor area than might be needed for a comparable horizontal pump arrangement. Vertical pumps are frequently employed to handle toxic or corrosive process streams as well as for floor wash-down from plant collection pits. A vertical pump is shown in Figure 31.

In a continuous vertical pump, the bearings are located at the top of the shaft. This is the desirable design in most situations. Tapered shafts are used, and hydraulic radial thrust is often balanced using dual suctions and volutes. For extremely long shafts, intermediate and foot bearings must be added. These are difficult to protect when pumping corrosive liquids.

Other specialty pumps include the canned motor pump and the submersible pump. The canned motor pump is a close-coupled pump in which the cavity housing the motor, rotor, and pump volute are interconnected. Consequently, the motor bearings run in the process fluid and seals are eliminated. Flows up to 3 m^3/min and heads from 75 m (one-stage) to 180 m (two-stage) have been achieved. This type of pump has been used for pumping of organic solvents, toxic liquids, and other difficult liquids. An example is shown in Figure 32.

The fully submersible pump is also a closed-coupled, motor-driven unit in which the pump casing and motor are designed for underwater operation. These are excellent pumps for use in flooding conditions such as those encountered in underground mining operations. A main feature of this pump is its slim shape, which facilitates its application in deep boreholes. Flows up to 90 m^3/min can be obtained by multi-staging a series of mixed-flow impellers, and heads in excess of 975 m are possible with the proper choice of impellers.

Positive Displacement Pumps

PD pumps function by displacing a fixed amount of liquid, either by reciprocating or rotary motion. The prime feature of a PD pump is that a definite quantity of liquid will be delivered for each stroke or revolution of the prime mover. Generally, only pump size, design, or suction characteristics will affect the pump capacity. An important consideration for PD pumps is their *volumetric efficiency*, which is defined as the ratio of actual volume displaced to the theoretical or geometric volume of the pump. The difference between these two volumes represents internal leakages within the pump.

Reciprocating pumps add energy to a hydraulic system using a piston or a plunger acting against a confined fluid. In either type, liquid enters the pump cylinder through a check valve that is opened by external pressure acting on the liquid. If the piston carries its own seals or packing with it, the pump is referred to as a piston pump, as shown in Figure 33.

Inflow of the liquid follows the piston movement backward through the cylinder on the inlet stroke. Forward movement of the piston forces the inlet valve to close and the discharge valve to open. The piston must have a close fit with the cylinder walls to minimize slip past the piston seals.

A plunger pump, as shown in Figure 34, uses a close-fitting rod moving through the cylinder past fixed packing. The plunger pump is often used to handle slurries, in which case the plunger may be designed to collapse slightly on the suction stroke. The system provides a small quantity of flush water to protect the seals and minimize wear.

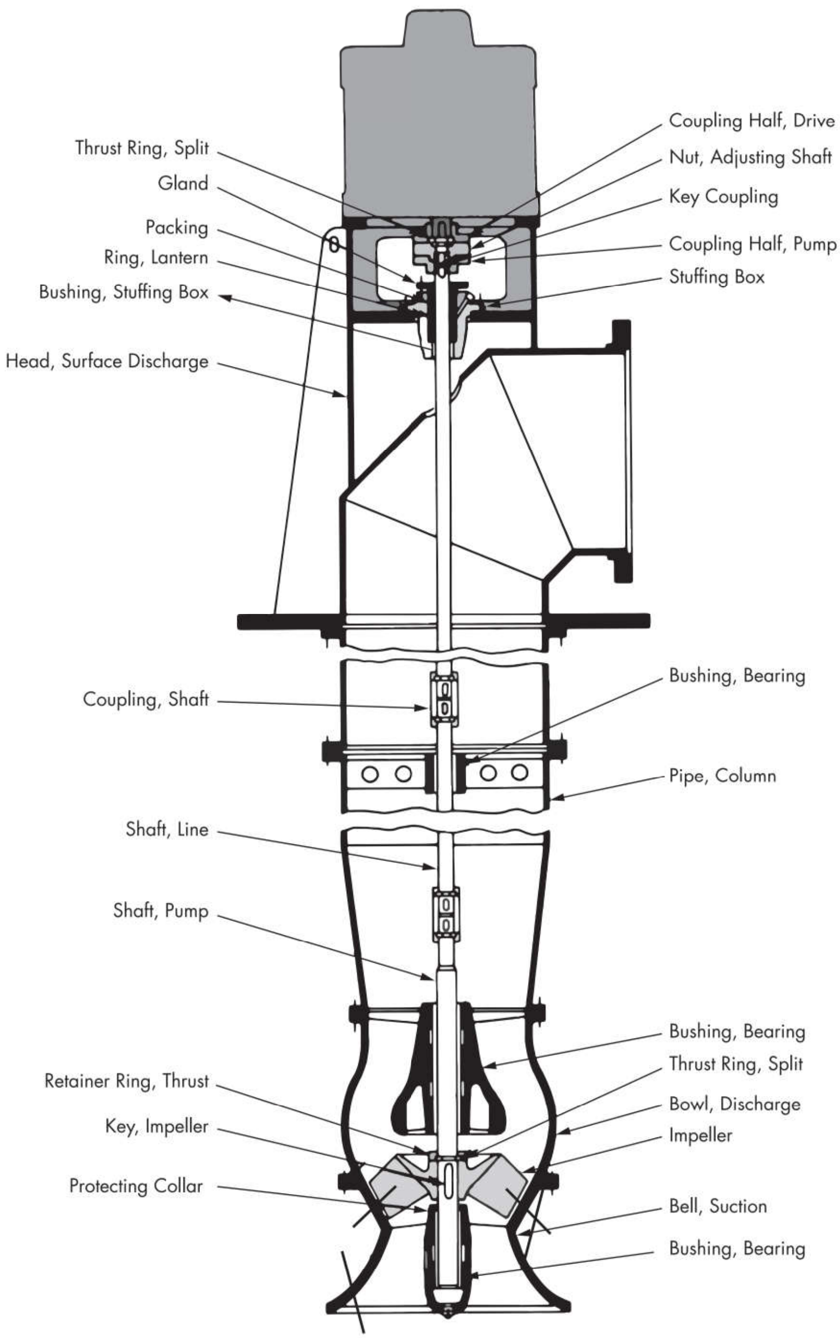
The suction stroke of a reciprocating pump must be designed to prevent cavitation. As the piston moves back on the suction stroke, the valve must open and allow the fluid to pass into the cylinder. Normally the valve head is much smaller than the cylinder, and fluid must pass through the valve at very high speeds. To prevent cavitation within the cylinder or suction lines, piston velocities must be kept as low as possible. Therefore, most reciprocating pumps are operated at 50–100 rpm. Suction lines are also kept relatively large so that velocities will be low and acceleration–deceleration problems minimized. These pumps will operate at relatively large negative suction heads.

The discharge side of a reciprocating pump should be designed to minimize vibrations. As the piston forces the fluid out of the cylinder, local pressures may increase substantially above the downstream line pressure. This results in a series of vibrations that must be damped out by using an air chamber or pressure accumulator. Very high discharge pressures can be attained with reciprocating pumps.

Discharge characteristics of reciprocating pumps are shown in Figure 35. Liquid moves from the discharge of the pump valve to the end of the scroll when the piston stops and begins to reverse. When the pump is a single-acting design, flow stops as the piston reverses. Flow from the discharge of a reciprocating pump can be made more constant using duplex, triplex, or even multiplexed designs. Duplex or double-acting pumps always have a flow moving into the discharge line. In such pumps, the discharge of one cylinder is displaced about one-half stroke from that of the other cylinder, so the total pump flow is the sum of the flow from both cylinders. Nearly pulse-free flow can be achieved by designing for duplex, triplex, or multiplex operation.

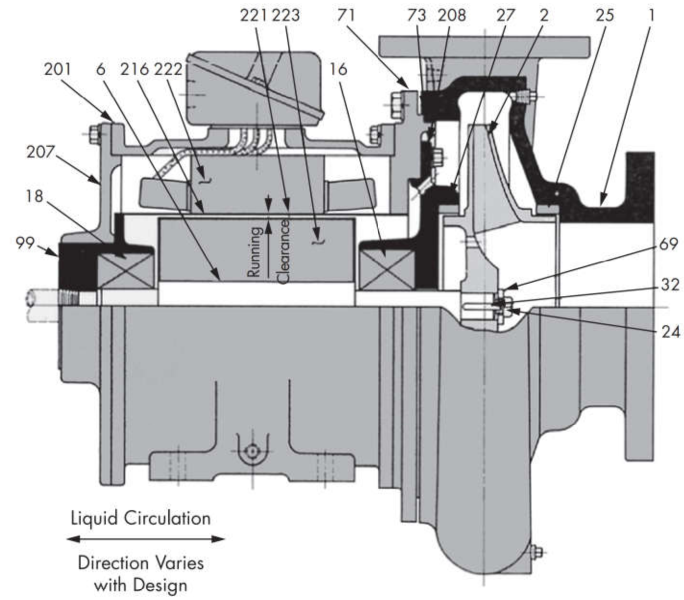
Rotary, PD pumps do not rely on valves to control suction or discharge. Rather, they draw liquid at the intake and move it continually to the discharge. This can be done with vanes, reciprocating pistons, gears, a flexible member, lobes, or screws—there are many variations. The pumps shown in Figures 36–38 are typical of those used in mineral processing applications.

The diaphragm pump, shown in Figure 39, is similar to the reciprocating pump. It functions by a back-and-forth motion of a diaphragm made of an elastomer or metal. The diaphragm may be actuated by a piston, as shown in the figure, or by a fluid (usually hydraulic oil or compressed air), which is introduced on one side of the diaphragm, displacing liquid to be pumped from the other side of the diaphragm. This pump requires no packing because the fluid being pumped is isolated inside the diaphragm. It is therefore well suited in pumping difficult materials such as slurries and corrosives. Air-operated diaphragm pumps are often used for slurries.



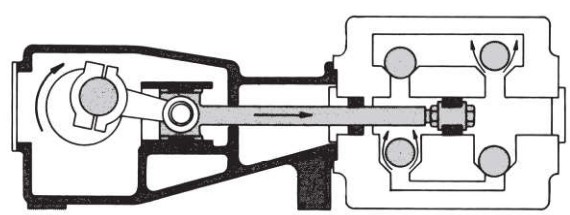
Source: Hydraulic Institute 1975

Figure 31 Vertical centrifugal pump

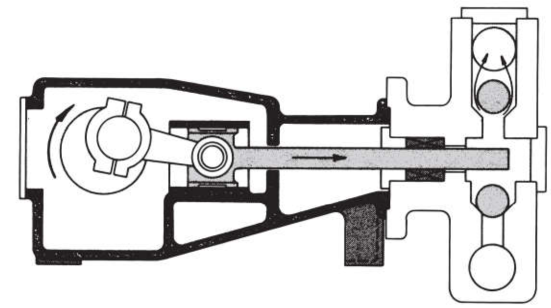


1	Casing	27	Ring, Stuffing Box	207	Cover, Motor End
2	Impeller	32	Key, Impeller	208	Housing Bearing and Wearing Ring
6	Shaft, Pump	69	Lockwasher	216	Can, Stator
16	Bearing, Inboard	71	Adapter	221	Can, Rotor
18	Bearing, Outboard	73	Gasket	222	Assembly, Stator Core
24	Nut, Impeller	99	Housing, Bearing	223	Assembly, Rotor Core
25	Ring, Suction Cover	201	Housing, Stator		

Source: Hydraulic Institute 1975
Figure 32 Canned motor pump



Source: Hydraulic Institute 1975
Figure 33 Horizontal, double-acting piston pump



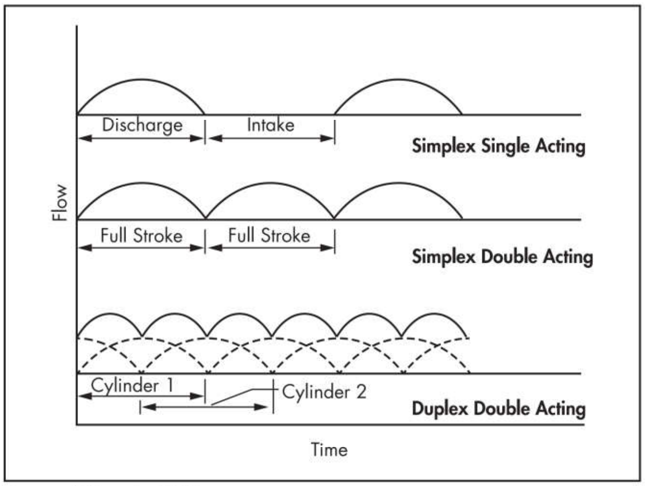
Source: Hydraulic Institute 1975
Figure 34 Horizontal, single-acting plunger pump

Another type of PD pump is the progressing cavity pump, which is often referred to as a "Moyno" pump, after René Moineau, its inventor. Moineau, a pioneer of aviation technology, was designing a compressor for jet engines and realized that this principle could also work as a pumping system. As shown in Figure 40, this pump consists of a rotor turning within a stator. The stator design creates a helical movement where the rotor contacts every surface of the stator. Voids between rotor and stator entrap material that is continually passed from suction to discharge.

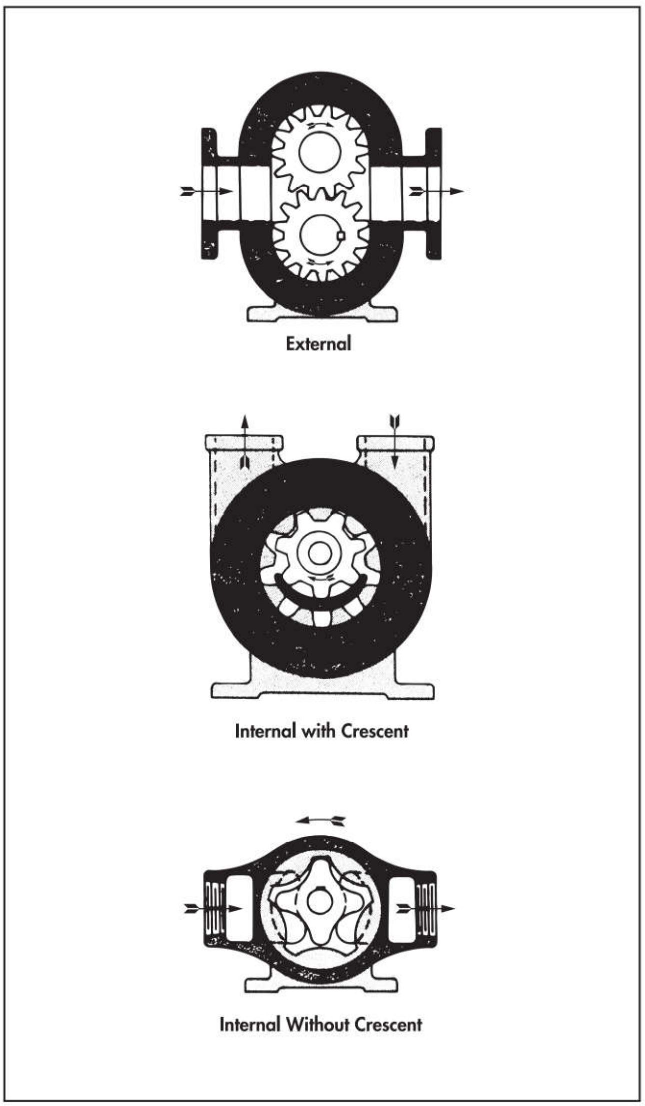
CONSIDERATIONS IN PUMP SELECTION

Net Positive Suction Head

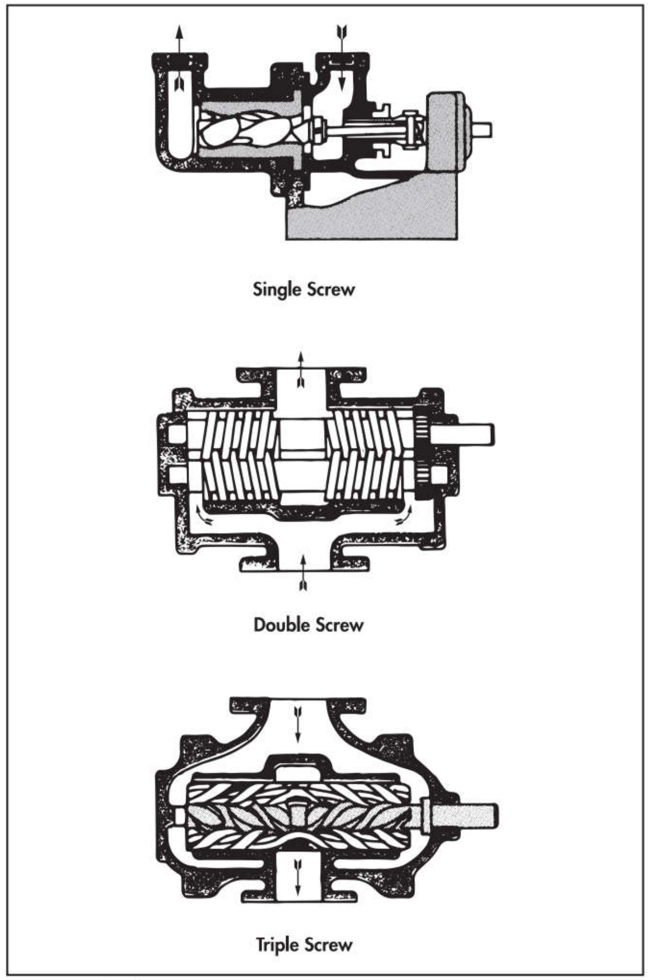
As noted earlier, the NPSHR, or net positive suction head required, is very important in the design of a pumping system. If the NPSH available from the system at the impeller eye does not meet the NPSH required by the pump, cavitation will result. The effects of cavitation vary in severity, from pitting of the impeller at the leading edge of the vanes and impeller periphery to large chunking of the impeller. There is



Source: Link et al. 1985
Figure 35 Discharge characteristics of reciprocating pumps



Source: Hydraulic Institute 1975
Figure 36 Gear pumps



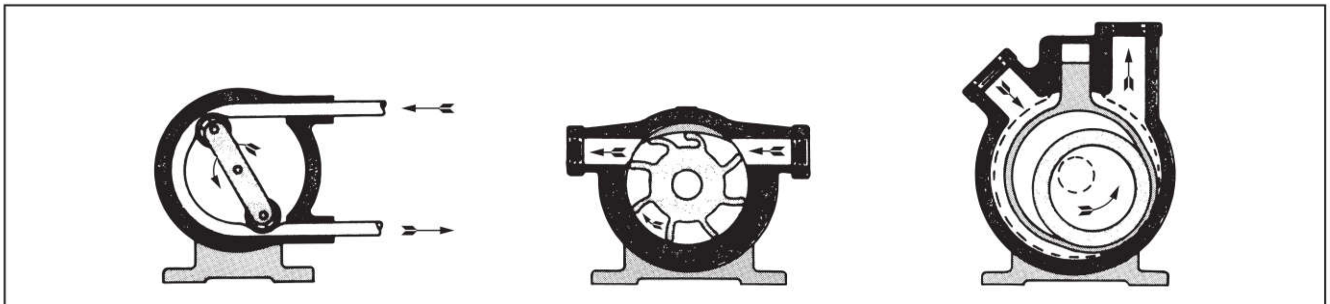
Source: Hydraulic Institute 1975
Figure 37 Screw pumps

generally a loss of pump performance, and a pump operating under a cavitation mode may be relatively quiet or sound as though pebbles are being pumped through it. Severe cavitation can cause vibration severe enough to move the pump off its foundation.

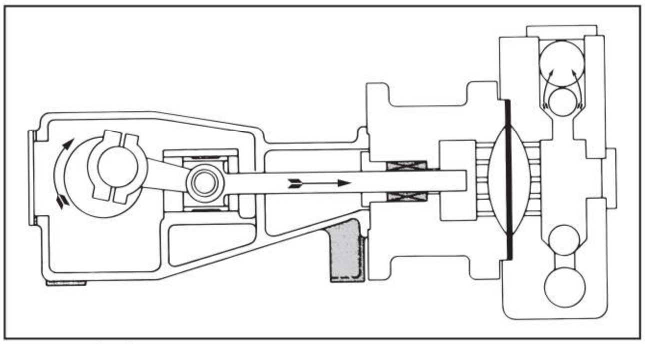
The calculation of NPSHA is a determination of the absolute pressure available at the eye of the impeller over the vapor pressure of the liquid being pumped in terms of meters of head of that liquid:

$$\mathrm{NPSHA} = (H_{\mathrm{atm}} \pm H_s - H_f) - (H_{vp})$$

where H_{atm} is the atmospheric pressure determined by dividing the atmospheric pressure in meters of water by the specific gravity of the liquid being pumped. H_s is the static height in meters of liquid above the center datum line of the pump volute on the suction side of the pump; H_s is positive if the suction head is positive and negative if the pump is lifting liquid to its suction eye. H_f is the friction loss in the suction line to the pump. H_{vp} is the vapor pressure of the liquid pumped in terms of meters of the liquid pumped and is obtained by converting to equivalent meters of water and then dividing by the specific gravity of the liquid being pumped. As a rule, NPSHA should exceed NPSHR by a sufficient margin to account for variations in system head, friction losses, density, temperature, and so forth.



Source: Hydraulic Institute 1975
Figure 38 Flexible member pumps



Source: Hydraulic Institute 1975
Figure 39 Horizontal, single-acting, flat diaphragm pump

Mechanical Considerations

Centrifugal Pump Drive Shafts

Mechanical considerations are important in centrifugal pump design. For analysis, consider Figure 41, which shows a pump impeller as an overhung load on a shaft, supported by two sets of bearings. Loads transmitted to the bearings include hydraulic loads and loads imposed by the weight of components attached to the shaft, such as those from the impeller and the drive pulley. Other loads that are applied include belt pull, unbalance loads, and so on.

Hydraulic loads may be divided into thrust loads and radial loads. Thrust loads are directed axially along the shaft and result because of pressure differentials across the shrouds of the impeller. Since the suction eye of the impeller is the low-pressure region, the axial load is directed toward this area. Radial hydraulic loads result from the interaction of the existing liquid from the impeller and the casing. The magnitude and direction of these loads depend on the location of the operating point with respect to the BEP of the head-capacity curve on which the pump is operating, as can be seen in Figures 42 and 43. The farther the operating point is from the BEP, particularly in the direction of shutoff, the greater the radial load.

The thrust factor, K, shown in Figure 42, is found in the empirical equation that describes radial thrust, T_r:

$$T_r = K\,H\,D_2\,B_2$$

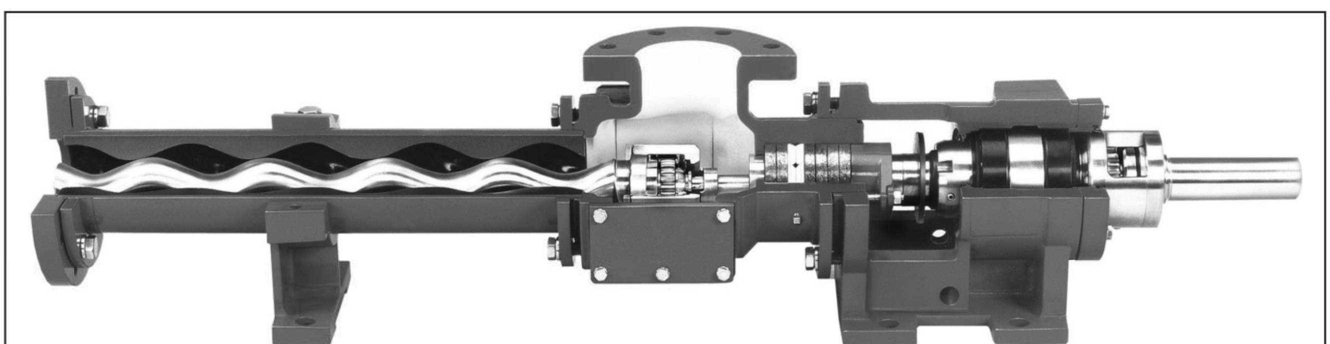
where H is the pump head in m, D_2 is the impeller diameter in cm, and B_2 is the impeller width, including back and front shrouds.

For the pump described in Figures 42 and 43, $D_2 = 27$ cm and $B_2 = 3$ cm. Furthermore, K is given by

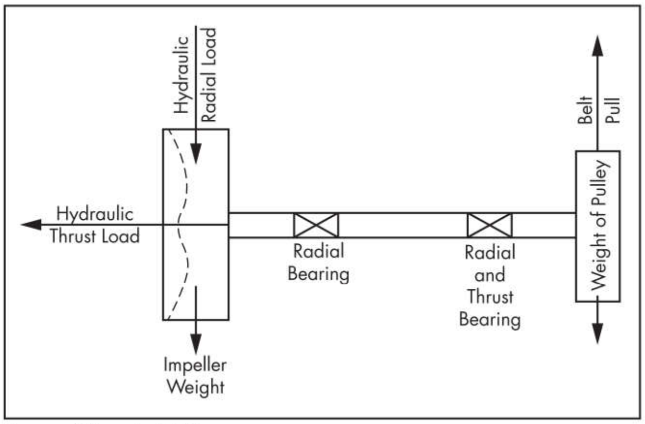
$$K = C\,[1 - (Q / Q_r)^2]$$

where C is a numerical constant that depends on pump geometry and specific speed, Q is the pump operating capacity in m^3/s, and Q_r is the pump rated capacity in m^3/s (Badr and Ahmed 2014).

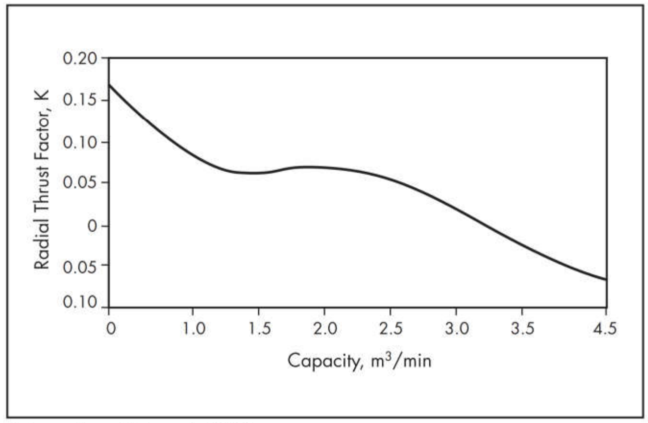
Pump designers may counteract thrust loads by using auxiliary vanes or expellers on the outside shroud surfaces of the impeller. Another alternative is to use a double-suction impeller, which is fed liquid from both sides. Finally, these



Courtesy of Carl Eric Johnson Inc.
Figure 40 Progressing cavity (Moyno) pump



Source: Link et al. 1985
Figure 41 Bearing loads in a pump



Adapted from Link et al. 1985
Figure 43 Radial thrust factor in a double-volute, 100-mm pump at 1,760 rpm

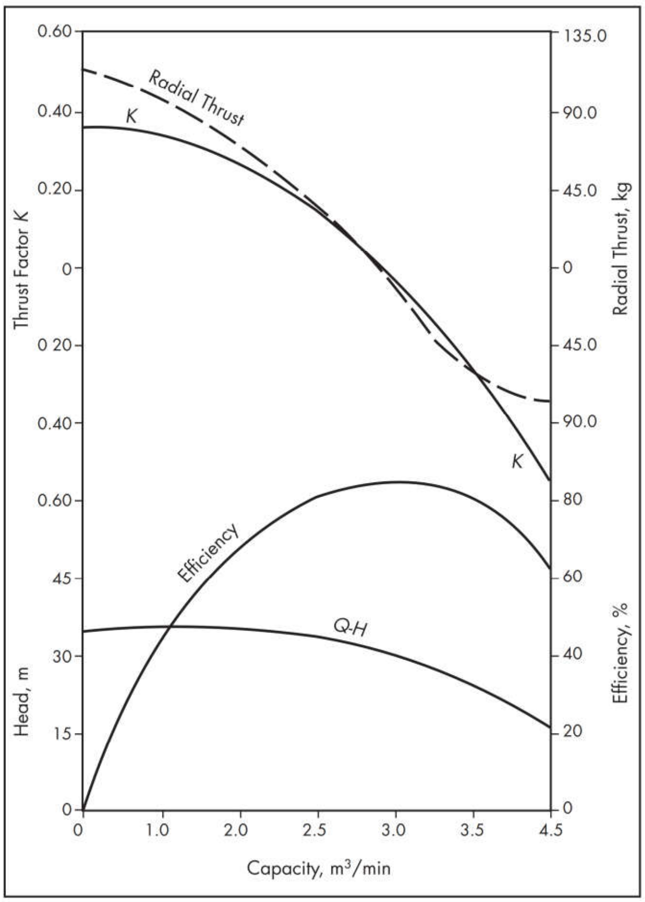


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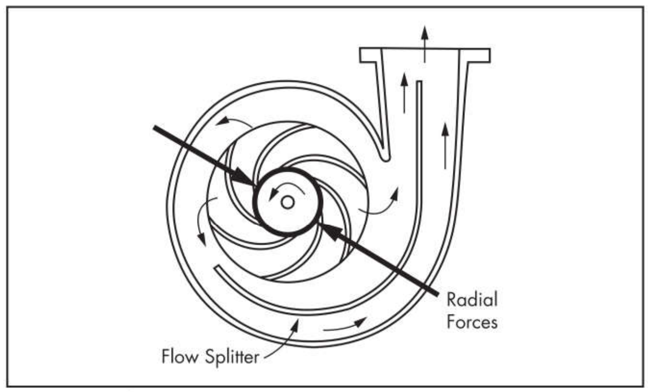
Figure 42 Radial thrust in a single-volute, 100-mm pump at 1,760 rpm



Source: O'Keefe 1972
Figure 44 Double-volute pump casing with flow splitter in discharge passage

forces are sometimes countered by employing a double-volute casing that uses a flow splitter, shown in Figure 44.

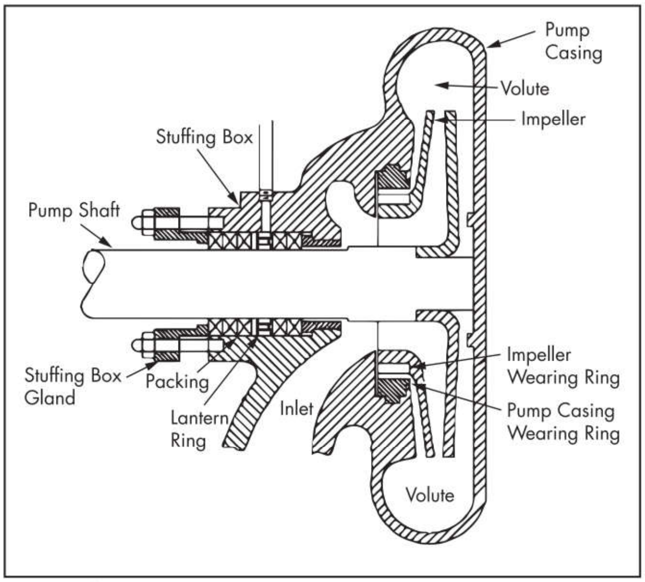
All loading imparted to the shaft is absorbed by the bearings. The radial loads are primarily absorbed by the set of bearings closest to the volute on an overhung impeller design, while the bearings nearer the driven end absorb radial and thrust loads. Distances between bearings and between the radial bearing and impeller must be properly established to minimize shaft deflections and stresses.

Lubrication of the bearings, whether by grease or oil, depends on the choice of bearings, running speed range, and service. In some elevated-temperature applications, water-cooled jacketed bearings are used.

Shaft Sealing

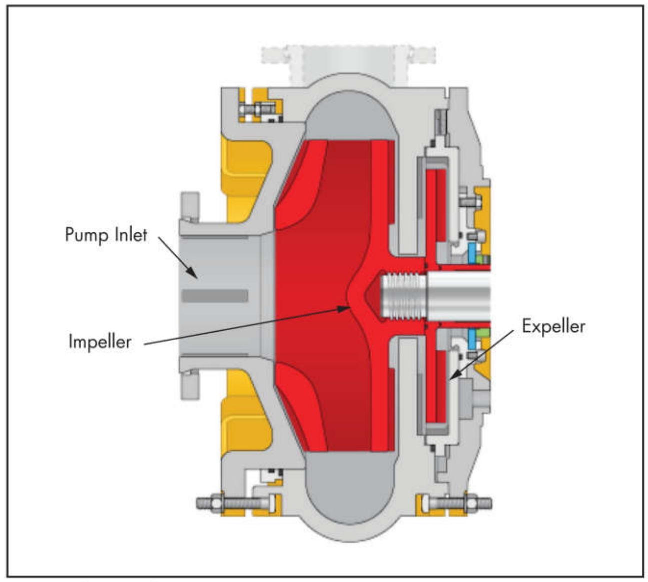
Shaft sealing is one of the most challenging areas of pump design and operation. Sealing at the volute or "wet end" of the pump is generally accomplished in a transition area known as the stuffing box, shown in Figure 45. One of the most common ways to seal the shaft is by using packing rings, which can be made from a wide selection of materials such as asbestos, graphite, carbon, polytetrafluoroethylene, and aluminum. The choice of packing depends on factors like pump discharge pressure, corrosiveness, and temperature of the liquid pumped; shaft diameter and speed; and the availability and type of packing lubrication.

Water or grease may be introduced into the packing area to lubricate and cool the packing. In the case of slurry pumps, water is introduced into the end of the stuffing box nearest the volute to prevent solids from entering the packing area. Such injection is normally made through a lantern-ring to ensure



Courtesy of Adamant Valves

Figure 45 Pump sealing arrangement, showing stuffing box and packing



Adapted from Wilfley 2014

Figure 46 Expeller installed in a centrifugal pump

even distribution of water around the shaft. A lantern ring is a perforated hollow ring located near the center of the packing box that receives relatively cool, clean water and distributes the water uniformly around the shaft, to provide lubrication and cooling. The fluid entering the lantern ring can cool the shaft and packing and lubricate the packing. It can also seal the joint between the shaft and packing against leakage of air into the pump when pump suction pressure is below atmospheric pressure. As packing wears, the gland follower is tightened in against the packing until such tightening ceases to be of value.

An alternate method of sealing the shaft is with mechanical seals, which may be installed as a single seal or double seal and may be connected internally or externally of a packing box. Mechanical seals are employed where sealing liquid to the pump must be limited or entirely omitted or where escape of pumped liquid is not permitted. The sealing faces of mechanical seals may be constructed from ceramics, carbon, and carbides as the application warrants. These seals are lubricated and cooled by sealing liquid. Successful operation with a mechanical seal depends on proper alignment, controlled shaft deflections, and a steady supply of clean sealing fluid.

Another method of sealing is the use of expeller vanes on the back shroud of the impeller, opposite the suction eye, as shown in Figure 46. These vanes are generally of a greater diameter than the pumping vane, and their purpose is to move liquid away from the stuffing box area of the impeller. These pumps often have packing that is grease lubricated, and the pumps are designed with a check valve or similar device to permit the entry of air, thus preventing them from forming a vacuum. In some pump designs, the expellers are aided by auxiliary vanes in the stuffing box area. The success of expeller-sealed pumps depends on the operating point on the pump curve and magnitude of the suction pressure at the impeller eye. Higher ranges of pump speed and low suction pressures are favored for successful operation.

Pump Drives

Electric Motors

Electric motors are the most common drives for most process equipment, including pumps. For a given pump application, the motor is usually selected based on the power requirement of the pump. Motors for centrifugal pumps are generally selected based on the normal operating duty point.

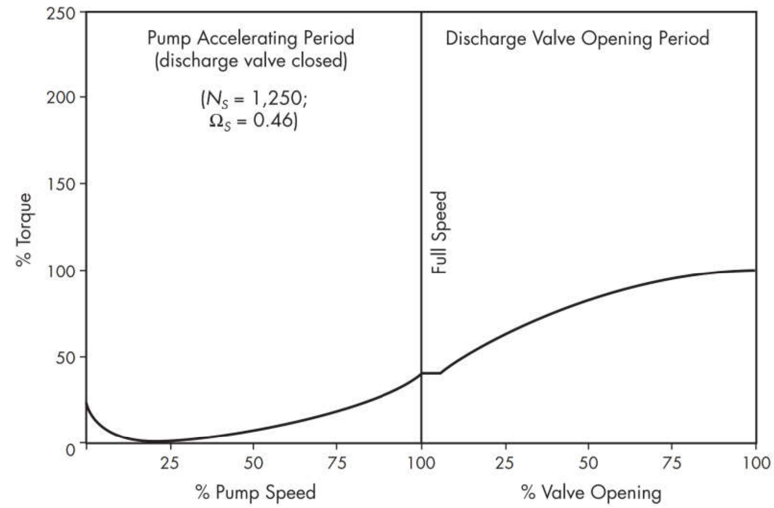
The use of variable-frequency drives (VFDs) for AC motors allows pump speeds to be easily varied, in response to process conditions. This capability provides many options for pump operation that were not previously available. The combination of VFD with an AC induction motor provides a drive system that will accommodate considerable variation in operating conditions, requires minimal maintenance, and provides line-to-load efficiency from 90%–97%.

In variable-speed applications, it may be necessary to either derate the motor or provide it with forced-air cooling. The motor cooling fan becomes less effective at lower than rated speed, and the available torque decreases at speeds greater than rated synchronous speed. Motor vendors provide detailed guides on how to select appropriate motors for variable-speed duty.

In centrifugal pumps, the relation between torque and speed is quadratic. Driving these pumps with a variable-speed electric motor generally has no complications. PD pumps may have a significant friction torque at zero speed or may present a constant torque with speed load to the driver. Selection of a motor and VFD for such a load must consider thermal derating (or forced cooling) of the motor as well as the current rating of the VFD.

Internal Combustion Engines

Portable pumps require a nonelectric source of power, as do pumps for critical-service systems such as fire protection backup. Internal combustion engines are the usual choice for



Source: Link et al. 1985

Figure 47 Torque–speed relationship in a start with a closed discharge valve

nonelectric power. Although selecting an electric motor based on power rating and pump speed is straightforward, the torque/speed characteristics of internal combustion engines are significantly different from those of an electric motor. There are also differences in the characteristics of gasoline and diesel engines. The pump speed and torque are used to select the correct internal combustion engine.

Matching Pump and Driver

It is very often useful to make a plot of pump speed versus required torque against a similar curve of speed versus available torque for the motor to determine the time required for the motor to accelerate the pump to full speed. Usually this is not a problem for most plant pumps using today's induction or synchronous motors; however, where there is reduced voltage starting or a relatively prolonged period of operation at low head or "runout" conditions, a decrease in static head analysis of speed–torque is desirable. This analysis also requires an assessment of combined inertia of the pump and driver rotor assemblies, referred to as the motor shaft. The conventional form is to plot percent pump torque on the vertical axis and percent full load speed on the horizontal axis. The pump torque required for any given head capacity is

$$T = (9{,}549 \times P)/N$$

where T is the pump torque in N-m, P is the power in kW, and N is the rotational speed in rpm.

At any speed between zero and full-load speed, FLS, the torque varies as the square of the speed. The curve derived from this is parabolic in shape. For example, at one-half the FLS, the torque will be $(½)^2$, or one-fourth the torque at FLS.

At zero speed, the theoretical pumping torque required is zero. To overcome the inertia of the rotating element, some torque is required, usually 5%–15% of the full-load torque. After the initial movement when the pump picks up speed, the torque requirement drops to follow the pump's hydraulic characteristic at about 10% FLS. Consider the following scenarios under which a pump may be started, and the speed–torque relationship in each.

Closed Discharge-Valve Start

When the pump is started with the discharge valve fully closed and its speed progresses from zero to FLS, the pump is operating in the shutoff condition at FLS and drawing shutoff power. The torque can be calculated from the power at shutoff. The discharge valve is then opened to its fully open position. The shape of the speed torque curve will vary, depending on the system characteristics and the type of valve. The inertia of the liquid in the pipeline will also have some effect on this portion of the curve. Figure 47 shows this case.

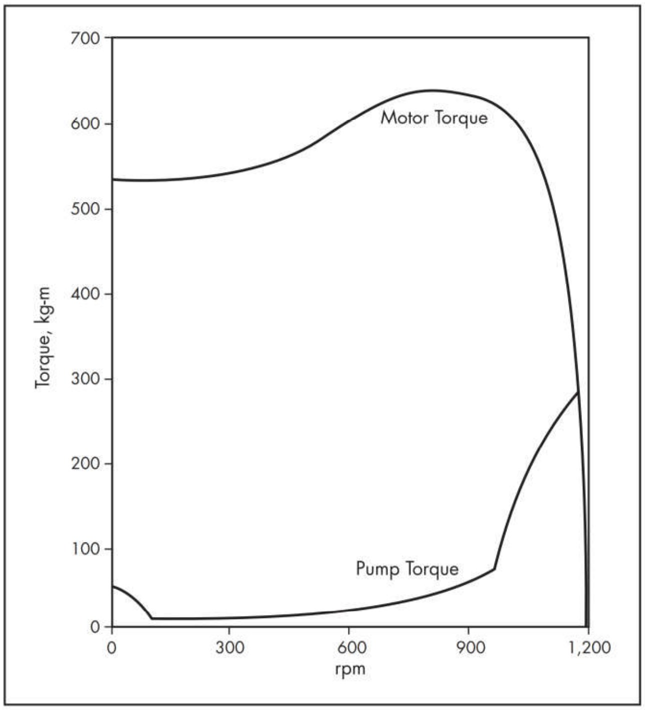
Open Discharge-Valve Start

With an open discharge-valve start, the speed torque curve depends on the head-capacity characteristics of the system into which the pump is delivering. In a system where all the head results from friction, the speed–torque curve is parabolic in shape and rises continuously. At 100% speed, the pump will be operating at the rated point and the torque will therefore be 100%. When there is a combination of static and friction head, the pump must initially generate sufficient head to overcome the static head of the system. This situation is the same as pumping with the discharge valve closed. At the speed where the pump overcomes the static head, the pump then delivers into the system and the torque will then rise to 100% torque at the 100% speed point.

When all the head is static, the pump initially operates as if the discharge valve is closed until all the static head is overcome and then delivers into the system along a constant head line until the operating point is reached.

Long Discharge Line

Another example of a virtually closed discharge-valve system is a long pipeline system where the mass of the liquid becomes so great that the time required to accelerate the liquid in the pipeline is much greater than the time required to bring the motor up to FLS. The pump torque during the starting period will approach that of a pump operating with a closed discharge valve.



Adapted from Link et al. 1985

Figure 48 Typical squirrel-cage motor curve shown with pump torque curve

In conclusion, if the speed at a given torque of the driver does not drop below that for the pump at the same torque, the pump will reach the operating point. An example of a typical motor speed–torque curve superimposed on a pump speed–torque curve is shown in Figure 48.

Speed-Reducing Systems

Where the pump shaft speed required at the duty point does not match the speed of the driver, it is necessary to provide a means of matching the motor and pump speed. In many cases, the impellers of centrifugal pumps for clear fluids can be trimmed to provide the duty point at an electric motor's operation speed. Trimming is not recommended for the impellers of slurry pumps, which are made of hard materials, so during pump start-up, the speed must be adjusted to achieve the duty point.

V-Belts

The design and construction of material handling pumps require that impeller speed adjustments must sometimes be made after installation, to match pump performance with system demands. At pumps of up to 225 kW, power transmission from the motor to the pump can usually be done efficiently and economically with V-belt drives.

As a rule, the selected ratio of driver speed to pump speed should not be too high. Some manufacturers recommend a maximum ratio of 3:1, while others suggest 4:1. As the ratio increases, the belt wrap angle around the driving pulley decreases, and the power capacity of the belt decreases proportionately. When the driver motor power exceeds 225 kW, V-belt drive selections must be closely evaluated to ensure that overhung loads and belt pull from the belt drive do not exceed pump and motor bearing limitations. At these higher powers, a gear reducer or variable-speed motor may be preferable to a V-belt drive.

Belt sheaves are often mounted to the shaft using a taper lock hub. This allows the sheave to be positioned correctly on the shaft and firmly attached without the need for a press. Sheaves should be dynamically balanced to minimize vibration. Belts can be used in matched sets to increase the power capacity of the drive. Belts should be periodically checked to ensure they remain correctly tensioned.

Maintenance and proper installation and alignment are keys to the successful operation of a V-belt drive. To select a V-belt drive, the following information is needed:

1. Power and speed of motor required (Selected motor power should be 10%–15% greater than the maximum anticipated pump power requirement to overcome losses in the belt drive.)
2. Required speed of driven unit
3. Drive service factor to apply to the motor, usually 1.2

Belt vendors provide catalog procedures for selecting the required belt section, number of belts, driving pulley diameter, power correction factors associated with the drive geometry and candidate belt lengths, and resulting shaft center distances.

Gear Reducers

When power requirements exceed 300 kW, gear reducers are used for power transmission. Gear reducers provide higher drive efficiencies and lower bearing than V-belt drives. The most commonly used gear reducer is a parallel-shaft, single-reduction unit. Gear reducers, motors, and pumps are mounted on a common "ladder-type" sub-base with a fixed high-speed coupling between the motor and gear reducer and a sliding low-speed coupling between the pump and gear reducer. The slide coupling allows for axial impeller adjustment as required by the pump.

When selecting a gear reducer for a specific application, the following factors must be considered:

1. Service factor required, usually 1.25 to 1.5
2. Power required, including service factor
3. Drive ratio (most applications require an exact ratio)
4. Shaft sizes for selection of couplings

Items 1 to 3 are used to pick a reducer that is capable of transmitting the required torque at the stated reduction. Equally important is the thermal power, which is the power (without the service factor) that a gear drive will transmit continually for three hours or more without overheating. Fans or a pump with a cooler may be used to achieve an adequate thermal rating.

Shaft Couplings

Shaft couplings transmit the torque from the driver to the pump. A wide variety of coupling designs is available, all of which are made to allow for misalignment between the driver and pump shafts. A hub is fitted to each shaft, and a flexible element transfers torque from one hub to the other while minimizing radial and axial forces that may arise from the misalignment as the shafts rotate.

Baseplates

Unless the pump incorporates a driver, as in a fully submersible unit, the bare shaft pump, its driver, and any speed-reducer components will require a baseplate to keep these items aligned and transfer the forces between them and the ground or structure on which they are mounted. Generally, bases are of welded steel construction. Jacking bolts are provided on the base frame to allow the bare shaft pump, its driver, and any speed reducer to be positioned relative to each other so that the shafts are aligned within the capabilities of the shaft couplings, or the belt sheaves are parallel to one another and the belt grooves are in the same plane.

Belt-driven pumps must also include a mechanism to move the pulley centers relative to one another to allow the belts to be tensioned correctly. This can be as simple as mounting the motor on threaded rods with adjusting nuts to set the parallelism of the shafts as well as their center distances, or by using proprietary slide rails or spring mounts, available from many vendors.

When a V-belt drive is used, the motor is often mounted to the side of the pump, with the motor shaft and the pump shaft axes in parallel and aligned, for convenient connection. When a gear drive is used, the motor and pump may be side by side, or shaft to shaft, depending on the gear drive configuration. Overhead mounting has been used for motors as large as 260 kW on adjusted tabletop-type bases that straddle the pump. Side mounting is preferred whenever possible, with the motor mounted on an adjustable slide rail base firmly bolted to a concrete foundation.

PUMP SELECTION FOR PLANT APPLICATIONS

Grinding Mill Circuit Pumps

One of the most severe pump duties in the minerals industry is the pumping of the primary mill discharge to cyclones or screens. A centrifugal pump is generally used and is normally required to handle slurries containing 30%–65% solids with top ore particle sizes ranging from 20 mesh to 25- and 50-mm lumps. In addition, the presence of tramp objects, such as lumber and pieces of grinding media, further aggravates pump wear and maintenance problems.

To lessen pump wear problems in this service, tramp protection should be used. This may include ball traps, trammel screens, sump screens, or magnets upstream of the pump suction. In very "dirty" circuits where there is no tramp protection, fully metal-lined pumps with metal impellers will probably give the most satisfactory service. Where tramp protection can be provided and maintained, successful pump service life can be obtained from fully rubber-lined pumps or from pumps with rubber shell liners and metal impellers. Regardless of the pump construction selected, it is imperative for good mill operation and economics that not only must the pump service life be of a reasonably long duration, but it must be reliably repeatable to permit a properly planned maintenance schedule.

Mill discharge pump duties may range from 3 to 300 m^3/min at TSH values from 8 to 40 m and slurry specific gravities to 1.8. In this application, the TSH includes not only static and friction losses, but also the cyclone pressure drop. Since TSH directly relates to pump speed and wear, any lowering of TSH values will result in a significant increase in pump life.

Pumps used for mill discharge should have a relatively large diameter so that a given flow and head can be attained at the lowest possible rotational speed and suction eye speed. Impeller tip speeds for both rubber and metal impellers should be kept as low as possible, preferably under 1,500 m/min. A pump for a mill circuit, however, should not be so oversized that the anticipated operating range is far from the BEP on the operating curve. Operation too far to the left (lower than 50% of BEP) decreases life of pump bearings, as a result of higher radial loading; operation at 75%–80% of BEP is recommended. In addition, wet end port life is decreased because of increased pump recirculation and prerotation of the slurry as it enters the pump suction. The effects of slurry prerotation can be partially offset using straightening vanes in the suction piping or pump suction throat.

Froth Pumps

Pumping of flotation froth can be difficult, especially when conventional horizontal centrifugal pumps are used. Uneven pumping usually results because the centrifugal action of the impeller leads to a separation of air and liquid, with an accumulation of air occurring at the impeller eye. Feed to the impeller eye is thus restricted until the sump level rises sufficiently high enough to compress the entrapped air, which can then be swept away. Where horizontal pumps are used, it is sometimes preferable to use a pump with a semi-open impeller that will be less likely to accumulate air in the pump volute.

Sometimes it is more effective to use vertical-type pumps without stuffing boxes. These pumps permit the ready escape of separated air. There are two versions: the vertical pump with integral tank and the sump pump. The former design consists of a casing placed under the tank that opens upward through a hole in the tank bottom. The shaft passes through the tank and into the hole, allowing clearance for the feed. This pump design requires a controlled inlet flow to the sump. The sump-pump design has its pump casing submerged in the liquid and permits feed to enter from both the top and underside of the casing. The impeller has operating vanes on both sides, and air escapes upward through the surrounding sump enclosure.

A variation of the sump-pump design has operating vanes only on the underside of the impeller, with sealing vanes on the impeller topside. The feed entrance is from the bottom only. The pump casing has two holes through which part of the liquid is sprayed back into the sump. These holes prevent the formation of the liquid ring into which air is trapped. Air release also occurs through an opening surrounding the shaft.

Even though special pumps have been devised to separate and eliminate air from frothy slurries, it is usually necessary to oversize froth pumps, whether horizontal or vertical, to handle the entering and exiting air volumes. This is usually done by applying a froth factor, an integer that is usually in the range of one to four. (Use of the froth factor is described in detail in Chapter 7.1, "Mechanical Flotation.") The actual volume of the slurry pulp is multiplied by the froth factor, and a pump is selected with a volute sufficiently large to handle the increased volume. A higher froth factor indicates that the froth is more highly aerated and, consequently, will likely be more difficult to pump. Application of the froth factor is based on the individual judgment of the process designer. The difficulty in designing froth pumping systems is not only a matter of determining the volume of air entrapped in the froth, but also assessing the ease of separation of air from the slurry. For example, in fatty acid and amine flotation processes, mineral binding to the air bubbles is weaker than that experienced in conventional flotation processes for sulfide ores.

Froth pumping systems should be designed with as low a required TSH as possible to limit the pump's rotational speed. Where horizontal pumps are used, deep, non-cylindrical sumps are recommended. Short, straight suction lines with fully open–closed isolation valves are desirable. Packed-stem valves that can admit air should be avoided. Destruction of froth prior to entering the pump is sometimes attempted using mechanical, chemical, or thermal means.

Thickener Underflow Pumps

Thickener underflow pumps are usually either diaphragm-type PD pumps, where required head and capacity are within the relatively low limits of such pumps, or centrifugal pumps, with their broad capability range. The conventional, gland-water-sealed style of centrifugal slurry pump is most commonly used; dry gland or centrifugally sealed types have the advantage of not re-diluting the underflow slurry, but such pumps are not designed to operate against the high positive static suction heads or high suction lifts usually associated with thickeners.

Pumping systems for thickener underflow applications generally fall into three classes. In the first class, the suction pipe is connected to the center bottom cone. The pump is usually located below the cone and horizontally close to it, in a tunnel or restricted space, but in some cases it may be as far away as ground level outside the thickener. In the second class, the pump is mounted on the thickener bridge some distance above liquid level, the suction pipe extending vertically downward to the bottom, at or near the center. In the third class, the pump is a vertical unit, usually placed in a center well in the thickener. Diaphragm pumps can be used only in the first class. In practice, variations are numerous.

When a pump pulls from the center bottom cone, the distance at which the pump can be located from the thickener's vertical centerline depends on many factors. The bottom cone area is congested, and suction piping arrangements are often complicated, as a result of having to find room for such fittings, valves, flush-out connections, and piping to the pump and spare pump. Since the particle size in the thickened slurry is normally fine, suction pipe velocity can be as low as 1.5 m/s. However, while this low velocity can help keep total suction losses down, there may still be large losses from pumping a high-gravity slurry through relatively small-diameter pipe. Also, under abnormal thickener conditions, the delivery of solids to the pump suction may be at a much higher rate than the design figure, or solids may slough off the accumulation at the bottom, coming down in surges that tend to plug the suction. All these factors make it important that the pump be placed as close as possible to the bottom cone.

Correction of a thickener installation with pump operating difficulties because of pump cavitation can be very difficult and expensive, so the NPSHA of the pumping system should always be checked against the pump's NPSHR as stated by the manufacturer. It is advisable to provide the data necessary for the pump manufacturer to make this check, based on its experience with similar applications. The NPSH check is particularly important when it is proposed to bridge-mount the pump, because with a conventional thickener, the pump centerline will be at least 1.2 m above liquid level. This can result in low NPSHA, even with the absolute minimum length of horizontal pipe following the vertical suction riser on the thickener centerline. Calculations in the design stage often indicate that the pump will not work without being dropped toward the liquid level, and it may even be found that no lift at all can be tolerated. Bridge mounting also requires a reliable means for priming the pump, such as a water-operated exhauster.

When the thickener has a center-column rake-lifting device and bridge-mounted pump, an inverted flexible loop is required in the suction line so it will clear the lifting gear. This introduces an additional factor into designing the pump system for reliable operation, as the loop may rise a few meters above the pump centerline. Here, an analysis similar to the calculation of NPSHA must be made to determine whether the absolute pressure at the high point in the pipe loop is great enough to exceed the vapor pressure of the liquid being pumped, thus preventing a loss of prime through the vaporizing of the liquid.

Placement of a vertical pump, either wet or in a dry well down inside the thickener center column, has obvious advantages over both the bottom-cone and bridge-mounted arrangements. However, vertical pumps available for this type service are of such size that they rarely will fit into the space provided by normal design of such items as the center well and rake drive gearing.

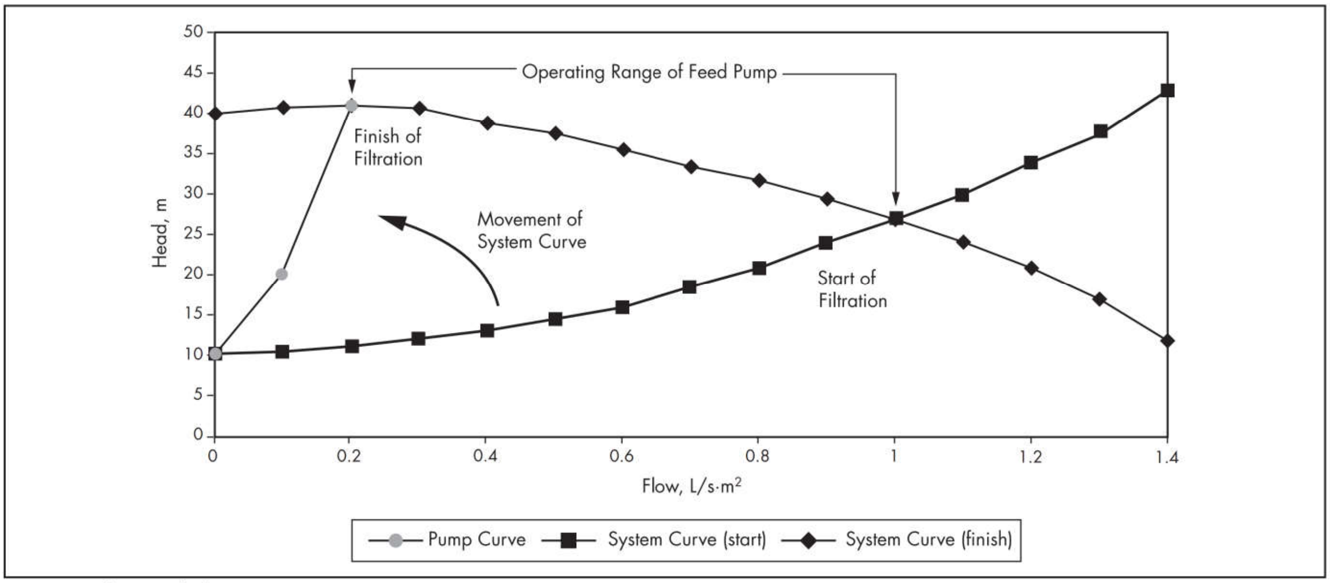
Use of the underflow pump for emergency emptying of the thickener is rarely practical. The normal capacity of an underflow pump is small, compared to the total slurry volume in a full thickener. Furthermore, most thickeners are so deep that pump capacity could approach shutoff, or zero delivery, before the thickener was empty, because of the significant increase in head on the pump.

As with all other slurry pump applications, every item used in thickener underflow pump calculations must finally be expressed in terms of pressure head of the slurry as pumped. In an ideal sump, the whole volume has a uniform specific gravity equal to that entering the pump suction, and it is assumed that this is the case on average actual installation, so the stated level in the sump can be directly used in computing static head. However, a thickener is specifically designed to induce sedimentation, and there can be considerable specific gravity variation between top and bottom with only the slurry near the bottom having a specific gravity as high as that passing through the pump. One method for accommodating this variation is to reduce the elevation of liquid in the thickener when calculating NPSHA. This has the conservative effect of decreasing static suction head in the case of a flooded suction and increasing static suction lift in the case of a negative suction. This correction is used when figuring either the total system head on the pump or the NPSHA of the system, the effect being rather critical on the NPSH figure that is so important in thickener applications.

As a further hedge against abnormal conditions that can occur during thickener operation, good practice calls for a factor of safety of 1.5 m between NPSHA of the system and NPSHR of the pump, versus a conventional specification of 0.7–1 m.

For most mineral processing applications, speed-regulated centrifugal pumps provide adequate control for the underflow pumping system. The high suction head on the underflow pump means that gland seal water is usually needed on a centrifugal pump. To prevent excessive dilution of concentrate slurries that are pumped to filtration stages, variable-speed peristaltic pumps have found increasing acceptance. These pumps can also handle dense slurries. The use of pulsation dampers on the discharge piping is required with this type of pump and is a critical design issue.

The use of gravity discharge for thickener underflows is common in the mineral processing industry. These systems



Source: Erickson and Blois 2002

Figure 49 Filter feed pump performance

involve large tonnages and are preferred because larger orifices can be used and plugging from tramp oversize is less likely. This gravity discharge approach can be used where the disposal of the slurry, usually tailings, is downhill from the thickener.

Filter Feed Pumps

A pressure filter is a batch unit, and the filling or filtration stage usually takes only about 20% of the total cycle time. For most mining and metallurgical applications, the filter feed pump runs only during filter feeding, and is started and stopped under control from the filter's control system, although in some cases the feed pump runs continuously, with a recycle at lower pump speed to the feed tank when the filter is not being fed. In systems with no recycle, the instantaneous flow rates during filtration are quite high. Centrifugal slurry pumps with gland seal water are the standard choice. Gland seal arrangements are best determined by the pump manufacturer, but they must account for the start/stop operation of the pump and the wide range of operating conditions. Where full-flow water flushing of the gland is provided, it can cause considerable slurry dilution between filling cycles, and solenoid control of gland water or reduced-flow sealing systems should be considered.

The pump runs against increasing head as the filter cake builds, as shown in Figure 49, and an understanding of the system curve is important for correct pump sizing. Once the chamber is about 50% full, the pressure will rise as the cake resistance increases, causing the operating point to move back up the pump curve. At the end of the filtration stage, the pump will be operating at close to full pressure and the flow will have decreased from 10% to 40% of the initial flow rate. Where the slurry is known to be highly abrasive, the maximum flow rate may have to be lower.

Where pump selection shows that flows greater than the recommended maximum are possible, a variable-speed-drive pump should be used. Pump speed may then be increased as the system head increases. To calculate the flow, the frictional losses for the pipeline and static head must be known. The resultant curve should then be drawn onto the pump curve for the proposed pump. The intersection of the system curve with the pump curve at the proposed speed will then determine the maximum feed flow rate. Where large feed tanks are used, the flow rate should be checked with full and empty tank levels. With flat system curves, there can be a significant change in flow with changes in the tank level.

Where there is significant variation in feed density, the flow with minimum and maximum densities should be checked. Once the maximum flow rate has been determined, the NPSHR should be checked against the NPSHA with the proposed suction pipe and tank design.

Filtrate Pumps

The type of filter determines the approach to the collection of the filtrate. The most important consideration is whether the filtrate flow is continuous or intermittent (batch). The filtrate is usually at atmospheric conditions for pressure filters and can often flow by gravity to the return circuit. However, for rotary drum and horizontal belt filters, the filtrate must be separated from the vacuum system. Design of vacuum pumps and associated equipment is outside the scope of this chapter.

The batch nature of pressure filters means that, like the solids, the filtrate flow will vary from a maximum at the start of slurry feeding to almost zero at the end of the feed cycle. If an air-blow step is included, there is likely to be very little filtrate entrained in the exhaust air stream. Thus the filtrate collection system must be sized for a range of conditions and must recover the fluid from what can be a very high airflow. At the start of the feed cycle, the filtrate can contain a significant quantity of suspended solids, so the filtrate is usually returned to a thickening step for the recovery of the solids. If the thickening stage is omitted, the quantity of the suspended solids must be understood prior to selecting the discharge point of the filtrate.

Process Water Pumps

Process water systems in mineral concentrators are usually closed-circuit recycled streams from which water is added to feed ore at the mill and recovered from the concentrates and tails by thickeners. Volume flow rates can be quite high, and the water is usually stored in lined ponds. This presents a challenging requirement of large flow rates and low available head, sometimes exacerbated by high elevation or high temperature.

A double-volute pump with a horizontally split casing will have a low NPSH requirement because of its large inlet area. However, the sealing rings between the impeller and casing may be prone to wear because of the fine solids in the process water. As an alternative, these pumps can be placed in a pit with suction lines that penetrate the wall rather than going over the top. This improves NPSHA, but does so at the risk of draining the pond should something go wrong.

Fire Pumps

Water is usually used for fire protection. Normal centrifugal pumps for water applications are suitable. Some jurisdictions require that any electrically powered fire pump be backed up by a pump driven by an internal combustion engine, in the case of power failure. Depending on application, foaming agents may be added to the water. This is the case for fighting liquid hydrocarbon fires. Foaming agents can be chemically aggressive in concentrated form. Vendors of fire suppression foam offer expertise in selecting materials of construction and foam-dosing systems for fire pumping systems where foaming agents are required.

Tailings Pond Pumps

Tailings pond pumps return tailings water from the pond to the mill for reuse. In many operations, the pumping load includes groundwater and rainfall in addition to the water that separates from the tailings during its residence in the pond. The rainfall rate at the site and the drainage area involved will determine the required pumping capacity.

As operations progress and tailings accumulate in the pond, the pond may be enlarged, and the location from which water is removed may move. In such cases, barge- or jetty-mounted vertical pumps may be used to allow optimal positioning of the pumps in the tailings pond. A pump barge is shown in Figure 50.

Vertical, cantilevered shaft pumps are often used in tailings ponds. Because these pumps have no submerged bearings, they are usually very reliable and require little attention beyond occasional greasing of the bearings. Where there is significant rainfall, the higher flow rates required may indicate a vertical, double-suction pump. The selection of a double-suction pump may permit the use of a higher-speed unit, but the selection may not be based on hydraulic considerations alone. If a vertical double-suction pump is mounted on a barge, it will work well as long as the water level in the pit is high enough. However, the water level in tailings ponds often varies seasonally or with changes in mill production. If this is the case, a top-inlet, vertical, single-stage pump may be better. A top-inlet pump mounted on a barge can pump the water level down without drawing the mud from the bottom. Of course, the higher capacity of the double-inlet pump will be lost. This analysis may be understood in terms of NPSH requirements.

Courtesy of Technosub.net

Figure 50 Pump barge in a tailings pond

A high-capacity, moderate-head pump may have a specific speed that requires a greater NPSH, which in this case means a greater submergence. When comparing NPSHA and NPSHR, the altitude of the installation and the temperature of the liquid must also be considered. In calculating NPSHA, altitude correction can be conveniently estimated as 0.3 m of head for each 300 m of elevation above sea level. Sometimes the hydraulic conditions will allow the use of either a top-inlet, single-stage pump or a vertical, double-stage pump. In such cases, lower cost and weight of the single-stage pump will make it the preferred option.

When pumps are mounted on a barge or a floating jetty, float design must be carefully considered for safety and convenience of operation. In addition to the weight of the pump and motor, a portion of the discharge line must be supported by the pump float. When the barge or jetty is large enough that workers may stand or walk on it, balance and stability under all conditions must be analyzed by a competent designer, with proper consideration of buoyancy and center of gravity. For example, if several workers gather at one side of the barge, it may overturn. The effects of wind and ice must also be considered.

Finally, seasonal or operational variations in pond levels that result in large variations in pumping rates must be considered in the design of the discharge line. If the friction losses in the discharge pipe are too high, addition of a second identical pump may not double the capacity of the system. In addition, as pipeline velocity increases, transients in the discharge line may damage the pipeline on shutdown. When significant variations in the required pumping rate will at times require much higher pipeline velocities, the preferred alternative is to install a second pump with its own discharge line. Of course, the additional costs must be considered.

PUMPS FOR CHEMICAL SERVICE

Chemical-service pumps are used in several mineral processing applications, including froth flotation, leaching, solvent extraction, and ore treatment in autoclaves and roasters.

Specifications

Pumps used for chemical service are different from those used elsewhere, primarily in the materials from which they are made. General-service pumps are usually made of cast iron, ductile iron, carbon steel, or aluminum- or copper-based alloys. Such pumps can be used for some chemical service, but most chemical pumps are made of stainless steel, Hastelloy, nickel-based alloys, or more exotic metals like titanium and zirconium. Chemical pumps are also available with liners of carbon, glass, porcelain, rubber, lead, and a wide range of engineering polymers, such as thermoplastics, thermosets, epoxies, and fluorocarbons. In every case, these special materials are used solely to eliminate or reduce the destructive effect of the chemical liquid on the pump parts. It therefore follows that the nature of the liquid being pumped determines which materials are used in the pump, and a careful analysis and description of that liquid is thus called for, including the following items:

- **Major and minor constituents.** In many instances, the effects of the minor constituents on corrosion rates will be equal to or greater than those of the major component.
- **Concentration.** Concentrations should be described quantitatively and as precisely as possible. General descriptions such as “concentrated,” “dilute,” or “trace quantities” should be avoided. Care should be used to clearly state whether concentrations are by weight or by volume. The concentrations of trace materials should be noted, even if they are in parts per million. In some cases, the presence of trace elements can markedly affect corrosion rates.
- **Temperature.** The operating temperature has a distinct effect on corrosion rates, so once again, operating temperature ranges should be described quantitatively, avoiding ambiguous, general terms. Maximum, minimum, and normal operating temperature should be specified. The effects of thermal shock should also be considered when the operating temperature range is large.
- **Acidity and alkalinity.** Acidity or alkalinity should be described in terms of pH. If the solution being pumped is subject to changing pH, during routine or upset conditions, this should also be noted. Some materials that are suitable for handling a given alkaline or acidic solution will not work for a solution whose pH changes during use.
- **Solids in suspension.** Erosion and corrosion of pump components are strongly affected by the solids in suspension and the flow velocity in the system. Solids should be specified quantitatively, again making sure to indicate whether by weight or by volume. See the “Slurry Pumping” section later in this chapter for more information on the effects of solids on pump design.
- **Aeration.** The presence of air in a liquid solution can significantly change the chemistry in a pumping system. For example, a nickel-molybdenum alloy pump is suitable for commercially pure hydrochloric acid, but if the acid becomes slightly oxidizing as the result of entrained air, the alloy will corrode. If the pump is self-priming, such aeration is likely. Furthermore, the presence of air will also affect NPSHR. The maximum amount of air a conventional centrifugal pump can handle is approximately 5% by volume.
- **Transferring or recirculating duty.** In a recirculating pump, the buildup of corrosion product or contaminants may damage the pump.
- **Continuous or intermittent duty.** Continuous or intermittent duty may have a counterintuitive effect on pump life. Intermittent duty can be more destructive than continuous duty if the pump retains some corrosive fluid during downtime, particularly if the presence of air increases corrosion. In such cases, it may be advisable to flush or drain the pump when it is not in service.

Pumps for Widely Used Chemicals

Sulfuric Acid

Sulfuric acid is used in many mineral processing applications, including froth flotation and leaching:

- **Dilution of commercially pure sulfuric acid.** When sulfuric acid is diluted with water, heat is evolved. When this dilution takes place in the pump transferring the acid, special materials are required for the pump components.
- **Sulfuric acid saturated with chlorine.** In a solution containing sulfuric acid and chlorine, the specific weight percentage of sulfuric acid determines whether the chlorine will accelerate corrosion. Concentrated sulfuric acid is hygroscopic and will absorb moisture from the chlorine, so with a sulfuric acid–chlorine solution containing more than 80% sulfuric acid, little corrosion will occur because dry chlorine is essentially noncorrosive. In this case, material selection can be made as if sulfuric acid is the only constituent. If the solution is saturated with chlorine but contains less than ~80% sulfuric acid, the material selection must be based not only on the sulfuric acid but also on the wet chlorine.
- **Sulfuric acid containing sodium chloride.** Addition of sodium chloride to sulfuric acid will result in the formation of hydrochloric acid, and a pump that is handling such a solution must be made of a material that will resist the corrosive action of both acids.
- **Sulfuric acid containing nitric acid, ferric sulfate, or cupric sulfate.** The presence of any of these compounds in sulfuric acid, in quantities of 1% or less, can make a sulfuric acid solution oxidizing, instead of reducing. They can therefore act as a corrosion inhibitor, thus allowing the use of stainless steel. However, the same compounds can act as accelerators for non-chromium-bearing alloys.

Nitric Acid

Nitric acid typically presents fewer problems than sulfuric acid. Nitric acid is strongly oxidizing, allowing the use of stainless steel in many applications. However, this same property limits the application of nonmetallics, and especially plastics. Solutions such as fuming nitric acid, nitric-hydrofluoric acid, and nitric-hydrochloric acid (aqua regia) produce more aggressive corrosion, and careful material selection becomes quite critical.

Hydrochloric Acid

Pumps for both pure and contaminated hydrochloric acid must be carefully designed. Ferric chloride, the most common contaminant, can cause the normally reducing solution to become oxidizing, completely changing the required materials of construction. The presence of only a few parts per million of iron in commercially pure hydrochloric acid can form enough ferric chloride to cause severe corrosion in materials like nickel-molybdenum, nickel-copper, and zirconium. Conversely, the presence of ferric chloride can make titanium completely suitable. Nonmetallic materials are widely used in pumps for hydrochloric acid systems.

Alkaline Solutions
Alkaline solutions such as sodium hydroxide or potassium hydroxide do not result in serious corrosion at temperatures below 93°C.

Organic Acids
Organic acids are much less corrosive than inorganic acids, but their particular properties must be considered carefully when evaluating pump materials.

Salt Solutions
Salt solutions are considered neutral and do not present a serious risk of corrosion. In process streams that are adjusted to a slightly alkaline pH, salt solutions are even less corrosive than when neutral. In process streams that are slightly acidic, the liquid will be considerably more corrosive.

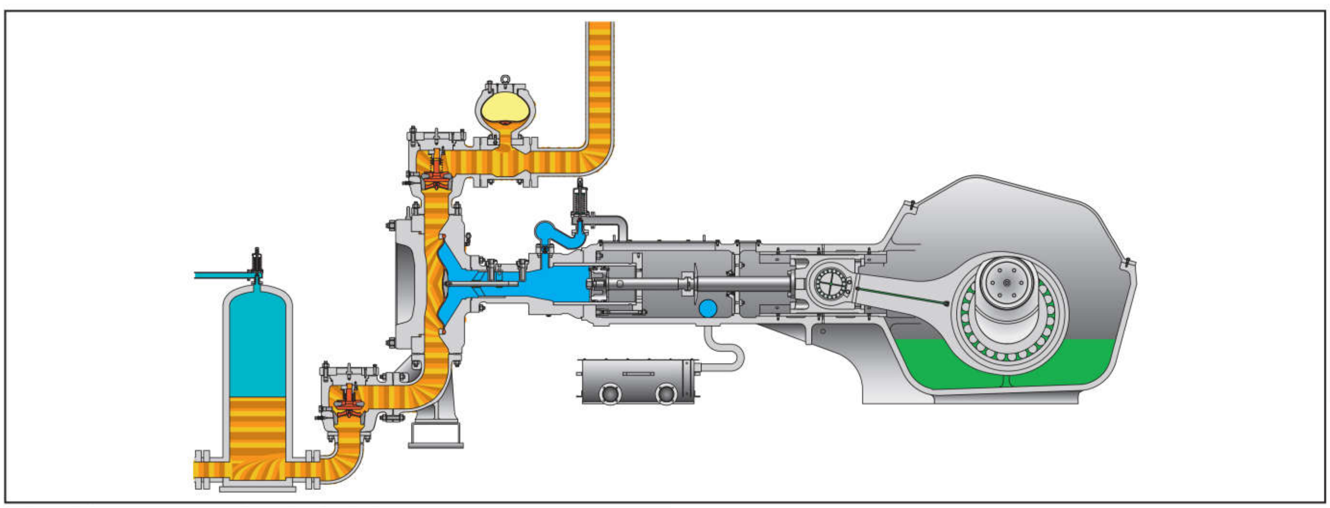
Organic Compounds
Most organic compounds are not as corrosive as inorganic compounds. Although this means that there will be more materials available to choose from when specifying a pump, each application should still be considered carefully. For example, chlorinated organic compounds and compounds that will produce hydrochloric acid in the presence of moisture will require resistant materials. Plastics provide excellent corrosion resistance to inorganic compounds within their temperature limitations, but they show less corrosion resistance to organic compounds.

Organic solvents used in solvent extraction are usually either flammable or combustible liquids. Here, pump construction techniques that minimize the risk of initiation and spread of fire are required. Generally, the American Petroleum Institute standards for pumps API 610 are overly restrictive for atmospheric pressure-settler tank solvent-extraction systems. Pump selection for solvent extraction circuits must consider the shaft sealing arrangement and materials of construction, as most aqueous phases in use are acidic or otherwise reactive.

Autoclave and Roaster Service Pumps
Pumps for service in processes employing autoclaves and roasters must be selected for the particular chemicals in the flowstreams being handled. Such pumps must be capable of pumping high-density slurry at the operating pressure of the systems. Usually, PD pumps of the type manufactured by GEHO, Toyo, and others are used. For example, GEHO series ZPM pumps are used at Newmont's whole-ore pressure oxidation facility at the Twin Creeks operation in Nevada, United States (Eichhorn et al. 2014). These pumps use a design that isolates the piston mechanism for the heated slurry, as shown in Figure 51.

Reagent Pumps
Reagents are usually added to a process based on a ratio or percentage of feed, requiring the flow to be metered. PD pumps have a known volume delivered per revolution or stroke and are thus suitable, in general, for reagent pumping. Helical rotor or lobe pumps are used for shear-sensitive materials such as flocculants. Peristaltic pumps are sometimes used for relatively low flow rates. Piston and diaphragm pumps can be provided with metering valves that allow accurate dosing of reagents and are typically used in water treatment applications.

Many types of chemicals are used as flotation reagents, and the selection of the reagent pump must account for the properties of the reagent it is pumping. For example, the frother MIBC (methylisobutyl carbinol) is considered explosive and requires a non-sparking pump. Furthermore, some frothers contain both an alcohol solvent and a glycol, and the alcohol can cause rubber and other polymers to deteriorate over time. In such cases, a piston-type PD pump would be preferred over a peristaltic pump. Other desirable features in reagent pumps include a totally enclosed, chemically resistant housing and completely enclosed electronics for protection against moisture, corrosive atmospheres, and damage from vibration or mechanical shock. Finally, pumps should have a simple and accurate mechanism for manual adjustment



Adapted from Weir Minerals Netherlands b.v. 2009
Figure 51 GEHO ZPM positive displacement piston pump

of the pumping rate, a large turndown ratio. They should be amenable to control through connection to the signal from an automatic control system, usually a 4–20 mA or pulse input.

It is often useful to install a reagent pump in a self-priming configuration. Some manufacturers provide auto-prime valves, which allow automatic degassing for reagents that are prone to gassing in the pumping circuit. The liquid handling portion of the pump should be carefully selected to allow safe and long-term handling of all the reagents whose use is anticipated in the circuit.

SLURRY PUMPING

The design of slurry systems is based on a combination of first principles, rules of thumb, practical operating experience, and common sense. Most in-plant slurry systems can be designed effectively using the present body of practical rules and data. However, design of largescale installations, where slurry is pumped over a long distance, usually require pipeline trials to ensure success. These are almost always justified because of the high costs of incorrect design.

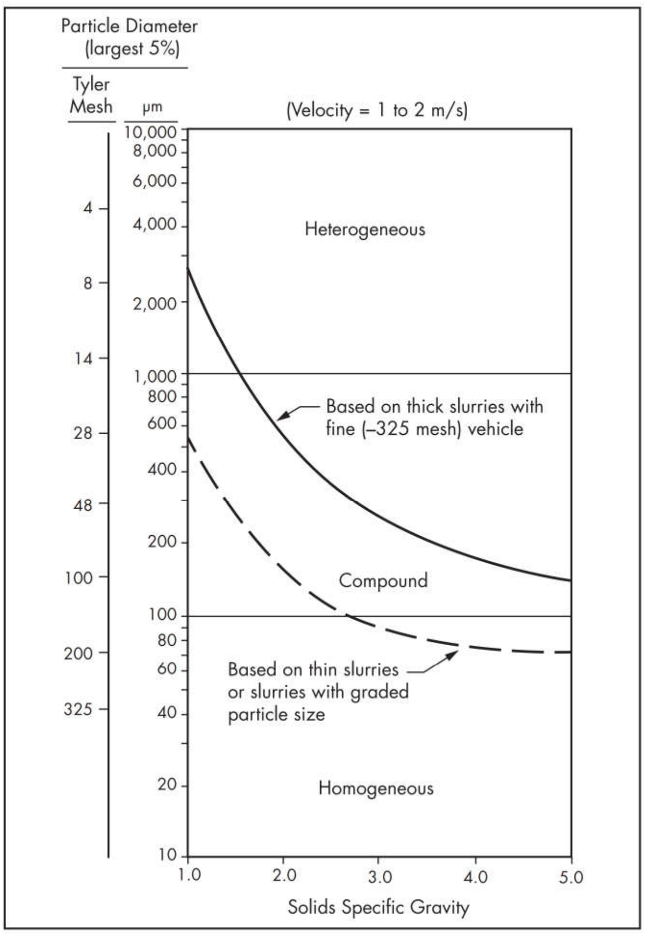
The Nature of Slurry Flow

Slurry flow in pipelines may be classified as homogeneous, pseudo-homogeneous, or heterogeneous. Homogeneous slurries are non-settling slurries that are characterized by a uniform distribution of solid particles throughout the carrier liquid and across the pipeline cross section. Typically, this type of slurry has a high concentration of fine solids (<30 µm) with low specific gravity and exhibits viscous behavior.

Heterogeneous slurries are settling slurries characterized by a nonuniform distribution of particles across the pipeline cross section. These slurries usually contain coarse, poorly graded particles that are carried along the pipeline bottom. Pseudo-homogeneous flow of such mixtures may occur, provided the mean flow velocity is sufficiently high to suspend particles uniformly in the pipeline. At a lower velocity, saltation or moving bed flow may occur where the largest particles are alternately dropped out and picked up by the carrier liquid and consequently bounced along the pipe bottom. Figure 52 shows conceptually the relationship among particle diameter, flow velocity V, and solids specific gravity in determining the characteristics of slurry flow. This type of flow has a narrow velocity range and gives the lowest friction losses.

Still lower velocities result in fixed bed flow where all particles are moving in one body in a sliding action along the pipe bottom, which leads to frequent plugging and very high pipe wear, and therefore should be avoided. Usually a practical design in heterogeneous slurry systems is to use a velocity about 0.3 m/s higher than the transition velocity between moving bed and pseudo-homogeneous flow. A combination of homogeneous and heterogeneous flow is also possible where fine particles flow homogeneously and coarser particles flow heterogeneously.

Flow in vertical pipelines approaches homogeneous flow for most solids suspensions, provided that the velocity of the carrier liquids exceeds the terminal settling velocity of the coarsest particle size.



Adapted from Scott and Hays 1985

Figure 52 General guidelines to slurry flow regimes

Slurry Pumping System Design and Operation

The design of a slurry transport system begins with an identification of the slurry to be conveyed. Particle size distribution and concentration must be known, a carrier liquid selected, and a minimum practical carrying velocity identified.

Friction losses must be determined as a function of pipe diameter and velocity. Selection of the final pipe diameter will result from an economic balance weighing such factors as initial pipe cost, pipe wear as a function of slurry velocity, solids abrasiveness, slurry corrosiveness, and projected operating costs, which arise mostly from power requirements. The power requirement is a function of system resistance and slurry specific gravity.

System design includes the following: selection of pump type (centrifugal or positive displacement), drives, and drivers; selection of control system and degree of automation; and sizing and layout of sumps and piping. Proper equipment selection and installation are necessary to minimize maintenance costs. Estimation of maintenance costs is often the most difficult task in preparing an operating budget.

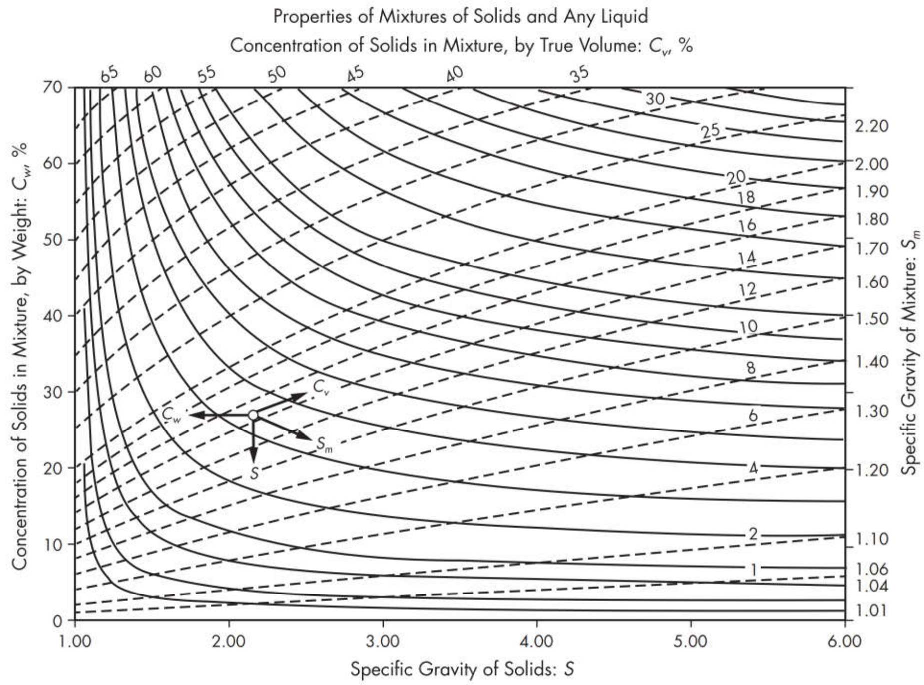
Slurry Flow Calculation

In most mineral processing plants, the slurries are mixtures of fine solids and water. Some basic equations used in the analysis of these applications are as follows:

$$S_{SL} = 1 + C_v/100\,(S_S - 1)$$
$$= 1/\{1 - [(C_w/100) \cdot (S_S - 1)/S_S]\}$$

$$C_v = 1/\{1 + S_S\,[(1/(C_w/100) - 1]\}$$

$$Q_{SL} = (M_S/60)/(S_S \cdot C_v)$$



Source: Grzina et al. 2002

Figure 53 Graph for calculating properties of slurries

$$V_{SL} = Q_{SL}/\pi(D/2)^2$$

$$P = (H_{SL}\, S_{SL}\, g)/1{,}000$$

where S_{SL} is the slurry specific gravity, C_v is the percent solids concentration by volume, S_S is the solids specific gravity, C_w is the percent solids concentration by weight, Q_{SL} is the slurry volumetric flow rate in m^3/min, M_S is the solids flow in t/h, V_{SL} is the slurry velocity in m/min, D is the inside diameter of pipeline in m, P is the pressure in kPa, H_{SL} is the head in m, and g is the acceleration due to gravity. These calculations may also be done graphically, using the graph shown in Figure 53.

Design Velocity and Head Loss

In the practical design of a slurry transport system, the designer must choose a velocity range in which to operate. After this determination is made, a pipeline diameter is chosen, the corresponding pressure drop evaluated, and pumping equipment selected. Unfortunately, there are no universal rigorous calculation procedures that can be relied upon to yield velocity and pressure drop information. The designer must rely on previous plant operating data or existing empirical correlations. The latter are, as one may expect, more highly developed for homogeneous slurries than for heterogeneous slurries.

Hydraulics for Homogeneous Slurries

Settling velocity is the rate at which a particle of a given size will settle in a given fluid. It is influenced by particle size, particle shape, particle and fluid specific gravity, and fluid viscosity. Settling velocity must not be confused with deposition velocity. In a pipe with slurry at rest, all solids are settled at the bottom of the pipe and the liquid is at the top. As pumping starts and water velocity increases, the water picks up progressively more solids until, at a certain velocity V_c (at any solids concentration), the last solids at the bottom of the pipe are on the point between moving and staying put. If the flow velocity in a pipe is below V_c, some of the solids will begin to settle in the pipe, and the slurry will become heterogeneous, or settling.

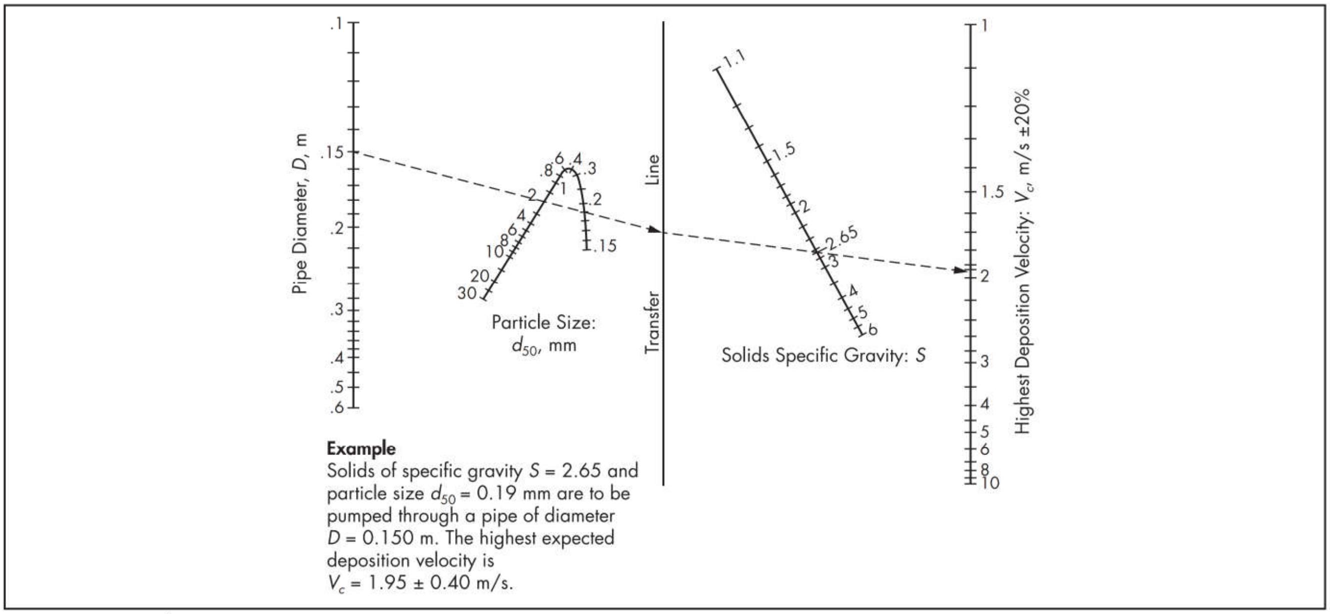
When handling slurries, one must first determine if the slurry is heterogeneous (settling) or homogeneous (nonsettling). If the slurry is heterogeneous, then care must be taken to operate above the settling velocity of the solids, which dictates a maximum pipe diameter for a given required flow. Wilson (1979) produced the nomogram shown in Figure 54 to estimate the deposition velocity V_c as a function of pipe diameter, particle size, and solids density. If the slurry is of the settling type, a determination of the settling velocity must be made to ensure operation above it to prevent solids deposition and possible pipe plugging.

Durand (1952) derived a formula that estimates settling velocity, V_L, as follows:

$$V_L = F_L \sqrt{\frac{2gD(S - S_1)}{S_1}}$$

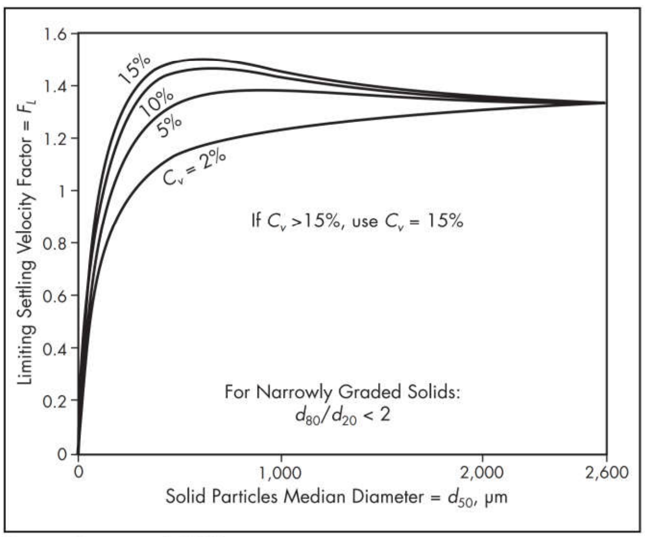
where F_L is a settling velocity parameter dependent upon particle sizing and solids concentration, g is the acceleration due to gravity (9.81 m/s^2), D is the pipe diameter in m, S is the specific gravity of the solids, and S_1 is the specific gravity of the liquid.

For closely graded particle sizing, F_L is obtained from Figure 55. Closely graded slurries are those in which the ratio of particle sizes does not exceed 2:1 for at least 90% of the weight of the solids in the sample, as described by the expression $d_{80}/d_{20} < 2$ in Figure 55. Figure 55 is conservative for slurries with more coarsely graded solids and significant portions



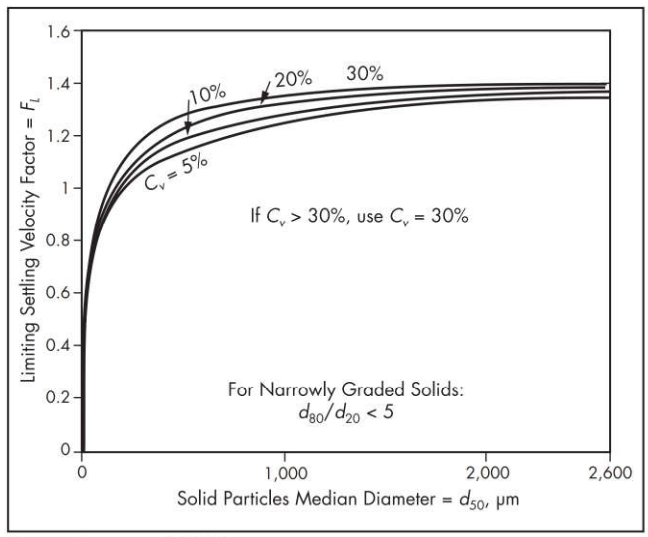
Source: Grzina et al. 2002

Figure 54 Nomogram for determining deposition velocity by Wilson's method



Source: Grzina et al. 2002

Figure 55 Durand's limited settling velocity parameter for closely graded particles



Source: Grzina et al. 2002

Figure 56 Modified Durand's limited settling velocity parameter for widely graded particles

of particles finer than 100 μm. For more widely graded particle sizing, Figure 56 is used to estimate F_L. The value of F_L from Figure 55 or 56 is then used to estimate V_L, thus

$$V_L = F_L\sqrt{2gD(\rho - 1)}$$

where D is the inside pipe diameter in m, ρ is the solids specific gravity, and g is the acceleration due to gravity in m/s^2.

For a heterogeneous slurry, accurately predicting the settling velocity is perhaps the first most critical factor in the design of a slurry pumping system. The settling velocity determines the required pipe diameter for a given desired flow, which is then used to determine the pipe friction. This information is used to make the appropriate pump selection and determine the proper operating speed, the appropriate materials of construction, and the expected power draw.

Performance Derating

The performance curves published by centrifugal slurry pump manufacturers are based on pumping clear water. Many manufacturers suggest that these curves be derated when handling slurries. Arguments for derating are based on considerations that the kinetic energy imparted to the solids by centrifugal action of the impeller cannot be recovered as pressure energy and that slurries exhibit viscous effects that lead to a loss in pump performance. Some pump manufacturers indicate that head and efficiency curves be derated by 5%–10% for all slurry applications, while others propose a more detailed derating procedure based on mean particle size, particle specific gravity, and solids concentration. One such derating procedure is described in the "Heterogeneous Slurry" section.

The operating point for a centrifugal pump is dependent not only upon the pump speed and the system in which it is placed, but also upon the material being pumped. Just as pipe friction can vary with the concentration of solids and the viscosity of the fluid being pumped, pump head output and efficiency can also deviate from the performance expected with water. It is desirable to be able to accurately predict these deviations, or deratings, in order to be able to determine if a centrifugal pump is the appropriate pump choice and to be able to compensate for these deratings by making the appropriate speed changes and providing adequate motor power.

Slurry Classification by Rheology

In addition to classification by deposition velocity, slurries may also be classified by rheological characteristics. A rheological classification is necessary in calculating the appropriate head and efficiency derating so that the appropriate corrections may be applied. Most slurries moved by centrifugal pumps fall into one of the following three categories:

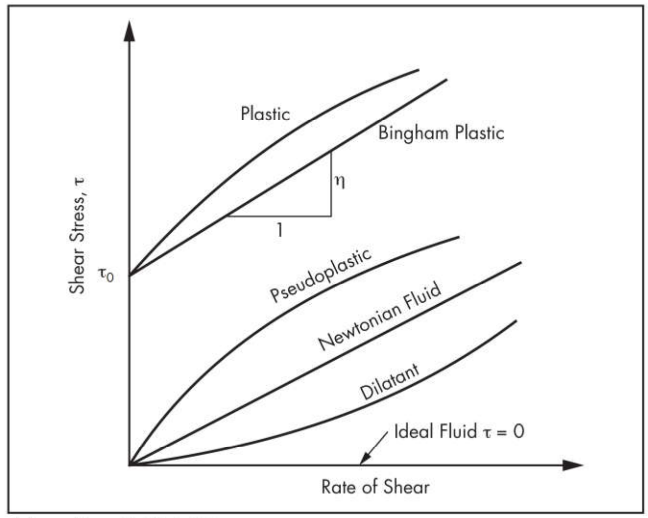
1. **Heterogeneous slurry.** Most solids handling slurries are heterogeneous. This rheological class corresponds directly to the "heterogeneous" class determined by deposition velocity. A heterogeneous slurry is a settling slurry and is typically water based, with most of the solids greater than 100 µm. A relatively low content of solids less than 100 µm results in the carrier fluid, which is the water plus the suspended fines being essentially similar to water.
2. **Viscous Newtonian.** This refers to any slurry or fluid that has no yield stress and a constant viscosity. For this discussion, consider a viscous Newtonian slurry to be one that has a viscosity greater than water but is still free flowing, albeit at a slower rate. Oil is perhaps the most common viscous Newtonian fluid.
3. **Bingham plastic.** In a Bingham slurry, the carrier fluid contains sufficient fines content to provide a yield stress. Bingham plastic fluids are not free flowing unless there is sufficient force to overcome the shear stress. Examples include ketchup, red mud, and concrete.

Figure 57 illustrates the rheological differences among the three slurry types (Wells 1991). For the sake of this discussion, it is assumed that the heterogeneous slurry has a stress-versus-strain plot similar to that of water. For reference, these plots of shear stress versus shear strain are called rheograms. Lab analysis of a small sample of representative slurry for rheological properties and solids composition can provide the most helpful information for the design of pumps and pumping systems.

Heterogeneous Slurry

Following are the principal reasons for derating centrifugal pump performance when handling solids:

- Slip between the water and the solid particles during acceleration and deceleration of the slurry as it passes through the impeller. This results in energy losses.



Adapted from Wells 1991

Figure 57 Rheological properties of various fluids

- Increased friction losses. These losses increase with increased solids concentration of the slurry.
- Inability of the suspended particles to store or transmit pressure energy.
- Mechanical friction changes in the gap between the impeller and side walls, which affect energy consumption and thus pump efficiency.

When discussing centrifugal pump deratings for heterogeneous slurries, the terms *head ratio* (HR) and *efficiency ratio* (ER) are used:

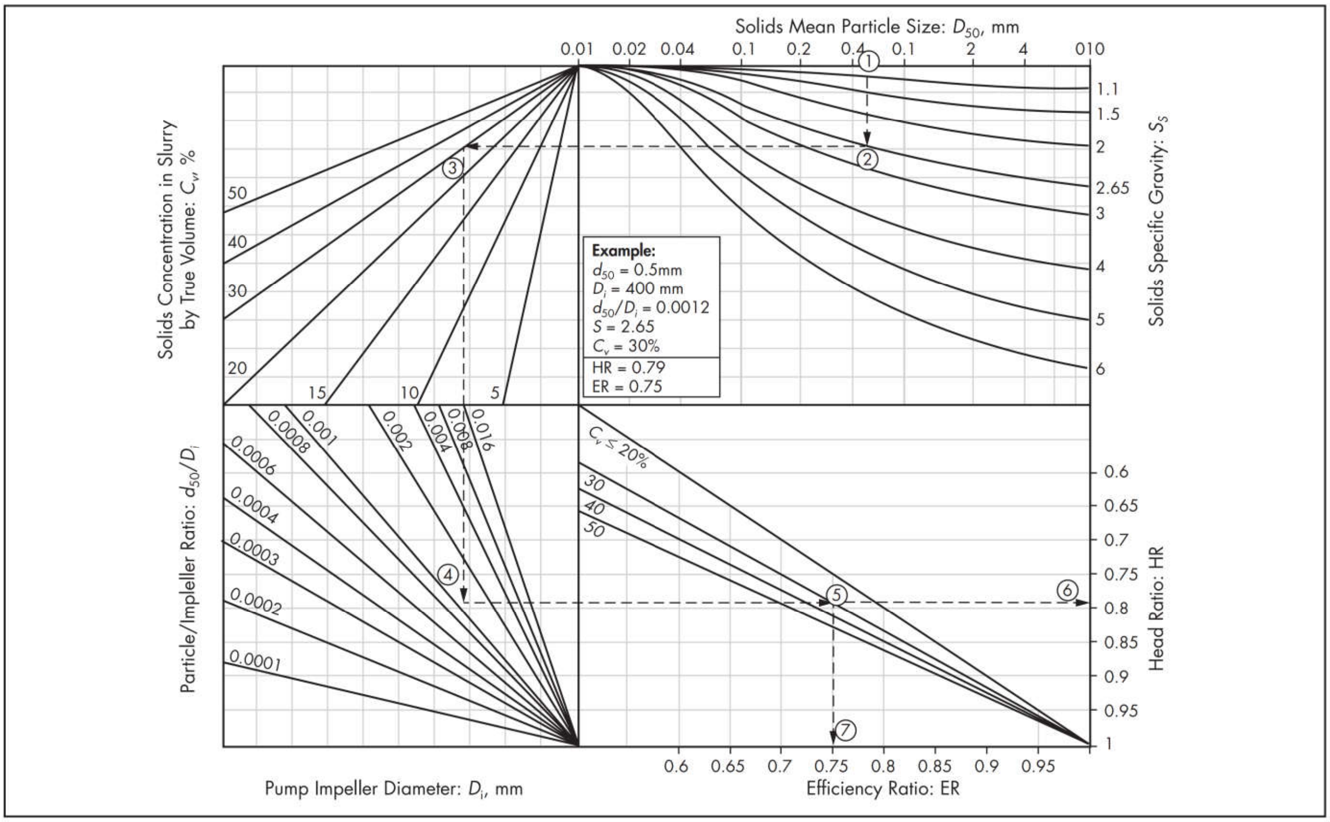
$$HR = head_{slurry}/head_{water}$$

$$HE = efficiency_{slurry}/efficiency_{water}$$

Grzina et al. (2002) explain showed that the degree of derating depends upon the solids specific gravity, the percent solids by volume, and the relative particle size (d_{50} particle size/impeller diameter). The results of this work are summarized in the graph in Figure 58. This graph is used by following the dashed lines that connect the circled numbers:

1. Start at the d_{50} particle size for the slurry.
2. Follow the line down to the solids specific gravity, S_s.
3. Follow the line left to the slurry % concentration by volume, C_v.
4. Follow the line down to the ratio of the d_{50} particle size to impeller diameter, d_{50}/D.
5. Then follow the line to the right, again to the C_v, but this time in the fourth quadrant.
6. From point 5, follow the line leading right to point 6 to find the head ratio, HR, and follow the line leading down to point 7 to find the efficiency ratio, ER.

Using the derating example given in Figure 58, consider a slurry that is 30% solids by volume, 2.65 specific gravity dry solids, 1,500-kg/m^3 slurry specific weight, and 350-µm d_{50} average particle size). The derating of a 6/4 pump at 1,200 rpm with the same 365-mm-diameter impeller is shown in the hypothetical system curve and plotted in Figure 59. In this example the intersection of the 1,200 rpm water performance



Adapted from Grzina et al. 2002

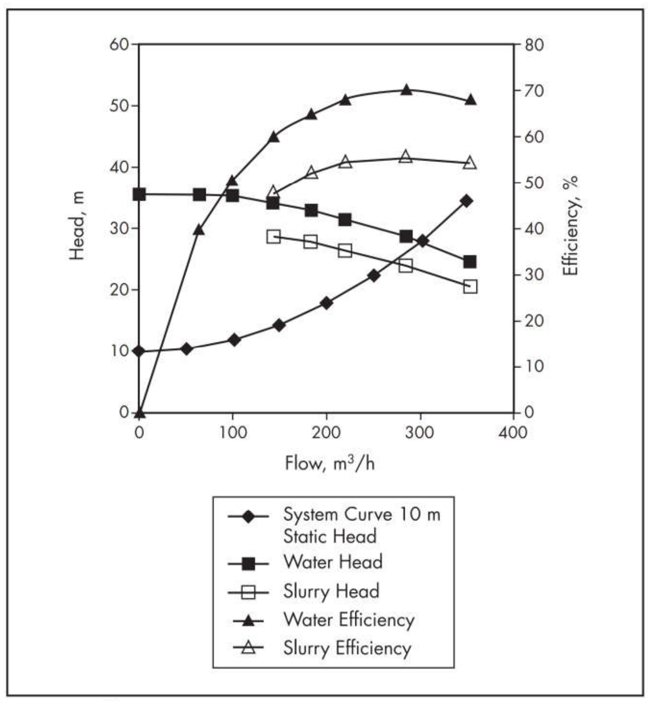
Figure 58 Head and efficiency ratios for pumping slurries

curve and the system curve occurs at 300 m^3/h, 28 m head, and 70% pump efficiency. With slurry correction (0.84 HR and 0.80 ER), the intersection of the derated pump performance curve and the system curve occurs at only 275 m^3/h at 24.5 m head and the pump's efficiency has been derated to 56%. The derated pump curves (for slurry) are shown in Figure 59. Power draw at a slurry flow of 275 m^3/h is expected to be

$$\text{kW}_{\text{draw}} = [(275 \text{ m}^3/\text{h})(1 \text{ h}/3{,}600 \text{ s})(24.5 \text{ m}) \times (1{,}500 \text{ kg/m}^3)(9.8 \text{ m/s}^2)] / [(56\%/100)(1{,}000 \text{ W/kW})] = 49.1 \text{ kW}$$

For operation at 300 m^3/h in this system, instead of the derated 275 m^3/h referred to previously, the required pump water head would be greater than the 28-m system head at 300 m^3/h by the inverse of the head ratio. This means the pump would have to deliver 28-m head/0.84 head ratio = 33.3 m head on water to achieve a 28-m head on slurry at 300 m^3/h in this system. Using the affinity laws or referring to the manufacturer's characteristic curve indicates this occurs at approximately 1,290 rpm with a water efficiency of ~70%. Derating the water efficiency with a 0.80 ER yields 56% slurry efficiency and the following power draw at 300 m^3/h:

$$\text{kW}_{\text{draw}} = [(300 \text{ m}^3/\text{h})(1 \text{ h}/3{,}600 \text{ s})(24.5 \text{ m}) \times (1{,}500 \text{ kg/m}^3)(9.8 \text{ m/s}^2)] / [(56\%/100)(1{,}000 \text{ W/kW})] = 61.3 \text{ kW}$$



Source: Bootle 2002

Figure 59 Effect of slurry correction on pump performance

Newtonian Viscous Slurry

Purely viscous Newtonian fluids are not very common in slurry pumping. In most instances, the carrier fluid is water with a small percentage of fines in suspension that has a viscosity similar to water. In these instances, when most of the particles are larger than 100 µm, the pump deratings come from the slip and friction caused by the relatively large particles, and the heterogeneous slurry correction discussed previously is recommended.

Recall from Figure 55 that particles smaller than 100 µm with the appropriate specific gravity will be non-settling and remain in suspension. These fines effectively increase the density of the carrier fluid, and if they are there in sufficient quantity, the carrier fluid exhibits a yield stress—it will not be free flowing.

Only when there is no yield stress and when the slurry has a viscosity different from that of water should the traditional viscosity correction charts be used. This indicates the benefit of testing a sample of the slurry for its rheological properties to determine its slurry type.

Bingham Plastic Fluids

A Bingham plastic fluid or slurry is a material that has a yield stress. This means it is not free flowing unless there is sufficient stress to overcome the yield stress. Bingham plastic fluids are often called pastes or paste slurries. As mentioned previously, when non-settling fines are present in a slurry, they effectively increase the density of the carrier fluid. At a certain fines concentration (typically 30%–50% by weight, dependent upon material), the slurry starts to exhibit a yield stress. The addition of flocculants and coagulants can also be used to develop slurry that has a yield stress. Figure 60 is a rheogram of a red mud slurry with Bingham fluid characteristics.

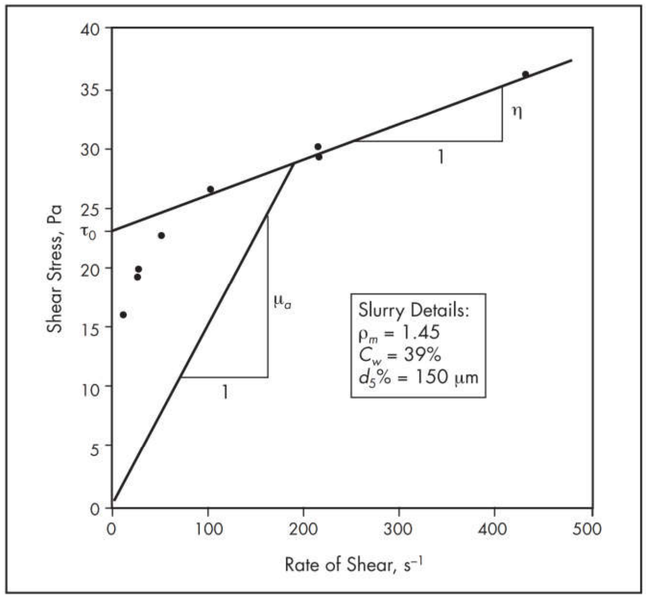
Important parameters for a Bingham plastic fluid are the yield stress, t_0, and the coefficient of rigidity, η, which is also referred to as the plastic viscosity, μ_p. The plastic viscosity is the slope of the shear stress versus shear rate line at high shear rates. The yield stress is the point along the y-axis where this straight line intersects the y-axis. For the example in Figure 60, the yield stress is approximately 23 Pa, and the plastic viscosity is calculated as

$$\mu_p = (38 - 0)\ \text{Pa}/(500 - 0)\ \text{s}^{-1}$$
$$= 0.03\ \text{Pa·s} = 30\ \text{centipoise}$$

It is important to differentiate between apparent viscosity, μ_a, and plastic viscosity, μ_p. For a Newtonian fluid, there is no yield stress and the viscosity is constant. Thus, its rheological plot is a straight line starting at the origin with the slope equal to the viscosity, as shown in Figure 60. A similar straight line can be drawn from the origin to any point on the rheological plot of a non-Newtonian, Bingham plastic fluid. The slope of this line will be the apparent viscosity for the particular shear rate and shear stress. The apparent viscosity is dependent upon the chosen point. At other shear stresses and rates of shear, the apparent viscosity will differ. For the red mud slurry described in Figure 60, at a shear rate of 200 s^{-1},

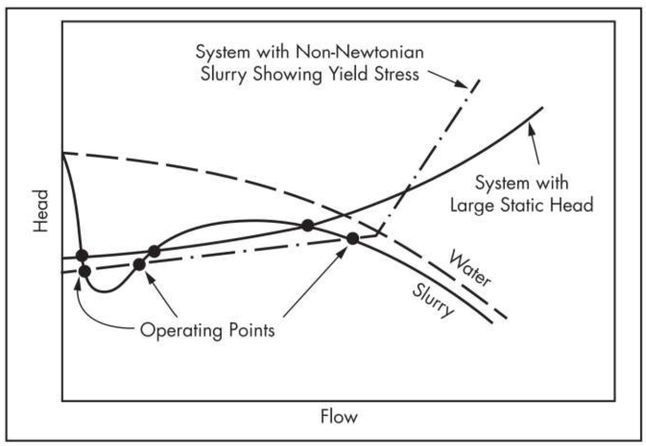
$$\mu_a = (29 - 0)\ \text{Pa}/(200 - 0)\ \text{s}^{-1}$$
$$= 0.145\ \text{Pa·s} = 145\ \text{centipoise}$$

while at a shear rate of 300 s^{-1},



Source: Bootle 2002

Figure 60 Rheology of a red mud slurry with 95% of particles smaller than 150 µm



Adapted from Bootle 2002

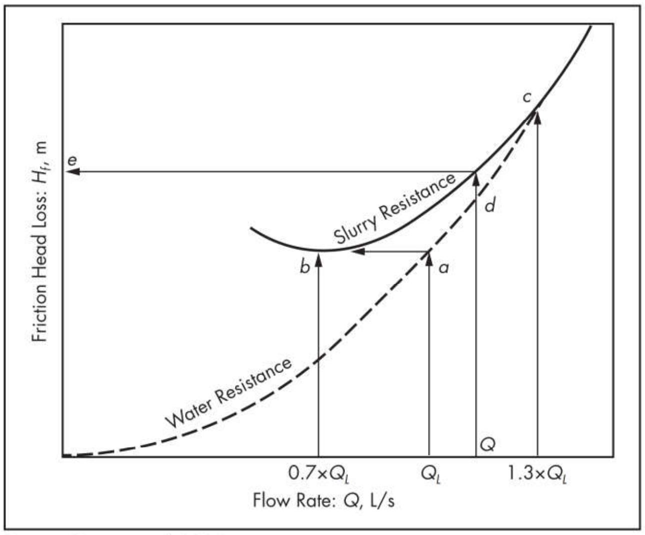
Figure 61 Effect of high yield stress on pump performance

$$\mu_a = (39 - 0)\ \text{Pa}/(300 - 0)\ \text{s}^{-1}$$
$$= 0.076\ \text{Pa·s} = 76\ \text{centipoise}$$

Thus the apparent viscosity decreases with shear rate, so this slurry is classified as *shear thinning*.

Walker and Goulas (1984) performed a series of tests with a 76-mm pump using kaolin clay and coal slurries of varying yield stress and plastic viscosity. The tests included three impellers, each at a different specific speed. On high yield stress material (~10–20 Pa), a severe drop in head occurred at low flow, as shown in Figure 61.

The significance of this drop in head is that the pump performance curve crosses over the system curve at three points. For comparison, Figure 62 also shows typical curves for two systems: a typical, turbulent-flow, exponentially rising system curve with high static head, and a high yield stress



Source: Grzina et al. 2002

Figure 62 Construction of the system resistance curve for a heterogeneous slurry

non-Newtonian system curve with low-flow, laminar, and high-flow, turbulent regions. The intersection of the pump performance curve with the system curves at more than one location indicates that there may be more than one operating point for the slurry pump, so that the pump may swing between high and low flows. This phenomenon has been observed on high yield-stress concrete slurries and thickener underflows (Bootle 2002).

The Hydraulic Institute (2015) provides guidelines for the effects of viscosity on pump performance, but they are limited to specific speeds less than 60 (dimensionless), and liquids exhibiting Newtonian behavior, with kinematic viscosity >1 and <40 cm^2/s.

Performance tests are recommended when a particularly viscous liquid is to be pumped, and this includes most mineral slurries. Bootle (2002) notes that the changes in performance can be approximated using a modified pump Reynolds number, but Rayner (1995) cautions that this approximation is best made by a "manufacturer who can utilize empirical data on a specific pump."

Pipe Friction in Slurry Systems

Several methods can be used for calculating friction losses in slurry pumping systems. In every case, it is difficult to predict friction losses correctly because of the infinite number of combinations of particle sizes that can occur in slurries. The method shown here produces values that are adequate for pipelines up to a few hundred meters long, with static heads less than about 30 m. More rigorous treatment and test work is advised for long pipelines and higher static heads.

It is common in these applications to use V-belt drive systems for the pumps. This is because pump duties often change—and usually increase—after installation. In a V-belt drive system, an increased speed and system throughput are easily obtained by changing the drive sheaves. Of course, power consumption will also increase, so it is wise to select the motors that have 10%–20% more than the initial calculated power requirement.

Calculation of friction losses when pumping liquids has been previously discussed. Friction losses in the pumping of homogeneous slurries of fine non-settling solids are calculated using the method described for Bingham plastic slurries. For determination of friction losses in pumping settling slurries, the limiting settling velocity, V_L, must be determined at the required mass flow, M, of solids. Pumping solids at high concentrations, C_w, requires small flow rates, Q, and small pipe diameters, D; the opposite holds with low concentrations. Usually only one combination will satisfy the required V_L. This value may be found using the Durand method (described here), the Wilson method (using Figure 54), or by test work.

To use the Durand method, proceed as follows:

1. For a fixed solids mass flow M, select a value of C_v and calculate the corresponding Q.
2. Select a standard pipe of inner diameter D.
3. Obtain from the Durand diagram in Figure 55 or 56 the corresponding V_L and multiply it by the pipe's cross section to obtain the limiting flow rate, Q_L.
4. If Q is 10%–15% larger than Q_L, the pipe diameter is correct and the solids will not settle during pumping.
5. If Q is equal to or smaller than Q_L, select the next smaller pipe diameter and return to step 3.
6. If no reasonable pipe diameter satisfies the set requirements, select another value of C_v and return to step 1.

This procedure provides a starting point for use when estimating a system curve, as shown in Figure 62. In this procedure, static head Z is assumed to be zero. If Z is positive (or negative), it must be drawn as a straight line, parallel to the bottom of the graph at a distance Z above (or below) the base. The left point of the water resistance curve then starts from this Z line instead of from zero.

In the method shown in Figure 62, proceed as follows:

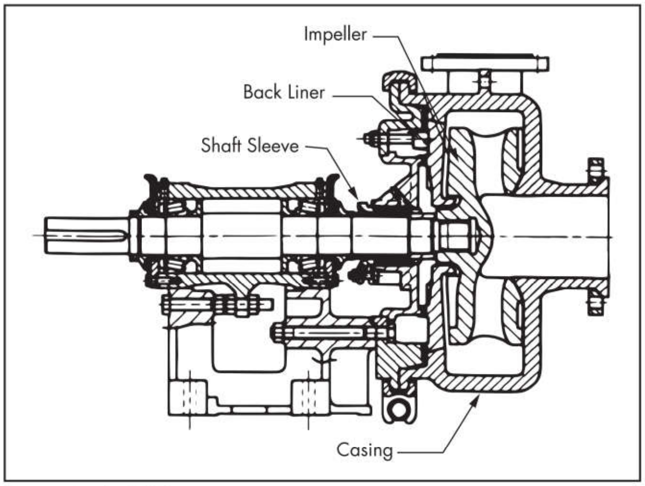
1. Calculate the water pipe friction loss, in meters of water at three flow rates, using the Moody diagram in Figure 2.
2. Plot the three points and draw the water resistance curve through them. Next mark the limiting flow rate Q_L estimated previously on the base line, draw a vertical line to the water curve at a, and from there draw a horizontal line to the left.
3. Draw two vertical lines up from the base: one from the value $0.7Q_L$ to meet the horizontal line at b, and the other from $1.3Q_L$ to the water curve at c.
4. Draw a parabola with vertex at b and tangential to the water curve at c. This is the slurry resistance curve.
5. Finally, mark Q on the base line and draw a vertical line to intersect the slurry resistance line at d, and from there a horizontal line to the left axis at e, which gives the friction head loss, H_f, in meters of slurry.

This method clearly provides only an estimate of the system resistance curve for pumping slurry, as the two curves are sketched by hand.

Centrifugal Slurry Pumps

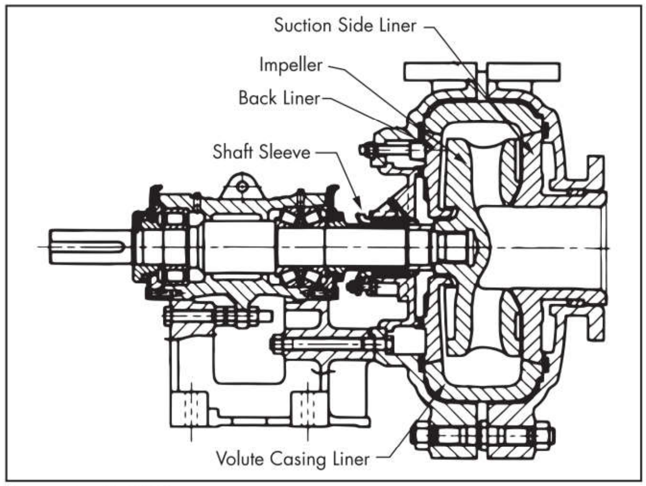
The design of a horizontal, centrifugal slurry pump is a balance of design considerations to best meet the requirements of a particular slurry duty. These requirements may include one or more of the following:

- The ability to pump high density, abrasive slurries with adequate wear life



Source: Bootle 2002

Figure 63 Wear components in an unlined centrifugal slurry pump



Source: Bootle 2002

Figure 64 Wear components in a fully lined centrifugal slurry pump

- The ability to pass large-diameter solids
- The ability to handle air-entrained or viscous fluids with reliability and minimal performance corrections

When compared with clear liquid pumps, the preceding requirements often result in the slurry pump being larger than its clear-liquid counterpart and sacrificing maximum efficiency in exchange for the ability to achieve the abovementioned goals.

As they come into contact with abrasive slurry, the inner components of a slurry pump will wear. Wear can be minimized by appropriate pump design, proper material selection, and proper pump application. Figure 63 illustrates a typical unlined, horizontal, centrifugal slurry pump and Figure 64 illustrates a fully lined version of the same pump. The corresponding wear components for each have been appropriately marked. To decrease wear, thick casting sections are provided on the impeller, the casing of the unlined pump, and the wear liners of the fully lined pump. The unlined casing casting thickness is often greater than that of a fully lined casing. This additional thickness is required because the unsupported, unlined casing must safely handle the internal pressure of the pump with an adequate factor of safety to account for wear. The lined pump also allows for the use of a wider variety of materials, such as elastomer liners, which often outperform metal in fine particle and corrosive applications. Therefore, while the unlined pump may offer the lower initial capital cost, the lined version allows for a greater number of material choices, which may have longer wear life and lower replacement spares cost. The lined design is also inherently safer from a pressure containment standpoint. Large clearances are provided within the impeller and casing to allow for the passage of large-diameter solids, while also reducing internal velocities and corresponding wear.

Slurry pump impellers tend to be larger than their clear-liquid counterparts. To minimize speed and maximize wear life of both the impeller and suction side liner, slurry pump impellers are rarely trimmed in diameter to meet the required duty point. For applications below 300 kW, belt drives are the most popular way to achieve the required speed for the duty point. Belt drives are inherently quiet and also allow for relatively easy speed changes if required. For higher-power applications, gearboxes are usually used to meet the desired speed and duty point. On applications where variable flow is required, VFDs are used to provide the necessary continual speed changes. Usually these VFDs operate with other speed reduction devices, such as belt drives and gearboxes, to allow for the use of higher-speed, lower-cost motors.

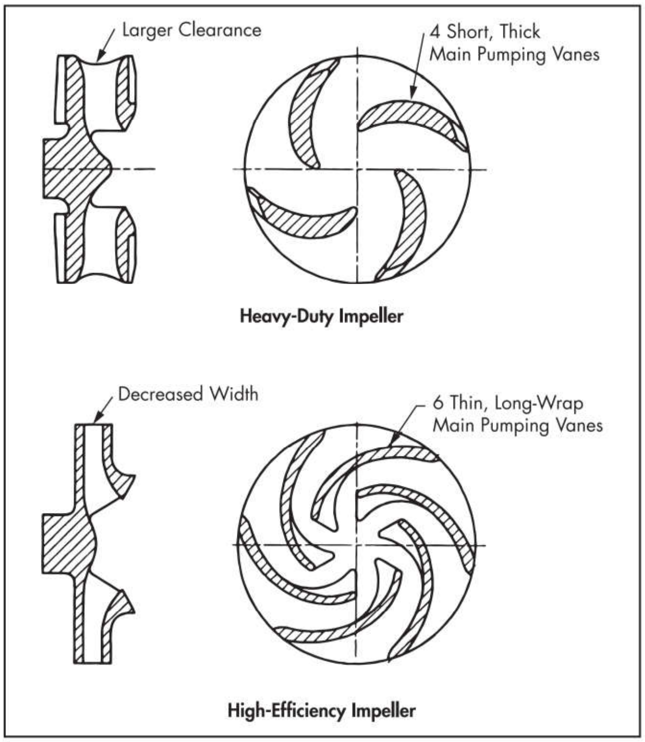
Bearing Assembly

The bearing assembly of a heavy-duty slurry pump should incorporate a larger-diameter shaft and larger bearings than those found on a clear-liquid pump. Roller bearings are often used to provide further additional load-carrying capacity to handle the higher forces that result from pumping a heavier liquid, the inherent imbalance due to wear, the shock loading from large particles, and the hydraulic imbalance from air entrainment. On belt-drive applications, most of the side load the drive-end bearing results from belt pull, which can be substantial. Most properly sized and properly tensioned V-drives exert approximately 9 kN of belt pull for every 100 kW of motor power.

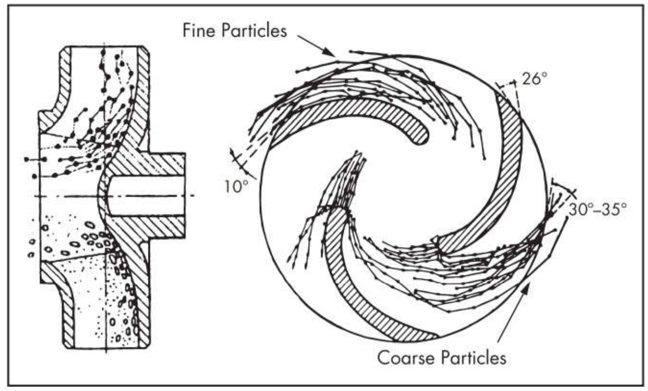
Figure 63 illustrates a bearing assembly with identical low-angle, angular contact bearings on both the pump end and drive end. The low-angle bearing is well suited to the radial loads from the impeller or belt pull from the drive, if so equipped.

Figure 64 illustrates a higher-capacity bearing assembly, which fits within the same packaging dimensions. This assembly features a duplex angular-contact roller bearing on the pump end and a cylindrical roller bearing on the drive end. The duplex angular-contact bearing has a higher bearing angle than the bearings shown in Figure 63, making it more suitable for higher axial loads, such as those seen with smooth-backed impellers, open-faced impellers, and series pumping applications. In this configuration, the cylindrical roller bearing on the drive end handles none of the axial load but has tremendous radial load capability, making it well suited to belt-drive applications.

When the shaft seal design allows it, shorter shafts with reduced impeller overhangs will result in reduction of bearing loads, shaft stress, and deflection through the seal area. When



Source: Bootle 2002
Figure 65 Heavy-duty and high-efficiency impellers



Source: Bootle 2002
Figure 66 Particle trajectories in a centrifugal slurry pump

packing is used as a shaft seal, a hardened or ceramic-coated shaft sleeve is recommended to prevent shaft wear.

Impeller Design

Slurry pumps typically have impellers that are larger than their clear-liquid counterparts. This is to lower the impeller speed required to achieve a given head and to provide more material for wear purposes. For high wear applications, closed impellers are preferred. In coarse particle applications, expelling vanes are recommended on the face of the front shroud. The purpose of these expelling vanes is to prevent large particles from becoming trapped between the impeller and suction side liner and to minimize recirculation. The benefit is a reduction in gouging and recirculation wear, with an acceptable decrease in efficiency.

Expelling vanes are also often used on the back shroud of the impeller in coarse particle applications to prevent the trapping of large particles between the impeller and back liner. In this location they also serve to reduce the forward axial load, improving bearing life by lowering the pressure acting on the back shroud and beneficially reducing the pressure at the hub and packing. This reduces the pressure differential at the shaft seal and reduces the tendency for slurry leakage from the pump. As with expelling vanes on the front shroud, back vanes usually absorb two to three percentage points of efficiency.

To decrease wear and allow for the passage of large-diameter solids, heavy-duty slurry pump impellers feature thicker main pumping vanes and fewer of them. Both of these factors further contribute to a reduction in efficiency when compared with a clear-liquid counterpart. While a clear-liquid impeller usually has five to nine vanes, most slurry pump impellers have two to five, with four and five vane designs being the most common. Two and three vane designs are usually reserved for very large particle passing sizes, as required in dredging applications.

Figure 65 illustrates the difference between a four-vane, heavy-duty slurry design with front and back expelling vanes and a six-vane, high-efficiency slurry design with smooth front and back shrouds. Notice the short "blocky" vanes on the heavy-duty design and the thin, long-length, long-wrap vanes on the high-efficiency design.

In slurries with large particles, the large solids follow a different path than the fluid, because of inertial effects. This is illustrated in Figure 66. This results in gouging wear at any location where the fluid is required to make an abrupt change in direction. For this reason, on large-particle applications, heavy-duty designs with blunt leading edges, wide spacing between shrouds, and thick back shrouds are recommended to reduce the impact of the large particles. Main pumping vanes should be short, with minimal vane overlap to allow the large particles to pass unimpeded. This heavy-duty geometry results in a head-versus-capacity curve that is flatter than the typical clear-liquid pump. The use of fewer main pumping vanes that are also thicker and shorter, along with expelling vanes on the front and back shroud, can reduce slurry pump efficiencies by up to 10% from those of a comparable clear-liquid impeller. These differences are minimized on larger pumps.

In slurries where d_{85} is less than 100 μm, almost all the particles will follow the path shown in Figure 66 for fine particles. In this case, high-efficiency designs that minimize the presence of vortices will improve efficiency and improve impeller life. In such applications, the absence of expelling vanes on the front and back shroud will improve side liner wear life.

Pump Materials

The primary materials used for wear components in centrifugal slurry pumps are hard metals and elastomers. Ceramics are also used, but less often. Hard metals and ceramics resist erosion because of their high hardness values. Elastomers resist erosion by their ability to absorb the energy of the impacting particles, which is due to their resilience and tear resistance. Elastomers generally have better erosion resistance than hard metals. In applications where particle size is smaller than 250 μm, impeller tip speed will be within the limits of the elastomer, and there is no risk of damage from occasional passage of large particles.

Metals

The three basic types of metals used to combat erosion in centrifugal pumps are the martensitic white irons, chromium-molybdenum white irons, and high-chrome irons. ASTM International Standard A532 (2014) provides specifications for these materials, which consist of hard carbides in a supporting ferrous matrix.

The bulk hardness of these materials depends on the carbide and matrix type, and on the volume of the carbides within the matrix. For applications with medium to large slurry particles, the bulk or combined material hardness is most important. For small-particle applications, a fine microstructure with smaller inter-carbide spacing is more important to minimize erosion of the softer matrix. For applications with particles larger than about 100 mm, the fracture toughness of the matrix is most important.

Ni-Hard 1 and Ni-Hard 4, which are martensitic white irons, have high hardness but their erosion resistance is not as good as the other ASTM A532 materials, and their low chromium content provides little corrosion resistance. In heavy-duty slurry applications, martensitic white irons have been largely superseded by chromium molybdenum white irons and high-chrome irons.

The hardness of chromium-molybdenum white irons and high-chrome irons can be increased by proper heat treatment, which can also increase corrosion resistance. The molybdenum white iron 15-3 alloy is suitable for use in slurry pumps where high erosion resistance is required and where impact-loading conditions and corrosion rates are minor to moderate, while the 27% high-chrome iron is suitable for use in slurry pumps where erosion resistance, corrosion resistance, and fracture toughness requirements are moderate to high.

The properties of high-chrome iron can be altered in many ways to suit particular pumping conditions. For example, high-chrome irons with an austenitic matrix and high impact resistance are available for extremely large-particle dredging applications where there are very large particles in the slurry.

Elastomers

Elastomers are typically used in applications with particle diameters <10 mm. They can be broadly broken into two categories: natural rubber and synthetic elastomers. Natural rubber has much greater resilience and tear resistance than the synthetic polymers.

Resilience is a measure of how high a ball of the material will bounce when dropped on a standard surface in relation to the initial drop height. Typically, this value ranges from 65% to 90%, depending on the rubber blend. Resistance to tear initiation for natural rubber is usually 30–110 N/mm, depending on the blend. For natural rubber, tear resistance tends to improve with increasing hardness, while resilience tends to decrease. For particles less than 100 µm, resilience is more important in combating wear, while for particles greater than 500 m, tear resistance is more important. Given that mineral processing slurries have a mixture of particle sizes, the best performing natural rubber will be one with the optimum combination of resilience for fine-particle wear resistance and tear resistance to prevent larger particle damage.

Synthetic elastomers are used in small-particle applications where natural rubber would be subject to chemical attack, causing swelling, hardening, or reversion of the natural rubber. Reversion of natural rubber is the decline in its physical properties, such as modulus and resilience, as a result of weathering or contamination. The most commonly used synthetic elastomers for wear materials and some of their typical applications are as follows:

- **Nitrile:** Generally used in fats, oils, and waxes. Moderate erosion resistance. Limited resistance to acid and alkaline environments.
- **Butyl:** Suitable for hydrochloric acid, phosphoric acid, and sodium hydroxide. Sulfuric acid causes degradation, and chlorinated hydrocarbons cause swelling.
- **Hypalon:** Primarily used in acid conditions with some resistance to vegetable and mineral oils. Not recommended for use in ketones or chlorinated solvents.
- **Neoprene:** Moderate resistance to oils, fats, grease, some hydrocarbons, and some mild oxidizing acids.

Synthetic elastomers also have higher temperature limits than natural rubber. While natural rubber is limited to 75°–85°C, depending on the blend, the above-mentioned synthetic elastomers have temperature limits ranging from 95°C for nitrile to 110°C for Hypalon.

Further comparisons can be made with respect to mechanical properties. The tear resistance for the above-mentioned synthetic elastomers ranges from 30 N/mm for nitrile to 50 N/mm for neoprene. The resilience of neoprene, at 58%, is the highest of the synthetic elastomers. The superior mechanical properties of natural rubber indicate it should be used in preference to synthetic elastomers, unless temperature or chemical resistance are overriding factors.

The high tear strength of polyurethane, 50–100 N/mm, makes it applicable for systems where damage by large particles is likely. Polyurethane is much less susceptible to damage from cutting than natural rubber and other synthetic elastomers. Most polyurethanes cannot be used above 70°C, but special compositions that will function up to 110°C with no degradation are available.

Tip-Speed Limits

Another important factor to consider in the selection of materials is the impeller tip-speed limit. For elastomers, the concern is vibration or fibrillation of the material due to the relative motion of the impeller with respect to the side liners. This vibration can lead to heat generation within the elastomer and a thermal breakdown of the material. For natural rubber, this often results in the elastomer reverting to its natural "gummy" state. Tip speed problems on elastomers are easy to diagnose, as the damage is always the most severe at areas with the highest relative speed. For elastomer impellers, this would be the periphery of the impeller, and for elastomer side liners, this would be the area adjacent to the periphery of the impeller.

The tip speed is the equivalent linear velocity of the impeller tip, and is a function of rotational speed and impeller diameter. The tip-speed limit is determined by the hardness of the elastomer and its ability to dissipate heat. Typically, natural rubber has a tip-speed limit of approximately 27.5 m/s. Highly wear-resistant soft natural rubber can have a tip-speed limit as low as 25 m/s, while proprietary blends with improved thermal conductivity can operate at 30 m/s. For a centrifugal slurry pump, this tip-speed limit is important, as the head generated (meters) is approximately equal to the quantity (0.5 times the square of tip speed in meters per second) divided by the acceleration due to gravity (9.8 m/s^2).

Approximate tip-speed limits and corresponding approximate BEP head limits for various materials are shown in

Table 3 Tip-speed and BEP head limits for various impeller materials

Material	Tip Speed, m/s	Head Limit, m
Highly wear-resistant soft natural rubber	25.0	32
Typical natural rubber	27.5	39
Anti-thermal breakdown rubber	30.0	46
Nitrile	27.0	37
Butyl	30.0	46
Hypalon	30.0	46
Neoprene	27.5	39
Polyurethane	30.0	46
Hard metal (impellers)	38.0	74

Adapted from Bootle 2002

Table 3. The tip-speed limit for hard metal impellers is based on the limited ductility of the material and not on thermal breakdown.

Pump Wear

Wear in a centrifugal pump will include various modes of abrasion and erosion. Abrasion, which is the forcing of hard particles against a wear surface, takes place within a slurry pump only on the shaft sleeve and between the tight tolerance wear-ring section of the impeller and the suction side liner. Erosion is progressive wear loss from the interaction with, or impingement of, the fluid and particles against the wear components. Three primary modes of erosion can take place, each of which usually occurs in a particular location in the pump. Deformation wear results from direct impact to the leading edge of the impeller vanes, the back shroud of the impeller, and the protruding portion of the cutwater within the volute. Random impingement is caused by impacts to the impeller shroud and trailing edge of the main pumping vanes. Low-angle impact wear results from the tangential or near-tangential movement of particles against the volute casing or vane surface.

Of these modes of erosion, deformation wear by direct impact is the most severe and low-angle impact is the least severe. The degree of wear depends on three factors: (1) the kinetic energy of the particle, which is a function of particle mass and velocity; (2) the particle shape—sharp particles have small contact area and produce high local stress, so wear is more severe than with rounded particles; and (3) the slurry concentration—higher percentages of solids result in more impacts for a given flow. Based on these factors, slurries are classified as follows:

- Heavy duty: $C_w > 35\%$, $d_{85} > 400$ µm, s.g. > 2.0, sharp particles
- Medium duty: $20\% < C_w < 50\%$, 150 µm < d_{85} < 400 µm, s.g. > 1.4, angular particles
- Light duty: $C_w < 20\%$, $d_{85} < 150$ µm, s.g., > 1.4, rounded particles

In addition to the general specific speed limits and the material tip-speed limits discussed previously, to maximize wear life, the following general impeller tip-speed limits apply based on the severity of the duty:

- Heavy duty: 25 m/s maximum, 32 m head at BEP
- Medium duty: 32 m/s maximum, 52 m head at BEP
- Light duty: 38 m/s maximum, 74 m head at BEP

Further recommendations for impeller type and flow range are as follows:

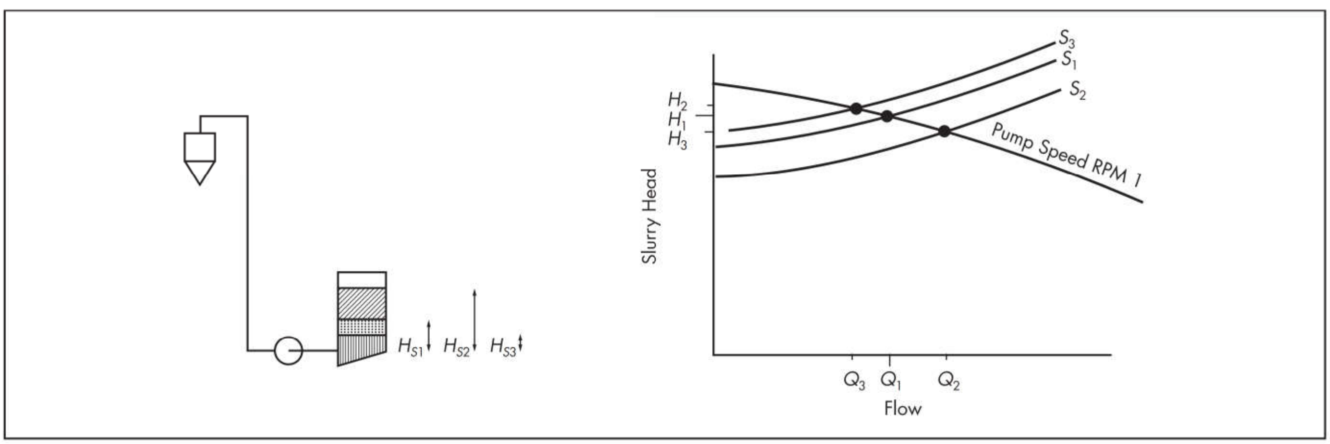
- Heavy duty: heavy-duty impeller at 0.60–0.80 BEP
- Medium duty: heavy-duty impeller at 0.70–0.90 BEP
- Light duty: high-efficiency impeller at 0.80–1.1 BEP

Operating Controls

Control by Sump-Level Variation

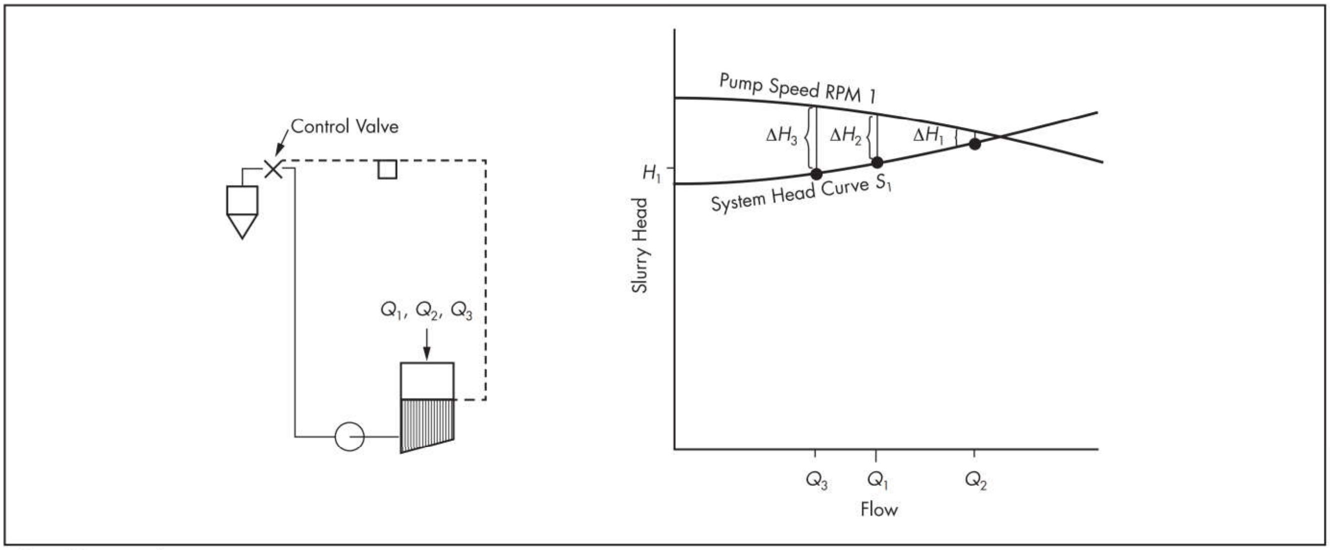
For lines shorter than about 300 m, flow must be varied to match process conditions while attempting to maintain minimum velocity. Most systems use the sump level to determine flow requirements. Because many centrifugal slurry pumps normally have a relatively flat output curve, with a large flow change for a relatively small head difference, many low-head systems use fixed-speed pumps with deep sumps and allow the fluctuating sump level to regulate flow.

Consider the system shown in Figure 67, delivering design flow Q_1 at a system TDH of H, calculated for a static suction head in the sump of H_{S1}. If inflow to the sump is allowed to vary so that the static suction head can vary up to H_{S2}, or down to H_{S3}, the system-head curve will vary, thus



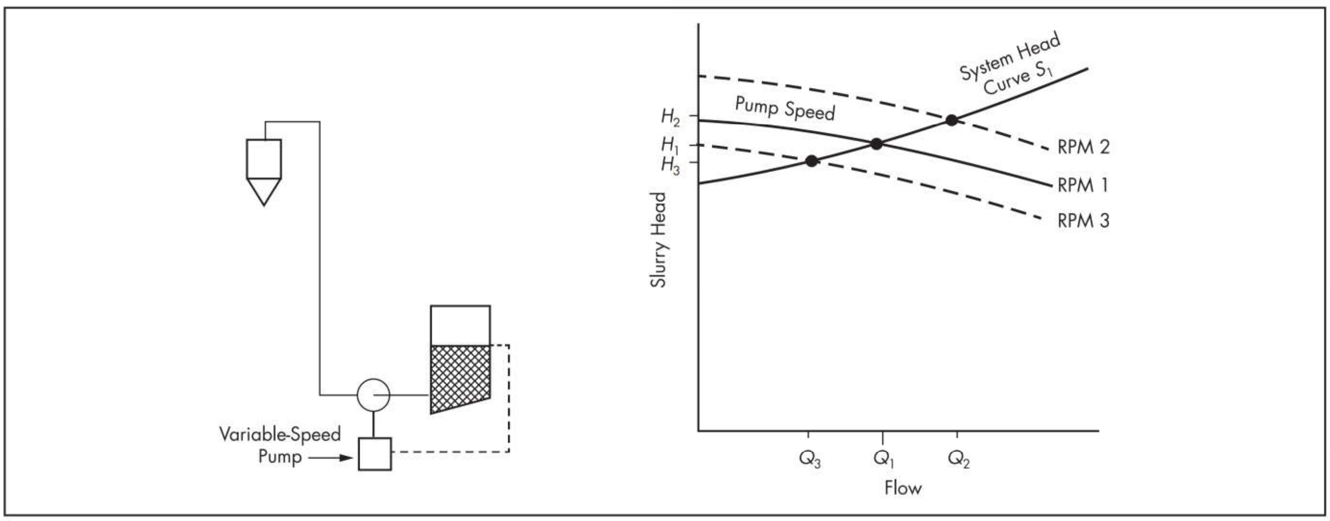
Adapted from Crosby 1985

Figure 67 Effect of sump-level variation on system-head curve



Adapted from Crosby 1985

Figure 68 Use of a control valve to handle varying flow



Adapted from Crosby 1985

Figure 69 Use of a variable-speed pump to handle varying flow

yielding corresponding flows Q_2 and Q_3 at new TDH points H_2 and H_3. If the pump is fixed at one speed, it can only operate between Q_2 and Q_3. In the figure, S_1 is the system-head curve at H_{S1}, S_2 at H_{S2}, and S_3 at H_{S3}.

This is an acceptable method of operating, but only if the slope of the system-head curve and pump curve will permit a wide enough variation in flow to accommodate the variation in incoming flow to the sump, flow is not allowed to drop so low as to cause sanding in the line, flow is not allowed to reach a maximum point where there is insufficient NPSH available for pump operation, and the sump static suction head stays high enough to provide sufficient NPSH and prevent vortexing and air entrapment.

Control Valves

Another method for handling varying sump inflow is to install a control valve in the pump discharge line, so that the pump discharge can be throttled to maintain a fairly constant sump level. Now consider the pumping system in Figure 68, again delivering design flow Q_1 at a system head H_1 calculated for a static suction head of H_{S1}. Suppose that sump inflow can vary between Q_2 and Q_3. If the pump curve is that of a fixed-speed unit, a variable resistance, indicated by ΔH_1, ΔH_2, and ΔH_3, is required to operate on the curve. This resistance is provided by a throttling valve in the discharge pipe that operates in response to a sump-level controller. Proper selection and sizing of the control valve is essential for good operation and maintenance.

Variable-Speed-Drive Control

Changing pump speed is another alternative for handling varying flow rates. Figure 69 shows the pumping system, again operating with design flow Q, at a system TDH of H_1 calculated at a static suction head H_S. When incoming flow changes from Q to Q_3, the pump speed can be varied to match the system-head curve requirement. As in the case of the control valve, a sump-level control device or flowmeter is used to generate a signal that slows pump speed or increases pump speed when the sump level rises or falls, respectively. Pump speed can be controlled by a variable-speed coupling, a variable-speed motor, or a variable-frequency drive for the motor.

Water Addition to Sump

Variations in sump inflow can also be smoothed out by the automatic addition of water to the sump. In this case, the pump has a fixed speed designed to handle the expected maximum steady sump inflow. Water is added to maintain the sump level and pumping volume.

Control for Safe Operation

Most centrifugal slurry pump systems require some safety controls to protect pumps, drivers, or pipelines on start-up, during operation, and on shutdown.

Start-Up

When a pump is started, particularly with an empty discharge line, it sees virtually no friction resistance and will instantaneously try to pump as far out on the end of its fixed-speed curve as NPSHA will permit. Operation approaches the curve design point as quickly as static head and friction resistances can be built. Motors must be supplied that will permit an overload for this start-up period. These start-up loads can be reduced by starting with the discharge line full or partially full of slurry carrier liquid, then building pump speed slowly to the design with the use of a variable-speed coupling or motor, and by providing resistance by removable orifices or a throttling discharge valve.

For short, in-plant lines, start-up problems are usually not serious. However, long-distance tailings lines that use centrifugal pumps in series are another matter. These pumps should be started sequentially from the pump in the series train closest to the sump to the last pump that discharges into the pipeline. The time between starts should be sufficient to allow each pump to operate at a point close to its intended design point. Starting intervals that are too rapid invite motor overloads and pump damage due to cavitation. The timing for the start of each successive stage can be controlled by pressure sensors located at the discharge of the preceding stage or alternatively to a single flowmeter connected to the pipeline beyond the discharge of the last-stage pump. When sufficient pressure is reached (depending on start-up slurry density) or the flowmeter records a flow sufficiently on the pump curve, the next stage is started.

During Operation

Probably the most hazardous event that can occur in centrifugal pumping is the blockage of the pump suction or discharge. This leads to heating of the liquid trapped in the volute with subsequent vaporization and explosive pressure buildup. This hazard is more acute in the case of slurry pumping where line blockages can readily occur and the inadvertent closing of either suction or discharge isolation valves can produce dire consequences.

At least two methods of protection are possible. One is to use a current-limiting switch to shut off the pump when a low current corresponding to pump shutoff flow is detected. The other is to provide a rupture disk to relieve pressure in the pump casing. In the case of gland water pumps, this could be placed in the seal water line as close to the pump as possible.

The extent and degree of automatic operation varies considerably, depending on the location and critical nature of pump station operation, as well as the availability of operating personnel. For simple in-plant pumps, seal-water flow alarms and controls may be interlocked into pump start-up and shutdown controls. High sump-level alarms may be used to detect insufficient discharge flow. Low-level sump alarms and detectors may be used to warn of insufficient pump NPSH or submergence, or even to shut down the pump. Automatic pump shutdown may be provided in the case of line rupture, where a considerable decrease in pump resistance results in over-pumping and subsequent motor overloading—similar to what may happen in pump start-up. Large, unattended pumps may incorporate alarms for excess vibration and high bearing temperature.

Shutdown

Emergency or planned shutdowns should be considered carefully to avoid major equipment damage. Most slurry systems are shut down and started up with water. It should be remembered that a centrifugal pump produces head or pressure, while a pipeline system relies on pressure for flow to occur. The pressure drop required to move a pipeline full of slurry is high, and when water only is introduced into the sump, pump discharge pressure will decrease, with an attendant decrease in line flow rate. This may be compensated for by an increase in pump speed if practical. The greater the slurry specific gravity and friction loss, the more closely the changeover from slurry to water must be evaluated, as transient flow instabilities are possible.

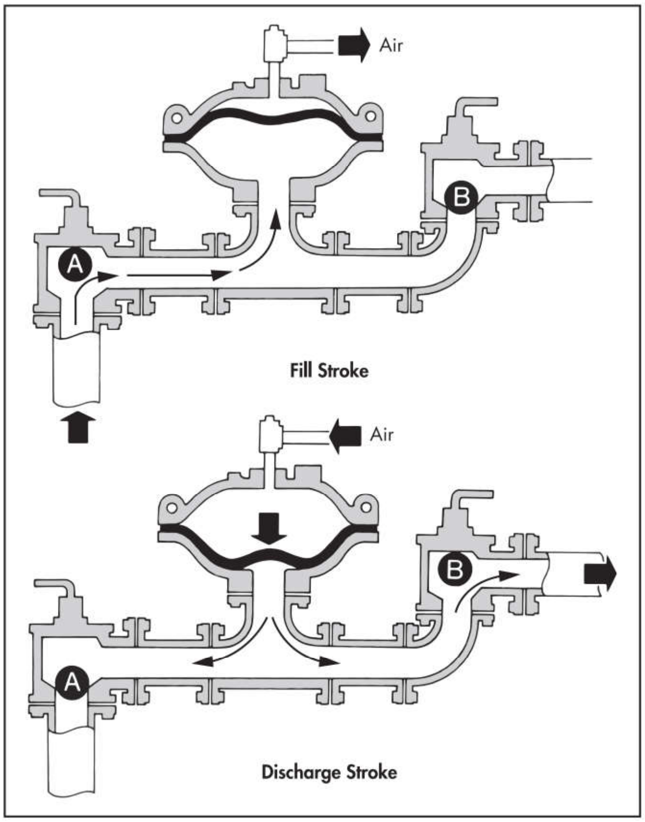
For long slurry pipelines, safe draining of the lines must be considered. High spots in the line should be opened to prevent line collapse under vacuum. During emergency shutdowns of such lines (e.g., due to power failures), the accelerating mass of slurry can cause high-pressure shock waves that can burst pipelines and equipment. Sometimes protection can be afforded by installing slow-acting shutoff valves, pulsation dampers, or similar devices. Restarting during reverse flow through pumps can cause equipment damage. During planned shutdown of a multistage pump system, pumps are shut down in reverse sequence to start-up.

POSITIVE DISPLACEMENT PUMPS

The most commonly used types of PD pumps for slurry handling are diaphragm pumps, progressing cavity pumps, and reciprocating piston or plunger pumps, all of which have been described previously. Figure 70 shows an air-operated diaphragm pump used for pumping high-density slurries containing up to 75% solids at a maximum of 90°C. The black spheres marked “A” and “B” in the figure are ball check valves that control flow through the pump during operation.

Progressing cavity pumps are widely used in handling abrasive slurries. Such a pump should be selected to maximize the life of its wear parts. In many cases, a small pump at high speed could handle the pressure and volumes required, but a larger pump operating at a lower speed will last longer and thus be more economical. Progressing cavity pumps have been used for pumping thickener underflow; slurry from dewatering applications; slurries of copper sponge, uranium yellow cake, and molybdenum; pilot-plant slurries; sewage; cement for grouting, at high pressure; and caulking compounds; and pressure spraying of oil on coal.

Reciprocating pumps are used with slurries in which the solid phase has a relatively low susceptibility to degradation during flow, such as with coal slurries. Ordinarily, the piston runs in a renewable metal cylinder or liner made of abrasion-resistant material. Piston rods and plungers are coated to resist wear. Valves and seats on reciprocating slurry pumps are designed to reduce erosion of the valve seat and components. Valve seats are usually replaceable and constructed of an elastomer or plastic material.



Courtesy of Dorr-Oliver, as cited in Scott and Hays 1985

Figure 70 Air-operated diaphragm pump

Controls for Operation and Safety

Large PD pumps inherently introduce system pressure pulsations as the result of suction and discharge displacements of specific volumes of liquid. In a duplex pump, two pulses are created; and in a double-acting duplex pump, four pulses are created. When multiple pumps are operated in a station, pump speeds must be staggered to keep the pulses out of phase and prevent a sympathetic cycle. Pulsations and vibrations must be accounted for in piping design and the location of instrument sensors, transmitters, and controllers.

PD pumps are constant-volume pumps. A change in volumetric flow rate is affected strictly by an adjustment of pump speed. Speed is normally adjusted using variable-speed motors, or by incorporating fluid or magnetic couplings units.

Sensing devices, gauges, and transmitters should not be located on piping immediately around PD pumps, as they will be subject to vibration failures. Such devices should employ a sealed capillary pressure-sensing system in which a diaphragm separates the slurry from the instrument and an air-free secondary hydraulic fluid transmits slurry variations from the sensor side of the diaphragm.

EXAMPLE CALCULATIONS

This section includes examples of typical calculations made in designing pump systems. Software packages for the design of pump systems include such calculations, of course, but it is still worthwhile for mineral processing engineers to be familiar with underlying concepts and calculations. The notation and units of measure in these examples are not entirely consistent with the preceding text but are used nonetheless to illustrate some of the variety of practice in describing pump systems.

Example 1. Calculating Pump Total Head by Two Methods

A centrifugal pump delivers 227 m^3/h of liquid having specific gravity 0.8 from an open suction tank to a discharge tank,

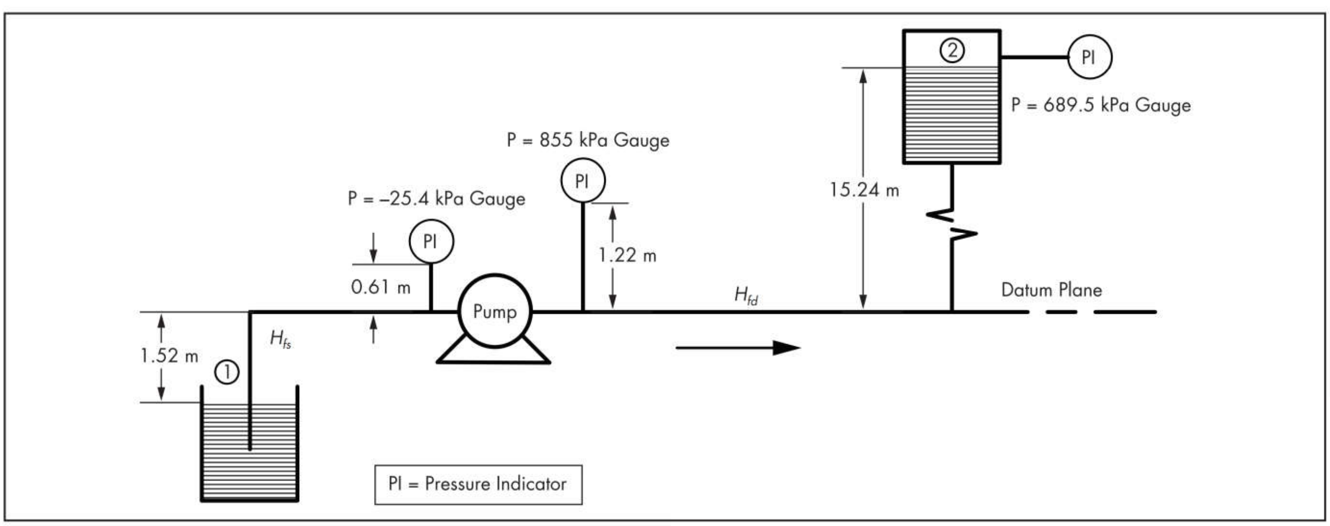


Figure 71 System for calculation of total pump head

through the piping as shown in Figure 71. The suction pipe and discharge pipe have inside diameters of 203 and 152 mm, respectively. The pressure friction heads for the pipe, valves, and fittings are 0.91 m for the suction side, H_{fs}, and 7.62 m for the discharge side H_{fd}.

Figure 71 shows four points at which pressures may be used for calculation of total pump head. The first two points are designated with circled numbers 1 and 2, and are at the suction and discharge tank, respectively. The second two points are at the two pressure gauges installed on either side of the pump.

The pump total head is first calculated using the gauge readings and datum plane shown. The total head, TH, in a pump system is the difference of the discharge head H_d and the suction head H_s, minus the total head loss due to friction ΣH_f, thus

$$\mathrm{TH} = (H_d - H_s)$$

which may be expanded as

$$\mathrm{TH} = [(V_d^2/2\mathrm{g} + (p_d/\rho) + Z_d] - [(V_s^2/2\mathrm{g} + (p_s/\rho + Z_s]$$

where

$V_{d,s}/2\mathrm{g}$ = velocity head, m
$V_{d,s}$ = flow velocity of the discharge or the suction, m/s
g = acceleration due to gravity, m/s^2
$p_{d,s}/\rho$ = liquid pressure head in the discharge or the suction, m (positive or negative)
$p_{d,s}$ = gauge pressure of the discharge or the suction, N/m^2 (positive or negative)
ρ = specific gravity of liquid
$Z_{d,s}$ = elevation above or below the datum plane of the discharge or the suction, m (positive or negative)

In using this expression to calculate total head, ensure that all pressure are either *absolute* or *gauge* values.

Pipe velocity in m/s is conveniently calculated as

$$V = (\mathrm{m}^3/\mathrm{h})(3.54)/(\text{pipe inside diameter, in cm})^2$$

where the required unit conversions are include in the factor 3.54, thus,

$$V_s = 227 \times 3.54/20.32 = 1.95 \text{ m/s, and}$$

$$V_d = 227 \times 3.54/15.22 = 3.48 \text{ m/s}$$

Similarly, liquid pressure head may be calculated as

$$p/\rho = 0.102 \times \text{pressure in kPa}/\rho \text{ m, thus}$$

$$p_d/\rho = (0.102)(855)/0.08 = 109.2 \text{ m, and}$$

$$p_s/\rho = (0.102)(-25.4)/0.08 = -3.2 \text{ m}$$

Finally, from Figure 71,

$$Z_d = 1.22 \text{ m}$$

$$Z_s = 0.61 \text{ m}$$

The total head is then calculated as

$$\begin{aligned}\mathrm{TH} &= [(V_d^2/2\mathrm{g} + (p_d/\rho) + Z_d] - [(V_s^2/2\mathrm{g} + (p_s/\rho) + Z_s] \\ &= (0.62 + 109.01 + 1.22) - (18.65 + 3.24 + 0.61) \\ &= 113.29 \text{ m}\end{aligned}$$

The total head may also be calculated using the pressures at points 1 and 2 and the same datum plane using the following equation:

$$\mathrm{TH} = [(V_d^2/2\mathrm{g} + (p_d/\rho) + Z_d] - [(V_s^2/2\mathrm{g} + (p_s/\rho + Z_s] + \Sigma H_f$$

where ΣH_f is the sum of pressure losses due to friction, m, and all other terms are defined as in the previous calculation.

Note that point 1 is at the surface of the *open* suction tank, so the gauge pressure and the liquid pressure at that point will be zero. Further note that because neither point 1 nor point 2 is in the system that is pressurized by the pump, the suction and discharge flow pressures are not included.

$$\begin{aligned}\mathrm{TH} &= [0 + (p_d/\rho) + Z_d] - [0 + 0 - Z_s] + [0.91 \times 7.62] \\ &= [(0.102 \times 689.5)/0.8 + 15.24] - (-1.52) + 8.53 \\ &= 113.20 \text{ m}\end{aligned}$$

The slight discrepancy between the two results (113.29 m and 113.20 m) results from rounding errors.

Example 2. Calculating Head Loss due to Friction

Use the Darcy–Weisbach formula to calculate the frictional head loss for 30 m of 54-cm, smooth concrete pipe. The pipe has an inside diameter of 48 cm. The system operates at 60°C, with water flowing at 2,500 m^3/h.

Recall that the Darcy–Weisbach formula calculates friction loss H_f as

$$H_f = fLV^{2/}2\mathrm{g}D$$

where the Darcy friction factor f is determined experimentally or from a chart, the pipe length L and the inside pipe diameter D are in meters, the liquid velocity V is in m/s, and the acceleration due to gravity g is in m/s^2.

The Darcy friction factor may be determined graphically from the Moody diagram (Figure 2) and reproduced in Figure 72. Figure 72 shows that smooth concrete has a roughness ε of 0.025 mm, so that the relative pipe roughness, ε/d (or ε/D) is equal to 0.025 mm/491 mm = 5.09E-05. (Note that ε and D must be in the same units, so that the relative pipe roughness is dimensionless.)

The liquid velocity is found from the volume flow rate and the pipe inside area, thus,

$$V = (2{,}500 \text{ m}^3/\text{h})/[\pi\,(0.48^2)/4] = 13{,}816 \text{ m/h} = 3.84 \text{ m/s}$$

where the denominator in the equation is the inside area of the pipe, in square meters.

It is also necessary to calculate the Reynolds number, N_{Re}, which is defined as

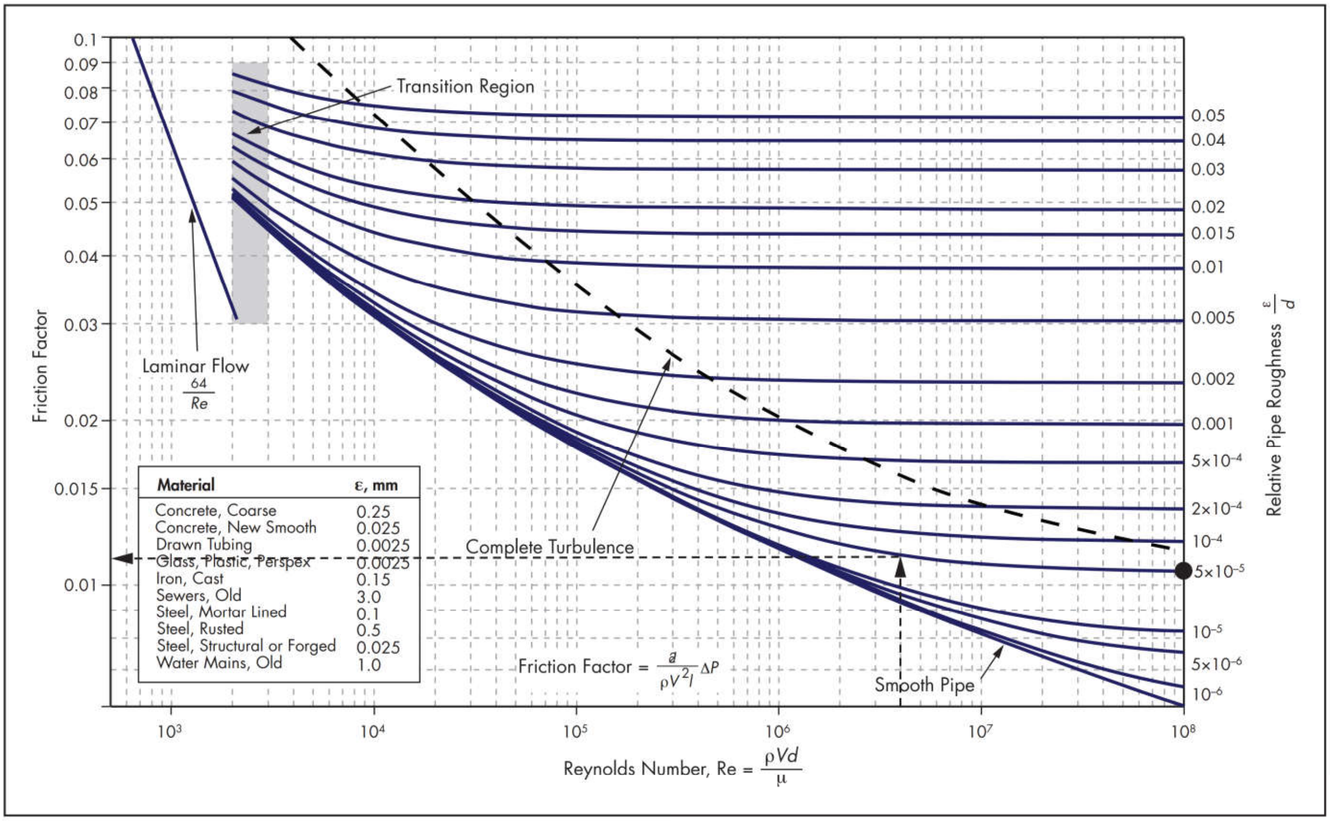
$$N_{\mathrm{Re}} = \gamma D v/\mu$$

where γ is the fluid density, D is the inside pipe diameter, v is the fluid velocity, and μ is the dynamic viscosity.

Fluid density and dynamic viscosity of water vary with temperature. In this case, at 60°C, γ = 983.4 kg/m^3 and μ = 4.66E-04 Pa·s. Thus,

$$N_{\mathrm{Re}} = \rho D v/\mu = [(998.4)(0.491)(3.83)/4.66\text{E-}04] = 4.03\text{E+}06 \text{ (dimensionless)}$$

The value of the friction factor f is found in Figure 72 by first locating the relative pipe roughness on the right-hand axis of the figure, as shown by the black circle. This defines



Adapted from Beck and Collins 2008

Figure 72 Moody diagram for Example 2

the curve that will be used. In this example, the relative pipe roughness is almost exactly on the 5E-5 curve, so that curve will be used. If the relative pipe roughness is between two curves, a curve may be lightly sketched on the diagram, maintaining proportional distances from the two bounding curves.

A dashed line is then extended vertically from the point on the horizontal axis corresponding to the calculated Reynolds number to intersect the indicated pipe roughness curve. From that intersection, a dashed line is extended to the left, intersecting the left-hand axis to indicate the value of f for these conditions—in this case, 0.0107.

The friction loss for the 30-m length of pipe may then be calculated from the Darcy–Weisbach equation:

$$H_f = fLV^2/2gD = (0.0107)(30)(3.84^2)/(2)(9.807)(0.491)$$
$$= 0.482 \text{ m}$$

Example 3. Calculating NPSHA

A tailings pump operates in a mill at sea level. The pump draws from an open sump and moves a fine slurry that has a specific gravity of 1.3. The pump centerline is 2 m above the controlled water level in the sump. Assume a temperature of 15°C and an absolute pressure at the liquid surface of 101.325 kPa (1 atm). The friction head, H_f, in the pump inlet line is 0.8 m.

To calculate the available net positive suction head, NPSHA, at the pump inlet, recall that

$$\text{NPSHA} = (H_{atm} \pm H_s - H_f) - H_{vp}$$

where

H_{atm} = atmospheric pressure
H_s = static height of liquid above or below the pump inlet or the center datum line
H_f = friction head in the pump inlet line
H_{vp} = vapor pressure of the liquid at the operating temperature and atmospheric pressure

NPSHA is expressed in height of an equivalent column of the liquid or slurry being pumped. Two components of the NPSHA, H_s and H_f, are given, but H_{atm} and H_{vp} must be calculated. The atmospheric pressure is converted to meters of water and divided by the specific gravity of the slurry to give meters of slurry, thus,

$$H_{atm} = 1 \text{ atm} \approx 10.34 \text{ m } H_2O \text{ (given) at } 15°C$$
$$= 10.342/1.3 = 7.96 \text{ m slurry at } 15°C$$

The vapor pressure of water is determined from standard tables at the given temperature, and converted:

$$H_{vp} = 0.0168 \text{ atm} = 0.174 \text{ m } H_2O \text{ at } 15°C$$
$$= 0.174/1.3 = 0.134 \text{ m slurry at } 15°C$$

Available net positive suction head is then calculated as

$$\text{NPSHA} = 7.96 - 2.0 - 0.134 - 0.8 = 5.03 \text{ m slurry}$$

The values of H_{atm} and H_{vp} will of course vary with climatic conditions, so NPSHA should always exceed NPSHR by an adequate margin.

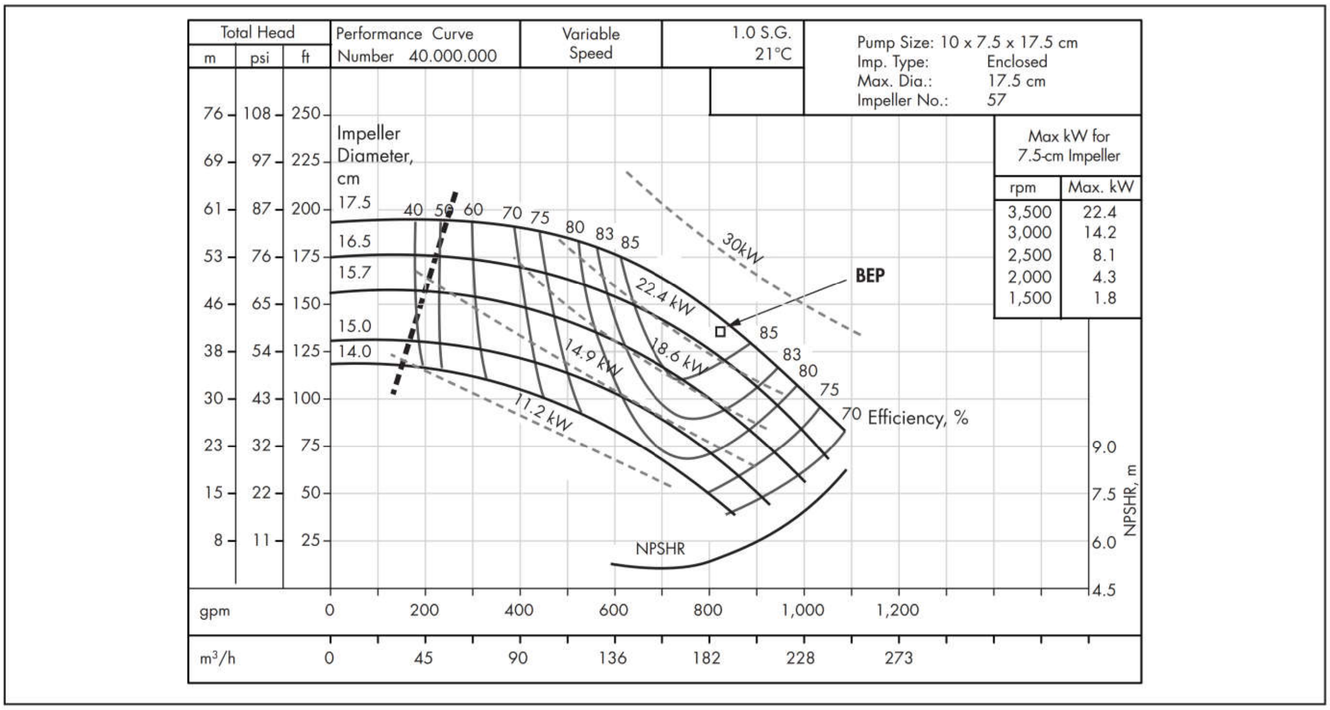


Figure 73 Typical pump curve chart

Example 4. Reading Pump Charts

A pump chart is a graphical representation of the performance characteristics of a pump. A basic curve shows the pump's output pressure or head versus flow rate on an x-y graph. The pump chart usually includes several nearly parallel curves that show the pump's performance for different impeller sizes, different rotational speeds, or some other variable. The chart may also show other operating parameters associated with pump performance, including rotational speed, impeller size, power, efficiency, and NPSHR.

In composite pump charts, all the parameters are shown on the same axes, as in Figure 73. Head versus flow rate is shown for several conditions—usually rotational speed or impeller diameter—by the group of similar heavy, solid curves that trend downward to the right. In Figure 73, these pump performance curves indicate the head versus flow for different impeller diameters 14–17.5 cm. Efficiency is shown by the lighter, solid lines that intersect the performance curves and are labeled at both their ends with numbers from 40 to 85, indicating the percent efficiency along each curve. In many pump curves, the BEP is shown, usually in the trough of the highest efficiency curve. In Figure 73, the BEP is indicated by a small square.

Pump power draw is shown by the dashed, nearly straight curves that are labelled from 11.2 to 30 kW. NPSHR is shown by the single, solid line at the bottom right of the performance curves, and a separate scale is provided for this curve. In Figure 73, the axes for flow and pressure head are labelled in metric and U.S. units. This is common in many pump curves.

A pump chart may also show one or more of the secondary parameters on separate sets of axes below the main graph. In these secondary graphs, the abscissas of the secondary graphs are the same as and aligned with that of the primary graph, indicating flow rate, so that the values of all parameters may be read as functions of a given flow rate.

A pump chart is used by plotting the system-head curve for a pumping system on the pump curve. In doing this, the fixed system head is usually neglected, since it affects both the pump performance curve and the system-head curve equally. The variable system total head loss through all piping, valves, fittings, and equipment in the system is calculated or measured for several flow rates, and plotted directly on the pump curve, as shown in Figure 73 by the heavy, dashed line on the left that extends upward and to the right from the origin. The points at which the pump will operate in the system are shown by the intersections of the system-head curve with the pump performance curves.

Considering the performance curve for the 17.5-cm impeller, it is seen that at zero flow, also known as shutoff or dead-head, the pump will develop about 58 m of head. This is often referred to as the shutoff head or the dead head and is the head the pump would develop operating against a closed valve.

The pressure measured between the pump and the closed valve might exceed this shutoff head value, because a pump adds head to the liquid being pumped. If this pump were operating at shutoff with suction pressure of 6 m, the total head seen at the pump discharge would be 58 + 6 = 64 m.

The location of the BEP relative to the operating condition for the pump is informative. Pumps run best at or near the BEP, and the preferred operating region (POR) is usually considered to be that in which flow is from 70% to 120% of the flow at BEP for most centrifugal pumps. In Figure 73, the BEP is at approximately 178 m^3/h flow and 63 m head, so the preferred operating range is from about 125 to 215 m^3/h. The intersection of the system-head curve with the performance curve for the 17.5-cm impeller shows that for the system in

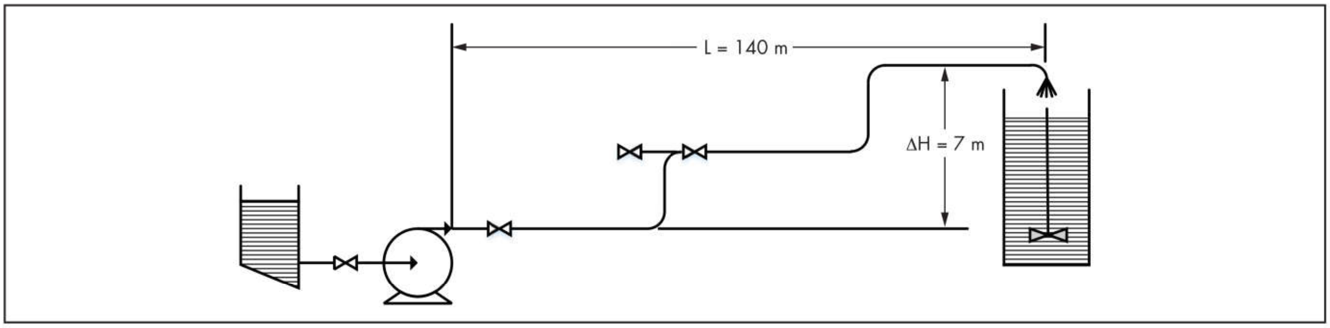


Figure 74 Tailings pumping system

question, the pump will deliver 182 m^3/h, and will thus be operating in the POR.

Many manufacturers also show a continuous stable flow line on their pump charts. The nearly vertical, dotted line that intersects the performance curves on the left side of the chart indicates the conditions at which the manufacturer has determined the pump should not be allowed to operate for any extended time.

Example 5. Calculating Critical Velocity and Pump Head for a Slurry Pumping System

For the tailings system shown in Figure 74 and described in Table 4, determine the critical flow velocity and the pump discharge pressure. The pipe size is specified on a trial basis and may change after the initial evaluation of the system.

Begin by determining the settling velocity for the particles in the slurry. This is the critical flow velocity V_L that must be maintained to keep the slurry in suspension while it is flowing. The settling velocity may be estimated using Durand's method. Assume that the slurry is closely graded, and that the d_{50} size is 270 mesh (53 mm). Referring to Figure 55 earlier in the chapter, extrapolate above the 15% C_v line to estimate the limited settling velocity factor F_L as 1.3.

The critical velocity is then calculated as

$$\begin{aligned} V_L &= F_L\sqrt{2gD(\rho-1)} \\ &= 1.3\sqrt{2\times 9.807\times 0.22(2.7-1)} \\ &= 1.22 \text{ m/s} \end{aligned}$$

This estimate indicates that the specified flow velocity, 1.58 m/s, may be too close to the critical velocity. The margin could be increased by using a smaller pipe or a higher-volume flow rate. However, because the flow correction factor was estimated based on an assumed d_{50} particle size in the slurry, it will also be worthwhile to repeat the calculation using a measured d_{50} value. If d_{50} is 40 mm, F_L will be approximately 0.6, and V_L will be 1.04 m/s, in which case the pipe size will be adequate for the specified flow rate.

The calculation of the required pump discharge pressure will be based on an assumption that the specified flow velocity, 1.58 m/s, is adequate. The total discharge pressure includes the pressure to overcome friction and the static head resulting from the increase in elevation from the pump outlet to the pipeline discharge.

To calculate the pressure required to overcome friction losses, H_f, use the the Darcy–Weisbach formula and follow the method used in Example 2. In this case, the Reynolds number is

Table 4 Conditions for a slurry pumping system

Condition	Units
Slurry type	−200 mesh copper tailings
Solids content	60% by weight
Solids specific gravity	2.7
Slurry volume concentration	0.356
Slurry specific gravity	1.61
Slurry density	1,600 kg/m^3
Slurry coefficient of rigidity	0.06 Pa·s
Flow	3.60 m^3/min
Pipe inside diameter, trial basis	0.22 m
Pipe material	Steel, commercial or structural grade
Flow velocity	1.58 m/s
Length (including fittings)	140 m
Inlet elevation	225 m
Discharge elevation	232 m (to atmosphere)

Adapted from Grzina et al. 2002

$$N_{Re} = \rho Dv/\mu = [(1{,}600)(0.22)(1.58)/0.06] = 9.26\text{E+03}$$

Figure 72 shows that commercial steel pipe has a roughness ε of 0.025 mm, so that the relative pipe roughness, ε/d (or ε/D) is equal to 0.025 mm/220 mm = 1.13E−04. The friction factor f is again determined from Figure 72 as 0.036.

$$\begin{aligned} H_f &= fLV^2/2gD = (0.036)(1)(1.58^2)/(2)(9.807)(0.22) \\ &= 0.02 \text{ m } H_2O\text{/m pipe} \end{aligned}$$

The total required discharge pressure is given by

$$H_f = (0.20 \text{ m } H_2O\text{/m})\ (140 \text{ m}) = 2.75 \text{ m } H_2O$$

plus

$$P_{static} = 7.00 \text{ m } H_2O$$

Thus, the total required pump head is

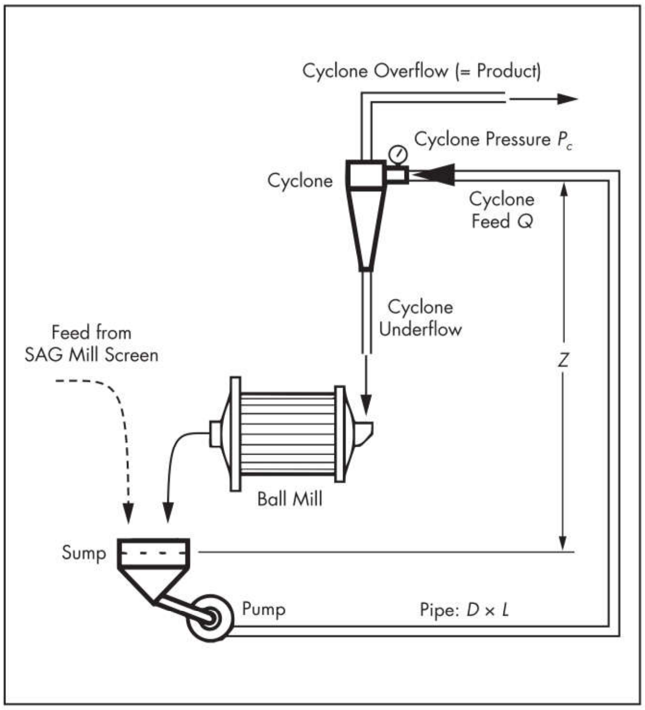
$$P_{total} = 9.75 \text{ m } H_2O$$

However, the pump is moving slurry with a specific gravity of 1.6, so the pump must provide

$$1.6 \times 9.75 = 15.61 \text{ m slurry at } \rho = 1.6$$

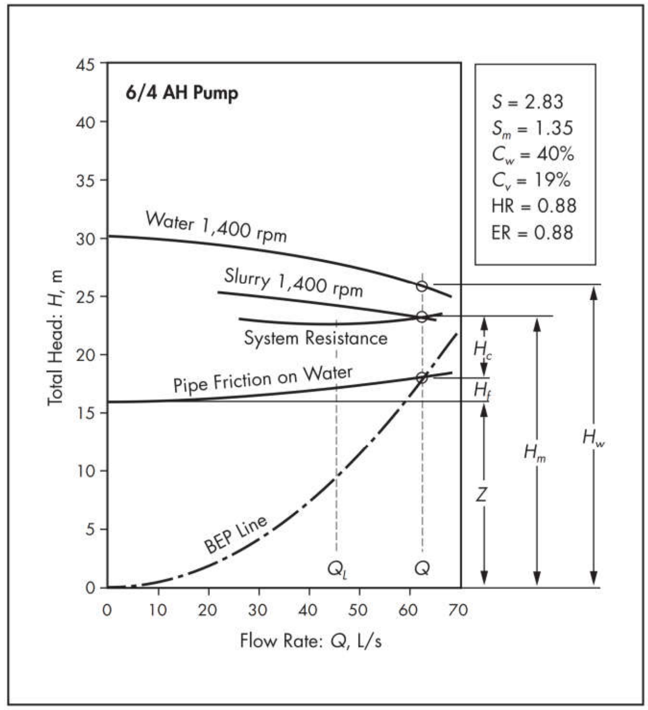
Example 6. Ball Mill Discharge Pump and Cyclone Circuit

In the system shown in Figure 75, the discharge from a ball mill reports to a sump, from which it is pumped to a cyclone



Source: Grzina et al. 2002

Figure 75 Ball mill, pump, and cyclone circuit



Adapted from Grzina et al. 2002

Figure 76 Mill discharge, pump, and cyclone performance curves

above the mill, for separation at a given particle size. The cyclone overflow, containing the fine particles, moves to the flotation circuit, while the coarser cyclone underflow returns to the mill for additional grinding. The height of the cyclone inlet above the mean water level in the pump sump Z is 16 m, the internal pipe diameter D is 0.150 m, and the total equivalent length of pipe L is 30 m.

The system operates in a concentrator treating copper ore that has a specific gravity of 2.85. The flow of slurry fed to the cyclone Q is 61.7 L/s, the solids concentration in the slurry C_w is 40% (C_v = 19%), and the specific gravity of the slurry is 1.35. The particle distribution of the cyclone feed is d_{20} = 60 µm, d_{50} = 250 µm, and d_{80} = 750 µm. Thus, d_{80}/d_{20} = 12.5.

The cyclone requires an inlet pressure, P_c, of 65 kPa at 61.7 L/s. The pump selected for the duty is a 6/4 AH Warman slurry pump with a hard-metal impeller of outer diameter D_i of 0.365 m. Its performance curves for water and slurry together with the system resistance curves are shown in Figure 76.

A flow of 61.7 L/s in a steel pipe of 0.150 m will have a pipeline velocity of 3.8 m/s. Because d_{80} is more than five times greater than d_{20}. Durand's limiting factor for pipe velocity, F_L, can be estimated from Figure 74 as 1.1 for conditions given. The limiting settling velocity is

$$V_L = F_L \sqrt{2gD(S_S - 1)} = 2.3 \text{ m/s}$$

This is less than the flow velocity, so there will be no settling in the pipe.

To determine the system resistance curve for the pipeline. Using Figure 2, with a relative pipe wall roughness e/D of 2.8×10^{-4}, the Darcy friction factor f is 0.016 and the friction head loss H_f=1.98 m of slurry. The cyclone pressure of 65 kPa and the corresponding cyclone head, H_c, are

$$H_c = P/\gamma g = 65{,}000/(1{,}350 \times 9.81) = 4.91 \text{ m slurry}$$

The units for the density γ are kg/m^3.

The static head Z is 16 m, so the total head required from the pump is

$$H_m = Z + H_f + H_c = 16 + 1.98 + 4.91 = 22.9 \text{ m slurry}$$

The head and efficiency ratios for the pump are estimated from Figure 58. The particle-size to impeller-diameter ratio, d_{50}/D_i, is 0.0007, so the estimated head ratio and efficiency ratio are each 0.88.

Figure 76 shows a portion of the performance curves for the Warman 6/4 AH pump when pumping water. The system resistance and cyclone head when pumping slurry are also shown.

The pump total head when pumping water at the required Q is

$$H_w = H_m/\text{HR} = 22.9/0.88 = 26.0 \text{ m water}$$

Figure 76 shows that the pump speed required to deliver 61.7 L/s of water against this head is 1,400 rpm.

The pump efficiency is 69% when pumping water. When pumping slurry, it is reduced by the efficiency factor to

$$69 \times 0.88, \text{ or } 60.7\%$$

The power input, P_i, required by the pump is given by

$$P_i = QH_mS_{SL}/\eta_m = (61.7)(22.9)(1.35)/(1.02)(60.7) = 30.8 \text{ kW}$$

where η_m is the efficiency when pumping slurry.

A 37-kW motor would give a 20% contingency margin.

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PART 6

Physical Separations